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POWER AND THE ENGINEER

DEVOTED TO THE GENERATION AND
TRANSMISSION OF POWER

ISSUED WEEKLY

VOLUME XXXII, PART II

July 1 to December 31, 1910

117270
13/7/11

Hill Publishing Co.
505 PEARL STREET
NEW YORK

[illegible]
v.52
no.7-12

POWER AND THE ENGINEER

INDEX FOR VOLUME XXXII, PART II

JULY 1 TO DECEMBER 31, 1910

EXPLANATORY NOTE.

Illustrated articles are marked with an asterisk (*). Book notices are denoted by a dagger (†). Cross references to a particular initial word may apply to any cognate word beginning in the same way. Thus, a reference from "Oil" to "Lubricant" would relate equally to 'Lubricating," "Lubrication" or "Lubricator." The cross references condense the matter and assist the reader, but are not to be regarded as conclusive. So, if there were a reference from "Boiler" to "Furnace" and if the searcher failed to find the required article under the latter entry, he should not regard it as useless to turn back and look through all the "Boiler" entries, or others that the topic might suggest, as he would have done had there been no cross reference. Letters are indexed under title or subject, general articles under writer's name as well. Not all articles relating to a given topic necessarily appear under the same entries.

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Power
and The Engineer

Vol. 32 NEW YORK, JULY 5, 1910 No. 27

THERE has been a great deal of discussion regarding the exceedingly small remuneration paid to operating engineers, and the subject deserves every attention that can be given to it. There is still another important member of the operative staff who should not be overlooked—the switchboard attendant.

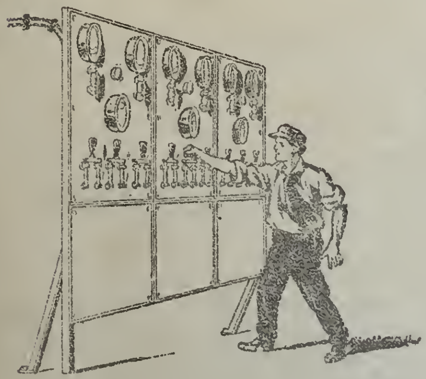
While a good operating engineer may cut a large slice from the coal bill, an inefficient switchboard attendant may add an equally large amount to the repair bill.

To be a good switchboard operator, a man should have had not only experience, but should also have acquired a certain amount of knowledge regarding the theory of electricity.

In the steam end of the plant, when anything goes wrong, it usually happens in plain sight or manifests itself in such a manner as to be easily traceable; but in the electrical end only the effect is evident at the switchboard, and it is up to the attendant to solve the cause and to decide almost instantaneously upon his course of action.

He has to use his brains and use them quickly.

An error in judgment at the critical moment, or the chance pulling of the wrong switch may mean hundreds of dollars of damage, as well as crippling the service.



A person with an excitable temperament should never be allowed to operate a switchboad.

It requires men of intelligence and sound judgment.

The question arises, can men possessing these requisites be obtained at the meager rate of wages paid for this class of service?

In New York City the average rate for a switchboard attendant is $3 per day, while a union electrician receives $4.50 per day. Without going into the respective requirements of the two—which would certainly be in favor of the former—and neglecting the question of continuous employment, let us confine ourselves to a consideration of the relative responsibilities.

The electrician is responsible only for the particular piece of apparatus upon which he is working. This is usually not in service, and before being put into use is inspected by the foreman.

On the other hand, the switchboard attendant may have the responsibility of a large part of the electrical service resting upon his shoulders.

Owing to the fact that the development in switchboard work has been so rapid, there is a younger class of men engaged in this line than in the steam end of the plant. Some of these men are highly capable, but the compensation is not attractive enough to hold them for any length of time, and they seek other fields of advancement.

The frequent breaking in of new men is not conducive to efficient operation and should not be tolerated for the sake of a small reduction in operating expenses.

Brains are needed in this position, and to secure intelligent men and retain them for any length of time demands a fair return for their services.

Hydraulic Turbines for 670-Foot Head

By H. Birchard Taylor

A distinct advance in the design and application of hydraulic turbines is marked by the construction of two 6000-horsepower reaction turbines which are to be operated under a 670-foot head at Noriega, Mexico. Owing to this high head, special attention had to be given to the proportioning of parts and to the means of regulation.

There are now in the course of construction, in the shops of the I. P. Morris Company, of Philadelphia, two hydraulic reaction turbines which will be applied to a head higher than has ever before been attempted for this type of waterwheel. They are to be installed in the power plant of the Michoacan Power Company at Noriega, Mexico.

The turbines are of the horizontal-shaft volute-casing, inward-flow, Francis type and each is designed to deliver at its flange coupling, 6000 horsepower, when operating under an effective head of 670 feet at a speed of 514 revolutions per minute. They will be directly connected to 3500-kilowatt, 60-cycle, three-phase generators, manufactured by the General Electric Company.

When delivering 6000 horsepower under the above conditions of head and speed, each turbine will require 98.7 cubic feet of water per second. Therefore, when both units are delivering full power, 197.4 cubic feet of water per second will be required. The water will be supplied to both turbines through a single penstock about 5600 feet in length.

It has been only within the last few years that the reaction type of turbine has been applied to heads much over 300 feet. Some of the more recent applications of the reaction turbine to high heads have proved conclusively that the erosion of the runner vanes in high-head turbines is due primarily to defective design, and not to foreign substances in the water, poor material, electrolysis, or chemical action.

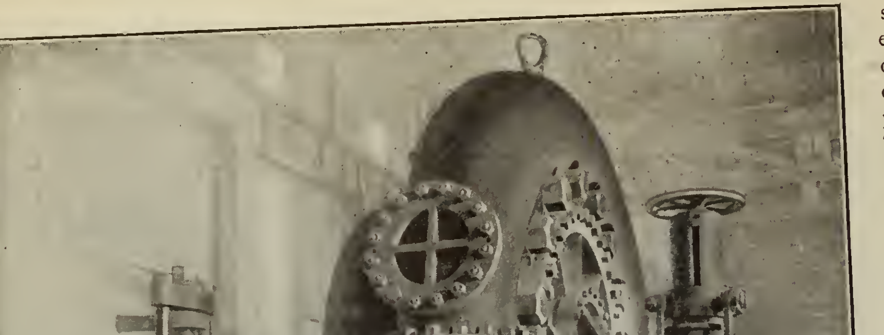


Fig. 1. View Showing Relief Valves

The successful operation of the four 18,000-horsepower turbines in the plant of the Great Western Power Company, on the Feather river, near Oroville, Cal., has greatly strengthened the confidence of engineers in the reaction turbine, as a type well adapted to compete with the impulse turbine, for heads probably as high as 1000 feet. The runners of the Great Western turbines are not all of the same material; two are made of bronze, one of cast steel and one of cast iron. All have been operating continuously for a year without showing any signs of erosion. The present head under which they operate is 420 feet, but this will be increased to 525 feet when the dam is completed.

Hydraulic engineers are awaiting with considerable interest the time when the Noriega turbines will be put into commercial operation, as the construction of these units marks a distinct advance in the art of hydraulic-turbine design and their successful operation will greatly broaden the field for the reaction type.

The turbines are of the single-runner, single-discharge type with outside operating mechanism. Each casing is made of cast steel in two sections, the division occurring on the horizontal plane which passes through the axis of the shaft. The normal pressure in the casing will be about 285 pounds per square inch, but owing to the wide fluctuations of pressure resulting from the sudden stopping and starting of the water column in the penstock, the turbine casing has been designed to withstand a pressure of 585 pounds per square inch. The sizes of the various parts of the turbine subject to this pressure are so proportioned that the resulting stresses will not exceed the elastic limits of the metals used in their construction.

Each casing is securely tied by 17 heavy forged-steel staybolts, which, aside from preventing excessive strains in the metal of the casing by taking a large proportion of the load upon themselves, are so arranged as to prevent an appreciable change in the width of the throat opening of the casing where the guide vanes are situated. Both casings have been subjected to a hydraulic-pressure test of 585 pounds per square inch. The accompanying illustrations show one of the turbines erected in the shop for this test.

The runner of each turbine is made of special bronze in a single casting and is securely bolted to a hub forged on the shaft. All the vane surfaces have been finished smooth so as to reduce to a minimum the frictional losses.

Provision has been made to balance the hydraulic thrust on the runner, but should any unbalanced thrust occur it will be taken care of by a thrust bearing located on the tail end of the shaft.

The forged-steel shaft which revolves in two ring-oiling bearings is made of such a diameter as to prevent any appreciable deflection at the runner. It is therefore possible to use scant clearances between the runner and the stationary parts, thus preventing unnecessary losses in efficiency by cutting down the leakage at the seals.

The water will be distributed to the runner through cast-steel movable guide vanes. Each vane has cast with it a fulcrum pin, which extends from both sides of the vane. The section of the pin on the generator side is short in length and extends into a bearing on the inboard distributer plate. The operating section of the pin, which is long, extends through a stuffing box and two bearings on the outboard headplate. Thus each fulcrum pin is held in alinement by three bearings.

The operating part of each pin is connected by a cast-steel lever and a cast-iron link to the cast-steel operating ring, which turns in a groove in the outboard headplate. The lower part of the ring is connected to the piston rod, which is controlled by the main operating cylinder mounted on the turbine casing.

The entire operating mechanism is so designed that should any two vanes close on a stick which has passed through the racks at the headgates, the two vanes will be capable of sustaining the entire force of the governor, without causing any permanent distortion of the vanes. This arrangement not only reduces the maintenance charges by increasing the life of the guide vanes, but it also prevents excessive leakage when the vanes are in the closed position.

The penstock will terminate in a cast-steel Y-pipe located outside of the power house and each turbine will be connected to a branch of this Y. Between each turbine and the penstock will be located a 36-inch, cast-steel body, bronze-mounted, hydraulically operated Ludlow gate valve. The operating cylinder of each valve is designed to operate the valve when acted upon by a pressure considerably less than that to which the valve itself will be subjected. Each valve is equipped with a 6-inch bypass for equalizing the pressure on both sides, but the cylinder is capable of operating the valve with the pressure on one side of the valve only.

The thrust-bearing pedestal is mounted on a subbase which in turn is mounted on a sole plate securely bolted to the concrete foundations. Thus, if it should be desired to dismantle the turbine, the bolts connecting the cap to the pedestal may be removed and the bolts which connect the pedestal to the sole plate may then be withdrawn. The subbase may then be slipped out from beneath the pedestal and the latter dropped down and removed from under the thrust collars on the shaft. The draft bend and the headplate containing the operating mechanism may then be unbolted and removed from the casing and slipped over the tail end of the shaft.

Owing to the high head under which these turbines will operate and the likelihood of the seals between the runner and the stationary parts increasing at a rapid rate, the turbines have been equipped with removable cast-iron plates at these points. These plates may be removed from time to time when the clearances become too large. Thus no permanent damage to the turbines will result from the action of the water.

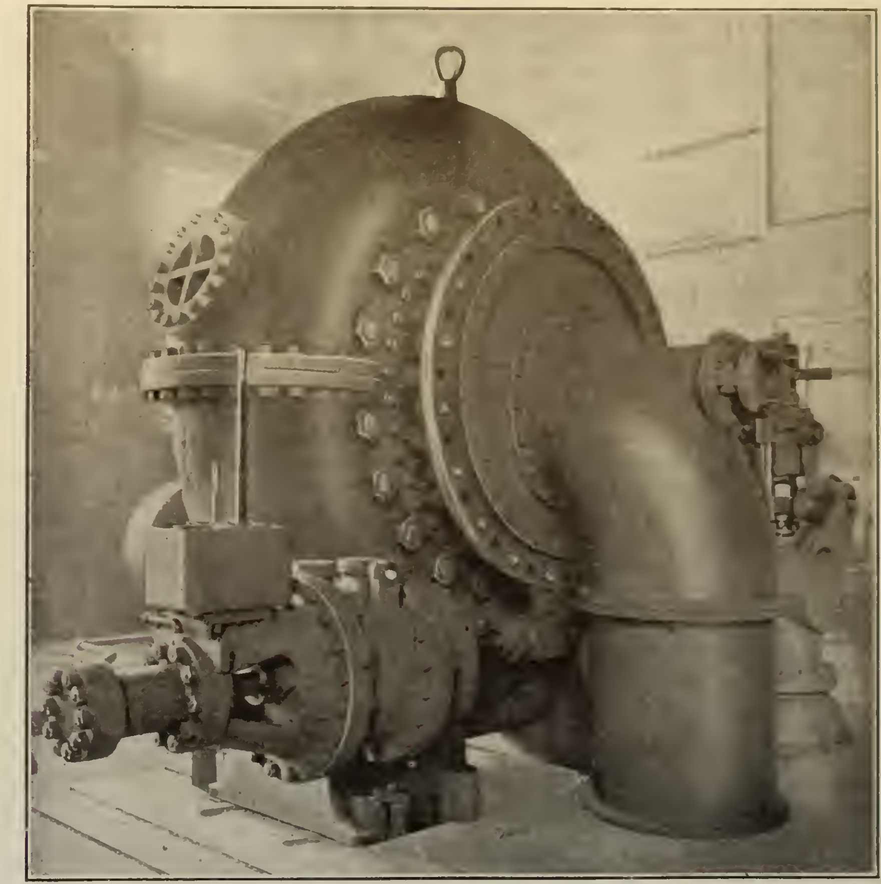


Fig. 2. Showing Operating Mechanism

Removable plates are also supplied on both sides of the throat of the casing where the guide vanes are situated. These plates are made of cast steel and will protect the turbine casing from wear caused by the scouring action of the water, as it passes with high velocity through the guide vanes.

Each turbine is supplied with a small hand-operated oil pump mounted on the operating cylinder and connected through piping to a jack cylinder superimposed upon the end of the operating cylinder. The piston rod which controls the operating ring passes through the main operating cylinder and terminates at a piston in the jack cylinder. In case the governor system should fail the turbine may be operated by the hand pump through the jack cylinder.

The problem of designing the turbines for good speed regulation has been an extremely difficult one, owing to the fact that they are fed through a penstock 5600 feet long and that the flywheel effect of the generator amounts to only 108,000 pounds at one foot radius.

The speed regulation of a turbine is not wholly dependent upon the installation of a first-class governor. Of course, a first-class governor is essential to good governing but the best governor constructed will not insure good regulation if the conditions are not favorable.

The principal factors entering into the regulation problem of every turbine which is supplied with water through a penstock are as follows:

The inertia of the water in the penstock.

The speed with which the turbine gates are opened or closed by the governor to suit the changes in the load.

The flywheel effect of the revolving parts of the turbine and generator.

Turbines which are fed by long penstocks are subject to wide fluctuations of the pressure in the wheel casing. The magnitude of the pressure change depends upon the length of the penstock and upon the rate of the change in the velocity of the water column in the penstock. The latter depends upon the velocity with which the gates are moved to suit the variations in the load on the turbine.

When the load suddenly drops, the gates are partly closed by the governor and the quantity of water delivered to the runner is thereby reduced to an amount required to balance the new load, at the normal speed. The closing of the gates reduces the velocity of the column of water in the penstock and, as it requires force to change the velocity of a body, this force is evidenced by a rise in pressure in the wheel casing.

Again, when a load suddenly comes on the turbine, the gates are opened by the governor and the quantity of water delivered to the runner is increased to an amount required to balance the new load at the normal speed. The opening of the gates allows an increase in the velocity of the water column in the penstock and as it requires a force to change the velocity of a body, this force is evidenced by a drop in the pressure in the wheel casing.

If the pressure in the wheel casing is increased or decreased, it is equivalent to operating the turbine under a higher or lower head and this change in the head exists during the time in which the gates are being adjusted to suit the change in the load. Should the gates be moved too rapidly, the pressure change will become a greater factor in controlling the speed of the turbine than the change in the quantity of water resulting from the movement of the gates.

To illustrate this point, let us assume that a portion of the load on the turbine drops off and the gates close to reduce the amount of energy supplied to the runner. The rise in the pressure will increase the velocity of the water through the gates and the quantity delivered through smaller gate openings will be considerably above the normal discharge for these openings based on a constant head. The product of the increased pressure times the quantity of water delivered per second is a measure of the energy supplied to the runner and this energy might be greater than that which would have been the case had the pressure remained constant during the closing of the gates. Thus, the whole scheme of regulation is upset.

In practice the pressure fluctuations are usually modified by the installation of standpipes, surge tanks and relief valves, but in all cases the allowable velocity of the gates is largely dependent upon the design of the penstock.

Assuming that the movement of the gates does not cause a change in the pressure in the casing, then if the load goes off and the gates close to suit the new condition of load, the excess energy supplied to the runner during the movement of the gates is taken up by the revolving parts and the speed of the turbine increases. For a given amount of energy the rise in speed depends directly upon the flywheel effect, or the weight multiplied by the square of the radius of gyration of the revolving parts. The greater the flywheel effect, the smaller the speed change.

Conversely, when the load on the turbine is suddenly increased and the gates open to suit the new load, the deficiency in the energy supplied to the turbine, during the movement of the gates, is supplied by the revolving parts as the speed is reduced. For a given amount of energy taken from the revolving parts, the drop in speed depends upon the flywheel effect.

Thus, it is seen that for good governing, it is desirable to have as great a flywheel effect as is practicable and in cases where the revolving parts of the generator will not supply the value sufficient to obtain the degree of regulation required, it is then necessary to increase the inertia of the revolving parts by the addition of a flywheel on the turbine shaft.

The various factors entering into the problems of regulation—the inertia of the water in the penstock, the relief valves, the standpipe, the flywheel effect, and the speed with which the gates are moved, are all more or less related and it is essential that they be carefully studied and each factor given its due attention by the engineer in making his guarantees of regulation.

In order to secure the regulation desired by the Michoacan Power Company, it was found necessary to increase the flywheel effect (Wr^2) of the revolving parts to 282,000 pounds at one-foot radius, by the addition of a cast-steel flywheel weighing approximately 20,000 pounds. It will be located between the generator bearing and the turbine bearing, and will be mounted and keyed to the generator shaft and bolted to the forged-flange coupling on the turbine shaft.

The flywheel is so designed that when operating at the runaway speed of the turbine, which will be about 900 revolutions per minute, the stress in the rim will not exceed the elastic limit of the material.

Owing to the question of safety as well as speed regulation, it is necessary that the variation of the pressure in the turbine casing, caused by the stopping and starting of the water column in the penstock, shall not exceed certain limits. In order to control these variations in pressure, each turbine is equipped with a mechanically operated, cast-steel body, relief valve connected to the turbine casing.

The relief valve is arranged to be actuated by a mechanical connection through a dashpot, to the operating ring of the gate-operating mechanism, in such a manner that, when the turbine gates are suddenly closed, the relief valve will open. The construction of the relief valve is such as to discharge practically the same amount of water that is shut off by the turbine gates.

The relief valve is also arranged so that it may be operated as a synchronous bypass, discharging water in inverse proportion to that being used by the wheel, thus keeping the velocity in the penstock constant and preventing variations of the pressure in the wheel casing. It may also be arranged so as to be used as a water-saving bypass, that is, when the turbine gates are suddenly closed, the relief valve will open and prevent an abnormal rise of pressure, and will then slowly close at such a rate as to keep the pressure in the pipe line within the required limits.

There will also be provided for each turbine and attached to the turbine casing a steel-body, pressure-actuated, relief valve which will be capable of discharging the full quantity of water used by the turbine. This relief valve will be operated by a rise of pressure in the pipe line, and will be so designed as to open in proportion to the rise in pressure and with a decrease in pressure, the valve will close and shut off a proportionate amount of water.

The gates of the turbines will be controlled by Morris governors of the double-floating lever, oil-pressure type. Each governor will be operated by a belt from the turbine shaft, and in addition is equipped with a motor so that it may be controlled from the switchboard. In order that the revolving element of the governor may be stopped during the operation of the turbine without disturbing the driving belt, a friction clutch is provided on the governor pulley.

The governing apparatus of each turbine consists of a governor, accumulator tank, three-throw oil pump and the interconnecting piping. The accumulator tank is divided into pressure and vacuum compartments. The oil pump is of the Deane, vertical, three-throw type and is driven from the turbine shaft by means of a belt. The interconnecting piping and fittings between the pump, governor, accumulator tank and operating cylinder are made of brass and all parts are so designed that the governor will control the turbine gates satisfactorily with a working pressure not exceeding 150 pounds per square inch.

When operating under an effective head of 670 feet at the normal speed of 514 revolutions per minute, each turbine is guaranteed to have an efficiency of at least 80 per cent. when delivering 6000 horsepower, at least 82 per cent. when delivering 5000 horsepower, and 80 per cent. when delivering 4000 horsepower.

Applying Patches to Boilers

By H. S. Jeffery

> *The method used in repairing a boiler plate depends upon the nature and location of the defect. Cracks may frequently be repaired by calking, while in other cases plugging has to be resorted to; or, if accompanied by a deformation of the plate, a patch may have to be applied.*

When it is necessary to patch a boiler in a section adjacent to the girth seam, a horseshoe patch may be employed, the defective plate being cut out in such a manner that a number of the girth-seam rivets are removed, and when replaced hold part of the patch in place. At *A*, Fig. 1, is shown the usual method of cutting out the defective plate. This method is recommended over that shown at *B* as it does away with a seam in the longitudinal plane.

When the patch is applied in this manner and on the inside of the boiler, to the small course, the shell sheet should be scarfed, as shown at *A*, Fig. 2. Should the patch be applied to the large course, then it must be scarfed instead of the shell sheet, and the inside and outside courses forced apart sufficiently to insert the patch between them, the whole being fitted as shown in Fig. 3.

When the shell sheet is scarfed, the operation has to be done when cold, and consists of tapering the edge of the sheet with a chisel, as at *A*, Fig. 4. This amounts to merely removing sufficient metal to prevent offsetting of the patch. On the other hand, when the patch is to be scarfed, the part concerned is heated and while hot is drawn out.

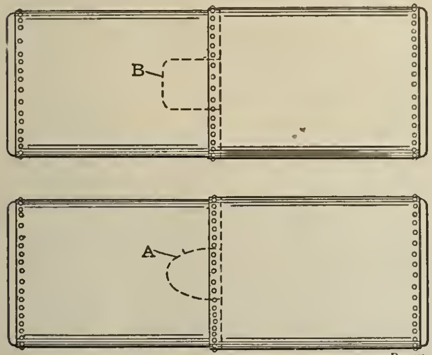


FIG. 1. CORRECT AND INCORRECT METHODS OF CUTTING

In order to insure a close fit the patch must be formed to whatever shape is required, and must be free from burs. The shell should also be free from all burs or foreign substances. The rivet holes may be either punched or drilled. If the work is performed in the boiler shop, then as many holes as possible, unless otherwise specified, may be punched by a power punch, and the balance punched by a hand punch or drilled. If the holes are to be drilled, it is advisable to employ guide lines, as the drill is liable to run to one side, and the guide line will indicate whether the drill is cutting the hole true.

Frequently it happens that the holes marked on one side of a plate can best be made from the other side, in which case the center-punch marks are transferred by the use of a tool, shown in Fig. 5, and called a "backmarker." The operation consists of placing the small hole *A* directly over the center-punch mark, and then marking through bole *B* on the other side.

PATCH ADJOINING THE HEAD

A patch on the shell sheet of a tubular boiler, having its smoke box formed by extending the shell sheet, as shown in Fig. 6, should be applied on the inside and scarfed to the calking edge *A*, thus permitting the rivets *B* to pass through the head, the patch and the shell sheet. Since the scarfed end of the patch is inserted between the head and the shell sheet, the latter will be forced downward at *C*. Before the rivets are driven the shell sheet should be fitted close to the patch, and the latter close to the head. The rivets should be driven from the outside of the boiler, both on account of accessibility and for the reason that the rivet hole on the fire side will be filled.

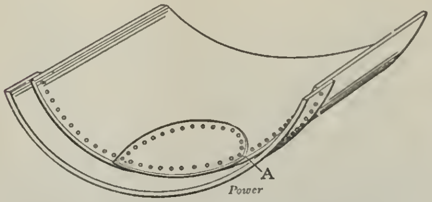


FIG. 2. PATCH WITH SHEET SCARFED

When a tubular boiler is constructed with the flange of the head inside the boiler the patch is applied inside the shell sheet, with the end of the patch scarfed and inserted between the head and shell.

PATCHING THE HEAD

Frequently a portion of the flange of the head of a boiler, tank or air reservoir cracks, or the metal so suffers from corrosion that a patch is necessary. When the defective part has been cut out, the patch should be shaped so as to be riveted partly to the head, as shown in Fig. 7. When a patch is thus applied, the ends of the patch are scarfed and inserted between the shell sheet and the head, and the latter at that point is slightly forced up. In every case the holes should be fair and the rivets well driven so as to completely fill the hole. The calking should be done with a round-nosed fuller and the lap should not be greater than one and one-half times the diameter of the rivet hole.

CRACKED FURNACE SHEETS

The firebox or furnace sheet of a locomotive, traction, or similar type of boiler frequently suffers from cracks at the stay-bolt holes, as shown in Fig. 8; these cracks often extending from one stay-bolt hole to another. How to best repair a fracture of this nature depends upon the length and the location of the crack. Frequently the crack can be closed by calking and gives no further trouble, but a long crack in any part of the firebox, however, cannot be calked over. This is also true of a short crack which is in contact with or very near the bed of the fire.

A small crack is often made steam tight by plugging, as shown in Fig. 9. The hole is usually drilled for a ½-inch tap, a solid plug inserted and then calked over. If more than one plug is needed, which would be the case with a long crack, the holes should be drilled so as to overlap. However, all the holes should not be drilled at one time. Assuming the crack to extend from stay bolt to stay bolt, the holes should be laid off with about ¾-inch centers. They should then be tapped, the plugs inserted and cut off to the desired length, usually about ⅛ inch beyond the sheet. Holes are then drilled between them so as to take in a portion of the adjacent plugs. These holes are tapped and plugs inserted and cut off as in the first instance. The entire row is then calked close to the sheet and forms a solid ridge, thus practically locking the plugs together. This operation is spoken of as "sewing up the crack." The work requires time and skill, and if satisfactory results are to be obtained, the plug

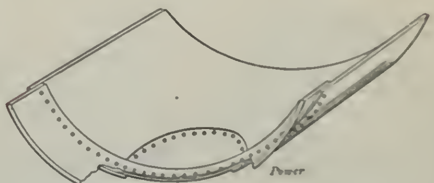


FIG. 3. PATCH INSERTED BETWEEN THE INSIDE AND OUTSIDE COURSES

centers should be as uniform as practicable.

BAGGING

The cracking of the furnace sheet is not the only cause for removing a part of the sheet. The plate may suffer from bagging to the extent that a portion of it may have to be cut out and a patch applied. Bagging is usually caused by overheating of the plate, or by too great a spacing of the stay bolts. If the water space or water leg, as it is sometimes called, is allowed to clog up with foreign substances, such as mud and scale, then the sheet will become overheated and bagging will result. Where a sheet bulges between the staybolts due to this cause, the sheet is said to be "mud-burnt," and, in the course of time, the stay bolts pull through the sheet. The spacing of the stay bolts too far apart, even though they are strong enough to carry the load, will result in bagging of the plate. The size of the stay bolts and their pitch must be in accord with the thickness of the plate that has to be supported.

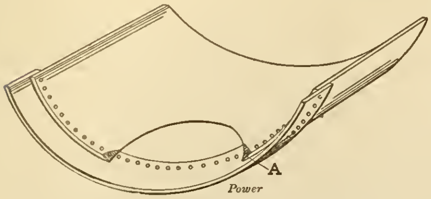


FIG. 4. PLATE PROPERLY SCARFED

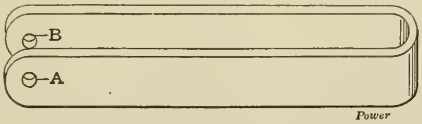


FIG. 5. BACKMARKER

APPLYING A PATCH

Firebox patches should be riveted whenever possible, although they are frequenty applied with patch bolts. Applying a patch so as to utilize the stay bolts in lieu of rivets or patch bolts does not give satisfactory results owing to the difficulty of keeping the seam of the patch steam tight; the stay bolts being screwed into both the patch and the sheet and hence not permitting the patch to be drawn tightly against the sheet. Even if this should be accomplished the pitch is excessive; that is, a 2-inch pitch is too great for ¾-inch patch bolts.

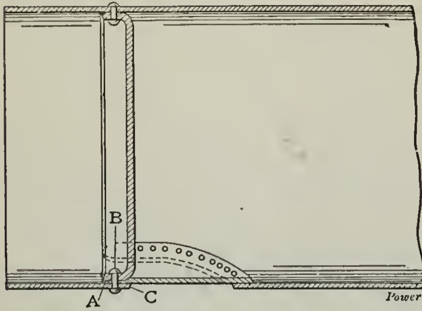


FIG. 6. METHOD USED WHEN SMOKEBOX IS CONTINUOUS WITH SHELL

Whether riveted or held with bolts, a patch should be applied as in Fig. 10, which shows half a patch. The first operation is to cut out the defective part, and if the sheet is cracked it is well to cut out an inch or so beyond, for in many cases the fracture extends further into the sheet than can be seen from a surface inspection. The dotted line indicates how and where the sheet should be cut in regard to the stay-bolt holes, and to avoid sharp corners. The distance *A* is approximately 75 per cent. of the pitch of the stay bolts. The distance *B* must be sufficient to permit a cape chisel and a ripper to clear the stay bolts. The distance *C* is no set standard. Some boilermakers make this distance 1½ times the diameter of the patch bolt or rivet, though more often this figure is 1⅛.

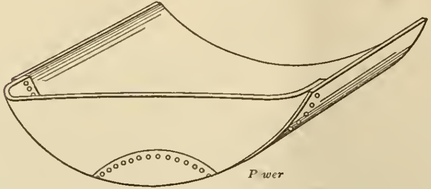


FIG. 7. PATCH ON FLANGE OF THE HEAD

The pitch of the patch bolt or rivet should be as uniform as practicable, the better practice being to lay off the holes, say for one quarter, and make the other quarters accordingly. Some cut the holes in the furnace sheet first, and then with a sheet of heavy paper make a templet of the holes and drill the patch accordingly. A better practice, however, especially if patch bolts are to be used, is to drill a few holes in the furnace sheet to allow the patch to be bolted in place, and then drill the balance of the holes. Still another custom is to punch small holes in the patch, bolting the latter to the sheet as described, and drill the holes in the furnace sheet from the patch. The holes in the patch prevent the drill from running out, which is liable to occur if the patch is bolted in place with a few bolts, when drilling the holes in the patch and the furnace sheet in one operation. However, both of the last two methods insure fair holes, and are superior to the paper-templet method.

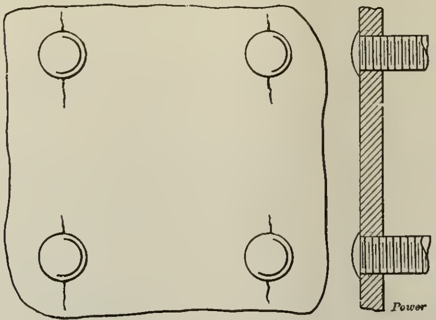


FIG. 8. CRACKS AROUND STAY BOLTS

If the centers of the respective holes are in line and the holes in the patch are properly reamed out, no trouble should be experienced in keeping the alinement.

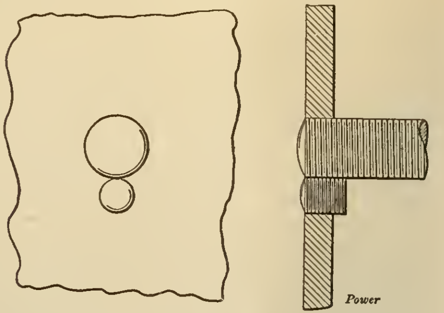


FIG. 9. PLUG INSERTED TO STOP CRACK

The countersinking of the holes in the patch is usually performed on a drill press, there being no objections to this practice if the patch is to be applied with rivets. But when patch bolts are used the above method of countersinking is not advisable, because of the difficulty in tapping the holes in the furnace sheet so the countersunk head of the patch bolt will fit the countersunk hole as intended.

The above difficulty is overcome by tapping the holes and then countersinking them while the patch is in place, a special tool with a guide being employed. Following the countersinking of the holes when the patch is bolted in position, the patch should be removed and all burs, cuttings, etc., removed, so as to leave nothing in the form of foreign substances between the patch and furnace sheet, unless it be a strip of copper, which is sometimes used as a gasket.

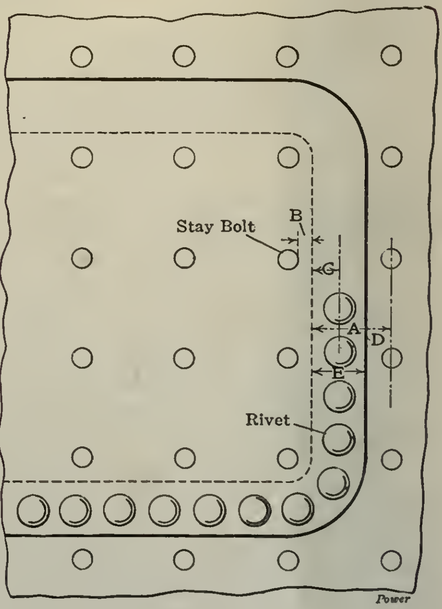


FIG. 10. HALF A PATCH

The use of a strip of copper is advocated by some boilermakers and condemned by others. If used it should be thoroughly annealed and placed only at the lap, that is, its width should be equal only to the distance *E*. Fig. 10, the copper line not extending to the calking edge *D*.

APPLYING THE PATCH BOLTS

It is not good practice to completely insert one patch bolt before applying the next. Instead, they should all be partially screwed into place and then gradually tightened up and the square head twisted off. To facilitate twisting off the head without undue strain, the neck is nicked with a chisel. A calking tool, called a "fuller," is used to work the edges of the patch bolt to the patch. After the bolt has been fullered down, another calking tool, called a "frenchman," is used to cut away the edge of the bolt, but the cut in operation does not imply a recess, which is often made by inexperienced workmen.

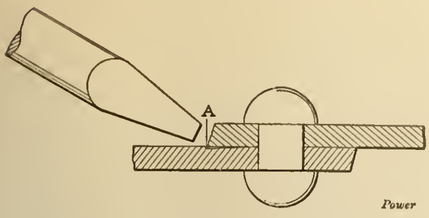


FIG. 11. IMPROPER CALKING TOOL

The patch may or may not have been leveled before finally applying, and if not, it should be chipped and calked. If chipped in place, there is no reason why the furnace sheet should bear marks of the chisel, providing the calking is done with a fine finishing tool.

A SOFT PATCH

What is known as a soft patch is a patch applied over a hole or crack without cutting out the defective part, but it should never be used on a furnace sheet or where it will come in contact with intense heat. At the best it should be employed only as a means of temporary repair. To install a soft patch so as to require no calking, it should be a close fit, and the inside of the patch and the outside of the sheet should be coated with cement paste. Care should be taken to see that the surfaces are free from oil or grease, to insure the adhesion of the cement. When the patch is in place with the cement paste between the patch and sheet, the bolts should be tightened up to within about ⅛ inch of home. The joint, depending upon the class of cement paste, should not be finally finished at the outset, it being the better practice to allow the cement paste to set slightly and then screw up on the nuts. If possible, the patch should stand ten hours before the boiler is placed in service.

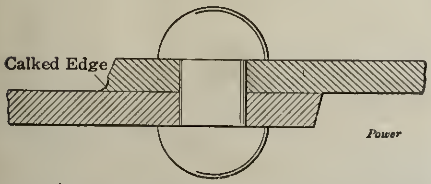


FIG. 12. EDGE UPSET WITH ROUND-NOSED FULLER

LEAKY FIREBOX SEAMS

Many leaks in firebox seams are due to faulty construction, the calking very often being to blame. The shape of the calking tool is an important factor, likewise the position in which it is held. Usually the plate is beveled, not to exceed 30 degrees. A square-end calking tool, as shown in Fig. 11, should never be used as it will cause the sheet to be calked as at *A*, cutting into the plate and forming a ridge along the edge.

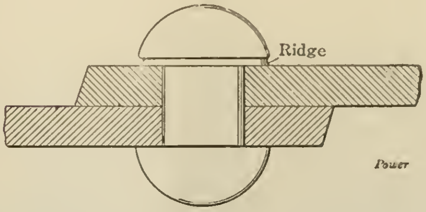


FIG. 13. RIDGE MADE BY CALKING TOOL

The seam should be calked with a round-nosed fuller, thus upsetting the plate, as shown in Fig. 12. This tool causes the swelling to extend further into the plate, though care should be taken not to use too small a fuller—the size depends upon the thickness of the plate that is to be calked—otherwise it will act as a wedge and press the outer surface up.

The proper calking of a leaky seam, which is subsequent to the original calking, is a matter of utmost importance. If the leak is a small one, it can be stopped by recalking, that is, without chipping the sheet. The operation should be begun an inch or so beyond the leak and directed toward it with very light blows. Frequently, if the fuller is placed directly over the leak and a hard blow is struck, the plate at that point will be calked, but a new leak will appear on each side of the old one.

If the seam leaks to any extent it is the best practice to chip and then calk it, ⅛ inch usually being sufficient to remove the recess formed by the original calking. When chipping, care must be exercised not to cut into the sheet.

LEAKY RIVETS

Any rivet that leaks does not thoroughly fill the hole, and the calking of the rivet head is nothing more than fencing in the leak. The water and steam often find an outlet through the small spaces existing between the rivets and rivet holes. As the ratio of volume between steam and water is about 1700 to 1. at a temperature of 212 degrees. but very little water is needed to cause the impression that the leak is one of great proportion.

A button-head rivet whose edges have not been brought down well to the sheet. as shown in Fig. 13, is the most difficult kind to calk, as the ridge becomes enlarged each time the operation is repeated. The rivet head should be calked as shown in Fig. 14, and while this method tends to pry up the rivet head, the effect is not as pronounced as in Fig. 13.

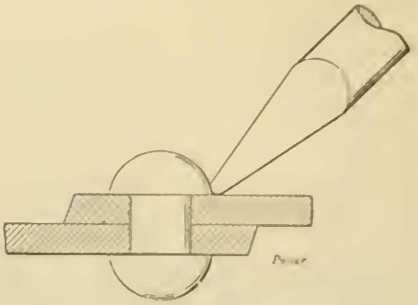


FIG. 14. PROPER METHOD OF CALKING

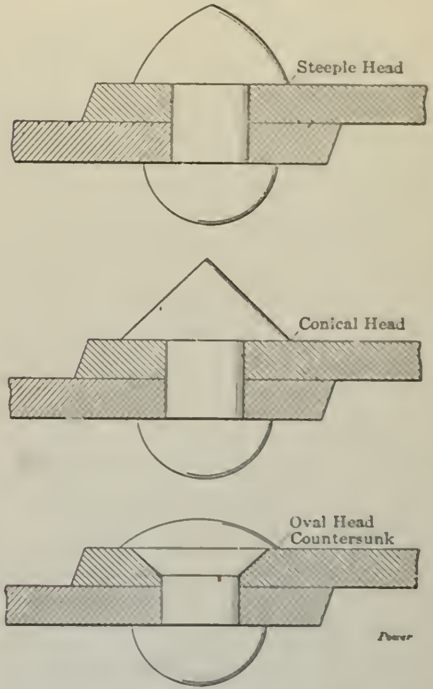


FIG. 15. COMMON TYPES OF RIVET HEAD

The oval-head countersunk rivet, the steeple head and the concial head, the latter being a compromise between the button head and the steeple head, are more readily calked, as the reason that the calking tool can be readily applied to the edges. These types of rivet are shown in Fig. 15.

Central Station Steam Heating*

By A. D. Spencer

Description and operating cost details of a live-steam central-station heating system and an exhaust-steam central-station heating system in Detroit, Mich.

The Central Heating Company, of Detroit, Mich., operates two heating systems. System *A* is located in the business section. At the present time it operates with live steam. The plant is designed, however, for the ultimate installation of noncondensing turbo-generators. During the past winter the system served 400,000 square feet of radiating surface, and furnished steam for power and other purposes equivalent in amount to that required for about 100,000 square feet of radiation.

System *B* is located in a residential section, and uses the exhaust from three McIntosh & Seymour single-cylinder engines which are direct connected to generators and which are capable of developing 1300 kilowatts against the back pressure of the heating system. Service is furnished for about 300,000 square feet of radiation. Babcock & Wilcox boilers equipped with Jones stokers are used. The boiler capacity is 2465 horsepower. The station is equipped with 1000-kilowatt capacity of motor-generator sets and is connected to the 4600-volt transmission system of the Edison Illuminating Company. The generating equipment is operated whenever there is a demand for exhaust steam, some of the direct current generated being used in the district and the balance converted and delivered to the 4600-volt transmission system. When the district demand for electricity is greater than the equivalent demand for exhaust steam, the excess current is taken from the transmission lines and converted. When the demand for steam exceeds the capacity of the engines, live steam is used. In brief, the engines are operated as nearly as possible to furnish exhaust-steam requirements and no more. With the present heating load it is necessary to furnish some live steam during the winter months. On the other hand, it is generally necessary to use some electricity from the transmission lines during the lighting peaks.

TABLE 1.
OPERATING DATA.

1909.	System "A."	System "B."	Combined.
Pounds steam sold—heating	167,700,000	141,700,000	309,400,000
Pounds steam sold—power, etc	41,300,000	5,250,000	46,550,000
Total steam sales	209,000,000	146,950,000	355,950,000
Electricity sold (kwh.)		4,523,200	4,523,200
Earnings per M. lb. steam	47.5	48.2	47.8
Earnings per M. cu.ft. space	2.10	3.90	3.0
Earnings per sq.ft. radiation	21.4	23.7	22.6
Number of customers	274	396	670
Cubic feet space	37,200,000	17,300,000	54,500,000
*Square feet radiation	372,400	287,800	660,200
Ratio radiation to space	1–100	1–60	1–83
Cost of coal per ton	2.50	2.55	2.53
Lb. coal per M. lb. steam sold	220	220	220
Lb. steam sold per lb. coal	4.6	4.6	4.6
Lb. steam sold per M. cu.ft. space	4,500	8,200	6,350
Lb. steam sold per sq.ft. radiation	450	490	470
Heating efficiency, per cent	37	37	37
Steam sold in per cent. of generated			55.2
Steam used in auxiliaries in per cent. of generated			19.2
Line losses in per cent. of generated			9.8
Total sold or accounted for in per cent. of generated			84.2
Unaccounted for in per cent. of generated			15.8
			100.0

*In these figures no allowance is made for hot water heaters, of which there are about 200 on System "B" and 100 on System "A."

The reasons for selecting a steam rather than a hot-water system are: A smaller investment is required; steam is capable of serving other requirements, such as for laundry work, cooking and power; steam can be adapted to hot-water installations by the use of transformers; and to hot-air systems by the use of indirect radiation; as hot-water service cannot be metered, charge for it must be made on a basis of radiation or on some other approximate plan.

SYSTEM *A*

The plant of system *A* is equipped with eight 500-horsepower Stirling boilers Jones stokers, forced draft, feed-water heaters and ash-handling apparatus. The building contains coal-storage bins of 1500 tons capacity and is equipped with cranes for handling coal from the alley in five-ton hopper wagons.

DISTRIBUTION LINE

In a live-steam system, consideration of back pressure is not present; hence, the allowable line pressure is limited only by distributing conditions. The higher the pressure, the smaller the size of the distributing mains. On the other hand, increased pressure means increased trap troubles and other troubles in customers' installations. Fifteen pounds, average, in the heating mains was chosen as a sufficient pressure to handle some old 15-pound installations and to do cooking and miscellaneous service. The customers having lower-pressure systems were required to install regulating valves to reduce the pressure. The distributing pressure has been increased from year to year, as increase in load made it necessary to increase either the pressure or the size of the mains.

To furnish power to operate steam pumps, and for other purposes, a system of 110-pounds pressure was installed in the heart of the business district. Incidentally, this system was designed for use as a feeder to the low-pressure system. The 110-pound service has proved extremely unsatisfactory, principally for lack of a steam meter.

Tunnel construction is used in the district where high-pressure steam is furnished. This opens the high-pressure mains to inspection and repair. The tunnel system is 3600 feet long and is at an average depth of 30 feet below grade. The standard tunnel is horseshoe shaped and 5 feet high. The construction is brick and hydraulic cement. Manholes are located about every 300 feet. Services are taken off at service manholes which are located wherever needed. Drainage flows to a sump at the plant and is siphoned out. An exhaust fan at the plant maintains a circulation of air and prevents accumulation of moisture. The normal temperature in the tunnel is about 150 degrees Fahrenheit, which is reduced by local ventilation when work has to be done in any section.

The 110-pound system consists of 3300 feet of 6-inch and 8-inch mains covered with magnesia and some hair felt; 1100

TABLE 2.

Cost in cents per 1000 pounds of steam sold in 1909.

	Plant A.	Plant B.	Average.
Production	32.7	38.0	34.9
Distribution	2.9	3.1	3.0
Sales and collections	0.4	0.4	0.4
General	2.5	3.6	3.0
Injuries and damages	0 1	0.1	0.1
Insurance and taxes	4.2	4.7	4.4
Electricity sold (credit)		(15.4)	(6.4)
Total of above items	42.7	34.5	39 5
Depreciation	6.7	8.9	7.6
Interest	11.6	16.1	13.1
Total cost	61.0	59.5	60.2
Profit to make total returns on investment 8 per cent	6 2	8.2	7.0
Total	67.2	67.7	67.2
Earnings	(47.5)	(48.2)	(47.8)

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*Abstract of paper read at the meeting of the National District Heating Association, at Toledo, Ohio, June 1, 2 and 3.

feet of 1-inch to 3½-inch service connections and a 10-inch tie line 4100 feet long connecting plant *A* to system *B*. This is to assist in maintaining pressure in the nearest part of the *B* system and serves also to divide the load between the two plants in the manner which at the time is most economical.

The low-pressure system in the tunnel consists of 3500 feet of 6-inch to 12-inch pipe mains covered with magnesia; also 1800 feet of 1-inch to 4-inch services. In addition to the tunnel construction and the tie line, there are 18,700 feet of underground mains of various sizes from 2½ inches to 8 inches and 91,300 feet of services from 1 inch to 6 inches in size. The construction is full-weight wrought-iron pipe insulated with asbestos paper and wooden log. Expansion joints of the copper-diaphragm type are provided every 65 feet and pipes are anchored at these points. Manholes are provided in each block and mains are graded to cause line condensation to flow to these manholes, whence it is trapped to the sewers. All underground construction is provided with tiled drain and is laid in 2 inches of crushed stone and covered with tar paper. Service connections are taken off from the top of the pipe at anchored points. Steam is furnished on all lines from October 1 to June 1. Lines carrying power are kept alive throughout the summer.

TABLE 3.

Cost (in cents) per square foot radiation.

1909.	A.	B.	Average.
Production	14.7	18.7	16.7
Distribution	1.3	1.5	1.4
Sales and collections	1.2	1.2	1.2
General	1.1	1.8	1.4
Injuries and damages	0.0	0.0	0.0
Insurance and taxes	1.9	2.3	2.1
Electricity sold (credit)		(7.6)	(3.0)
Total of above items	19.2	17.0	18.5
Depreciation	3.0	4.4	3.6
Interest	5.2	7.9	6.3
Total cost	27.4	29.4	28.4
Profit	2.8	4.0	3.3
Total	30.2	33.4	31.7
Earnings	(21.4)	(23.7)	(22.4)

SYSTEM B

The distribution arrangements of system *B* are practically the same as those of system *A*, except that there is no tunnel construction and, as the distribution pressure is but five pounds, the mains are considerably larger.

There are in this system 29,900 feet of 2½-inch to 18-inch low-pressure mains and 26,900 feet of 1½-inch to 5-inch services; also one 4-inch, 80-pound line 800 feet long laid in concrete, furnishing heat and power to a hospital.

The weak points of this system are in the customer's installations. Regulating valves fail to operate, causing excessive high or undesirably low pressure. Traps fail to open, thus flooding the system, or fail to close, thus allowing steam to be lost to the sewer. Automatic air valves fail to open, causing an accumulation of air in the system. Meters operate unsatisfactorily under the extremely severe conditions of heat, moisture, dirt and steam from defective traps.

Efforts are made to minimize these troubles by inspecting installations before service is turned on and advising customers to use proper equipment and methods of installation, and by refusing to connect service when, in the judgment of the company, the installation is not in condition to give satisfactory results to the customer or to the company. The company carries repair parts for certain lines of valves and traps, and makes all necessary repairs, charging to customers the cost of the material only. No other makes of valves and traps will be approved unless customers employ an engineer competent to make repairs. Continual study has been made of the meters and considerable improvement in metering has been effected.

Meters are read and inspected every six days, and traps, valves or meters out of order are quickly brought to the attention of the company and repaired. In case a meter is stopped it is necessary to estimate the consumption for a few days only and not for the entire month.

Twenty-four-hour emergency service is maintained and all ordinary installation troubles are remedied by the emergency men.

STEAM LOSSES

Table 1 gives steam sold, steam lost, etc. The steam used in auxiliaries is practically all used to heat the boiler-feed water as the auxiliaries exhaust into the heaters. At plant *B* the heater is connected to the exhaust heating line, and the feed water is brought to a temperature of about 220 degrees.

Steam loss is calculated by the use of a radiation factor determined by actual metering of line losses. For a number of years meters were maintained on line bleeders, and actual line condensation per square foot of line radiation was determined. The condensation per hour per square foot of radiation was found to be 0.086 pound on 20-pound lines and 0.051 pound on 5-pound lines. These factors are for log construction. No data have been obtained on concrete construction. Radiation losses in the tunnel have been calculated from handbook data. Some part of the unaccounted-for steam may be due to imperfect metering, to steam losses by leakage, and to errors in estimating evaporation. It is possible that the progressive deterioration of the wooden-log insulation has increased the radiation losses.

TABLE 4.

Present Rates.

Monthly bills for steam consumption aggregating—	Gross.	Net.
0 lb. to 12,500 lb	58	52.2
12,500 lb. to 25,000 lb	57	51.3
25,000 lb. to 37,500 lb	56	50.4
37,500 lb. to 50,000 lb	55	49.5
50,000 lb. to 75,000 lb	54	48.6
75,000 lb. to 100,000 lb	53	47.7
100,000 lb. to 200,000 lb	52	46.8
200,000 lb. to 300,000 lb	51	45.9
300,000 lb. to 400,000 lb	50	45.0
400,000 lb to 500,000 lb	49	44.1
500,000 lb. to 600,000 lb	48	43.2
600,000 lb. to 800,000 lb	47	42.3
800,000 lb. to 1,000,000 lb	46	41.4
Above 1,000,000 lb	45	40.5

Minimum monthly charge, 5000 lb.

COSTS

Operating costs are set forth in Table 1. Table 2 gives the cost per thousand pounds of steam sold. In Table 3 the cost per square foot of radiation is given, and Table 4 gives the existing scale of charges per thousand pounds of steam consumed. In establishing these rates, the idea was to make the cost to the customer about the same as his own cost had been, as it was believed that, costs being equal, the advantages of the service would bring the business. Data on 40 residences showed that the average cost of heating by coal in Detroit was 4.60 dollars per thousand cubic feet of space with coal selling at 6.75 dollars per ton. Comparing this figure with the actual earnings for system *B*, it is seen that the actual earnings are 15 per cent. under this. At the present price of hard coal, 7.50 dollars, the customer's cost would be 5.10 dollars per thousand cubic feet.

The following rules were recently found hung on the wall of a large power plant in Richmond, Va., which, if observed, would doubtless eliminate a great many accidents in the boiler room:

Do not empty the boiler while the brickwork is hot.

Do not feed cold water into a hot boiler.

Do not allow filth to accumulate around the boilers or boiler room.

Do not leave your shovel or any tool out of its appointed place when not in use.

Be alert and ready-minded about the boiler and furnaces.

Do not read newspapers while on duty.

Do not fail to see that there is plenty of water in the boiler in the morning.

Keep the water level at the same hight in the boiler during the running period.

Do not allow anyone to talk to you when firing.

Do not allow water to remain on the floor about the boilers.

Do not fire too quickly.

Do not fail to blow off the boiler once or twice a day, depending on the condition of the water.

Do not fail to close the blowoff cock when blowing down, when the water in the boiler has lowered 1½ inches.

Daily see that all safety valves move freely and are tight.

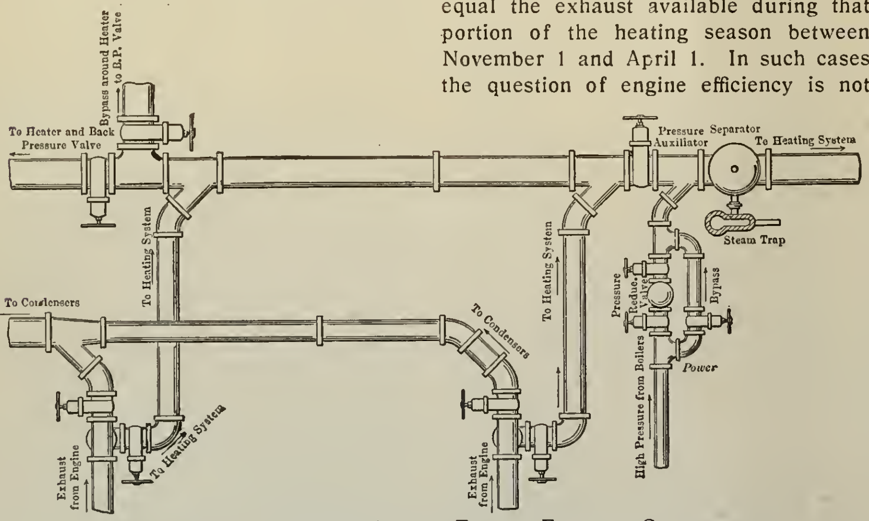
Factors Affecting Cost of Heating*

By Charles R. Bishop

> *For the successful operation of a heating plant or a combination heating and electric generating plant, all the factors entering into the generation and utilization of the steam and the maintenance of the system should be carefully noted. For this purpose complete daily records are an absolute necessity.*

The first factor requiring consideration in any heating plant is the thermal value of the fuel burned, whether it be coal, wood, oil or natural gas. Many companies are now purchasing coal upon a basis of analysis, but where this is not practicable, it is advisable to thoroughly test the different grades under actual operating conditions to determine the number of pounds of steam which can be produced not only per pound of fuel burned, but more particularly, per dollar expended.

With a good quality of fuel and water and a generating plant containing economical boilers of the proper size, attention should be given to the means of utilizing the greatest percentage of heat units in the fuel to be burned. In this connection, it is considered good practice to equip plants of 1000 or more horsepower with mechanical stokers adapted to the kind of fuel used. Such an installation not only increases the efficiency but decreases the firing cost per ton. In hand-fired plants, one man can attend to the coal, water and ashes for 200 horsepower, while with mechanical stokers and overhead bunkers he can easily handle from 2000 to 3000 horsepower. It is also advisable to install coal- and ash-handling apparatus for cutting down the labor costs.



Arrangement of Heating System Taking Exhaust Steam from Several Engine Units

Proper feed-water heating and clean boilers will also result in a marked fuel economy.

After considering the factors which enter into the generation of steam, the next question to be taken up is its utilization. If the plant in mind is a combination electric-generating and steam-heating plant, it will first be assumed that it intends to have or does have a market sufficient to utilize the full amount of exhaust steam at such times as it is developing the greatest electrical load, and that its steam demand will at least equal the exhaust available during that portion of the heating season between November 1 and April 1. In such cases the question of engine efficiency is not as serious as engine capacity, up to a point of approximately 65 per cent. of engine capacity, although this figure varies with different companies, due to the proportion of summer to winter maximum electric load. Some companies desire to install a sufficient number of high-efficiency engines or turbines to permit of summer operation under economic generating costs, while others believe the installation of exhaust turbines for summer operation is advisable.

It is a fact that engines act, in a sense, as reducing valves between the boilers and the heating system, but while performing such functions produce mechanical energy without appreciably lessening the heat energy; therefore, were there a heat demand in excess of the amount of steam exhausted by a high-efficiency engine, the deficiency in steam would have to be made up direct from the boilers, and thus the quantity of steam generated in the boilers would not be less than if all had passed through the engines in the production of a given quantity of engine horsepower or its equivalent in kilowatt-hours. By this statement, I do not mean to convey the idea that it is advisable to install or continue to operate engines of very low efficiency; I desire to more particularly call to mind that it is not necessary to successful operation and low kilowatt cost in combination plants to install engines of the highest efficiency.

Where the station contains several engines, an additional exhaust main should be installed, so that any engine can be run condensing or exhaust to the atmosphere, while other engines are exhausting into the heating mains. Such an arrangement of piping is shown in the accompanying illustration.

Transmission losses in a heating system are affected by the following factors: Efficiency of insulation; outside water coming into contact with the distributing mains; air currents coming in contact with the distributing mains; and actual leakage of steam. Frequent inspection of the distributing mains should be made and street traps should be kept in perfect working condition.

Normal transmission losses are practically constant in properly installed mains and should never exceed 0.045 pound per hour per square foot. The percentage of loss to output depends upon the load factor and should not exceed 15 per cent. at times of minimum demand nor 1½ per cent. during periods of maximum demand with an average of about 4 per cent. during the heating season, which lasts about five months.

The rate charged the consumer, while taking into consideration the total cost to the company with profit added, must still be reasonable and fair and is naturally dependable upon local conditions. Sliding-scale rates are usually preferable, as they automatically fix a lower average price per 1000 pounds to the large consumer than to the small consumer.

The degree of success of a heating and power plant depends largely upon the manager's knowledge of all the conditions existing at the plant, in the distributing system and at the consumers' premises. Such information can be had and the economical operation of the plant maintained only by a system of complete daily records in such form as to be readily comparable for reference at all times.

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*Abstract of a paper delivered before the National District Heating Association, at Toledo, Ohio, June 1-3, 1910.

The Operating Engineer's Problem

By Hubert E. Collins

The following thoughts were offered by Mr. Collins at a meeting of the Modern Science Club held recently to consider the proposed organization of an Institute of Operative Engineers. He points out the entirely inadequate remuneration received by operating engineers in comparison with those paid in other vocations, and how the proposed organization may be expected to improve their condition.

The editorial in POWER of February 8, 1910, issue, entitled "Can You Beat It," really brings up the most vital problem relating to the material welfare of the operating engineers' vocation. The correct solution of this problem is the aim and object of all engineering organizations and individuals who would better their own conditions and that of their fellow workers.

The fundamental fact brought out in this editorial was that engineers work longer hours and receive much less pay for services than many kindred workers who are tradesmen only. The case cited was not an isolated one, but is generally true.

An all-round engineer, for instance, is a trained machinist, for it is a part of his training to become familiar with this work. In this day of advanced operating engineering the man in charge has also been trained in other vocations.

To realize the comparative remuneration received by an operating engineer and a pipefitter, we have but to remember the training of the two.

The operating engineer first began as a machinist. While an apprentice in the machine shop he was taught pipe fitting better than 99 out of 100 pipefitters who learn that one vocation only. In fact, if this apprentice boy were kept at pipe fitting he would feel disgraced or he would never make a good mechanic. Now this training as pipefitter is only a small part of that of a machinist, and the machinist trade is only a small part of the equipment an operating engineer needs in an uptodate plant. Yet the fact remains that operating engineers as a class are paid less and work longer hours per day than steamfitters, plumbers, carpenters and masons. Many operating engineers could occupy a position in any one of these trades with credit, but how many members of any one of these trades could take the operating engineers' job? This is not mentioned to belittle the other trades. It is only to emphasize the fact which does exist. The operating engineer does not get the compensation due one who fulfils the requirements of a good man in this vocation. Many work on dissatisfied, knowing that this condition grows more acute as the years go by. Others become disgusted and leave the field for more profitable employment in other trades. Some who realize the conditions do what they can to aid themselves and others to better conditions in the profession.

The solution is to organize and make better engineers first; then better pay will come naturally. To organize first for better pay irrespective of qualification is wrong to both employer and employee if we are just. Before the organization of operating engineers is brought about for the permanent betterment of the vocation we must consider the elements that are to make up this body:

1. The theoretical man who enters the practical field better equipped for ultimate qualification in the practical field.
2. The practical man who requires enough training in the theoretical field to grasp all the principles of why and wherefore.
3. The employer.

The first two elements of this organization must meet on a common field to acquire that which they do not have to convince the third element that he is securing just what he wants in the way of all-round engineering ability.

The fraternal engineering associations have been trying for years to solve this problem and have fallen short. It has required more concerted effort than they have been able to maintain.

The Institute of Operating Engineers is the best effort so far conceived to bring about ideal conditions by which operating engineering can be placed on a professional plane and be recognized as such by the employers and general public.

The editorials in POWER, March 20 and May 24, as well as the editorial in the June *Practical Engineer*, very clearly set forth the plan and give an idea of the scope of this latest phase of engineering development. One thing we all see is that fundamentally it is the greatest move ever made for the good of the operating engineer. Another thing we see is that the more we think it over the greater the whole plan looks.

Those who are taking up the development of this plan are finding the work worthy of their best efforts and a truly herculean task, the accomplishment of which will be an honor to all concerned.

To the prospective members the requirements will seem hard, but if the plan of the Institute is carried out as it will be there will begin to crop up in a few years one of the best bodies of organized operating engineers in the world. The employer will recognize members of the Institute as the best operating engineers to be secured and will have them and no other to serve him in this department.

The idea is creating untold interest already. Many schools and colleges stand ready to establish classes on these lines as soon as the work is laid out for them, and in school centers it is a foregone conclusion that this will be an established Institute throughout the country as soon as it can be given them.

The Institute must provide for the member in isolated localities also where he will need to work his way by correspondence and home-study methods in connection with his practical training. But the plan is practical in all cases. It may interest many to know that J. C. Jurgensen, the proposer of the Institute, graduated a class from Columbia University this spring from the first year's work, on the lines laid down in the first year's work of the Institute of Operating Engineers. This class comprised twenty-eight members, one of whom was a college graduate, two firemen and twenty-five engineers. Every member was working and attending this class. Every one stuck it out and passed a creditable examination, securing the coveted first year's certificate. This shows that this work is far from being visionary. The Institute of Operating Engineers is started and will succeed. It will be the solution of the operating engineers' problem.

Blast-furnace creosote oil is produced in vast quantities in Scotland, and the question of application has received a considerable amount of attention. To a certain extent it has been used for creosoting timber, liquid fuel, and for illuminating purposes. It furnishes from 20 to 35 per cent. of phenoloid substances, soluble in caustic soda, as against 4 to 7 per cent. in London coal-tar creosote, and about 16 per cent. for Midland or county make. One ton of creosote is capable of yielding 13,300 cubic feet of gas, with an illuminating power of 14 candles, or 29,300 cubic feet at 8½ candlepower. Information from *The Consular and Trade Reports* indicates that the annual production of creosote oil in the United Kingdom is approximately 50,000,000 gallons.

Hydraulic Rams and Their Capacity

By N. A. Carle

> *For certain kinds of service the hydraulic ram is a very efficient piece of apparatus, and within the past few years they have been built in comparatively large sizes. To obtain the best efficiencies it is necessary that the proper factors be taken into consideration and to facilitate such operations the accompanying set of charts has been prepared.*

A hydraulic ram is an automatic pump by which the fall or force of running water is utilized to pump a part of that water to a higher elevation.

The common idea of a hydraulic ram is that it is an extremely inefficient piece of apparatus suitable only for raising a small quantity of water for domestic purposes with a minimum amount of attention, practically no operating expense, and a low first cost of installation.

Hydraulic rams have been built for the past hundred years but it is only within recent years that they have been constructed in larger sizes and their efficiency materially improved. At the present time manufacturers are willing to contract for rams capable of delivering as high as 150,000 gallons of water per day of 24 hours, and with these units combined in batteries there is practically no limit to the size of the proposed installation, provided sufficient water is available.

These rams are guaranteed to operate on heads of from 2 to 50 feet at efficiencies ranging from 60 to 90 per cent., these efficiencies varying with the ratio of power head to discharge head. The usual power head for a ram is from 8 to 12 feet, because heads in excess of this are usually wanted for power purposes. Where all the power is required for pumping water, the modern hydraulic ram is more efficient for heads less than 50 feet than a waterwheel direct connected to a centrifugal pump or any arrangement of electrically operated pumps.

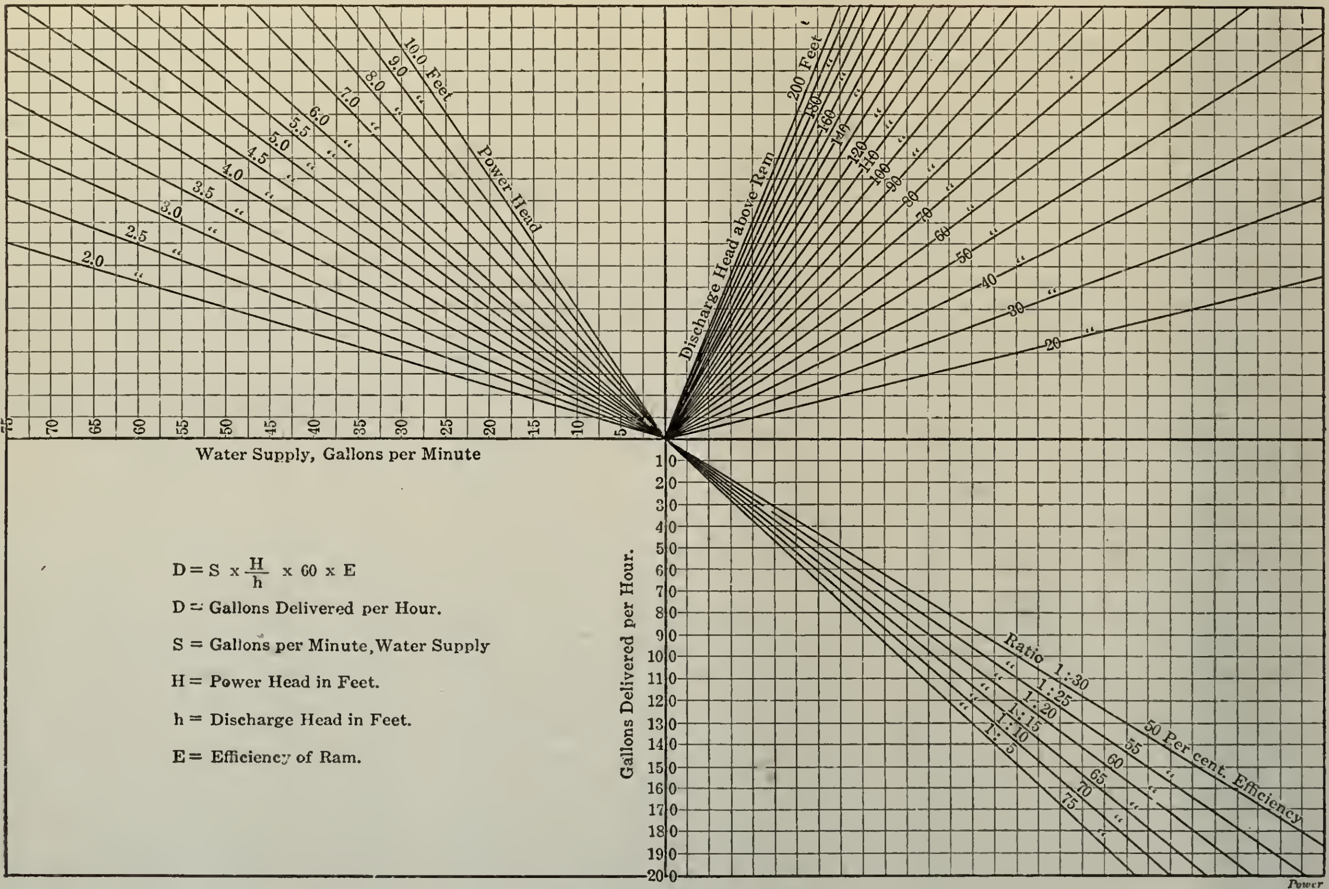


FIG. 1. CAPACITY OF HYDRAULIC RAMS UP TO 200 GALLONS PER HOUR

A ram will pump water to a hight of 45 feet for every foot of power head, but the reduction in the volume of the water pumped at this ratio is so great that it is not a practical operation. The ratio of 30 to 1 with a maximum lift of 500 feet is taken as the commercial limit of the ram.

There has been considerable discussion as to what factors govern the efficiency of a ram and the correct formula to be used in computing the efficiency. Rankine proposes the following:

$$E = \frac{q\,h}{(Q - q)\,H}$$

where,

E = Efficiency,

q = Quantity of water delivered per second,

Q = Quantity of power water used per second,

h = Difference in elevation between the water-supply level and the water level in the delivery tank,

H = Difference in elevation between the level of the water supply and that of the escape valve on the ram.

D'Aubisson proposes the following formula, the factors being represented by the same symbols as in the above:

$$E = \frac{q\,(H + h)}{Q\,H}$$

An analysis of these will show that Rankine's formula deals with the operation of elevating water from the supply reservoir to the delivery tank, while D'Aubisson's formula considers the ram as a machine receiving energy in the form of water under pressure and delivering some part of it at a higher pressure.

It is immaterial which formula is used to obtain the value of E, provided the formula for the capacity of the ram which uses this value of E is derived along the same lines.

The capacity of a ram is expressed by the following formula:

$$D = S \times \frac{H}{h} \times 60 \times E$$

where,

D = Quantity of water delivered in gallons per hour.

S = Quantity of water used in gallons per minute.

H = Difference in elevation between the water-supply level and the escape valve on the ram, commonly called the power head.

h = Difference in elevation between the escape valve on the ram and the water level in the delivery tank, commonly called the pumping head.

E = Efficiency of the ram calculated by D'Aubisson's formula, the values of which vary with the ratio of the power head to the pumping head as follows:

E = 75 per cent. when ratio = 1:5
70 per cent. when ratio = 1:10
65 per cent. when ratio = 1:15
60 per cent. when ratio = 1:20
55 per cent. when ratio = 1:25
50 per cent. when ratio = 1:30

The accompanying charts are intended to show graphically the quantity of water per hour and the hights to which it can be delivered by different amounts of power water used per minute under various heads. Chart No. 1 covers quantities of water supply up to 75 gallons per minute under heads of from 2 to 10 feet, delivering water up to 200 gallons per hour under heads of from 20 to 200 feet. Chart No. 2 can be used with water supplies up to 750 gallons per minute under heads of from 4 to 20 feet, delivering water in quantities up to 2500 gallons per hour under heads of from 40 to 400 feet. An example may serve to make this clear.

Example—With a power water supply of 30 gallons per minute and a 10-foot fall from the supply reservoir to the ram, how much water per hour can be raised to an elevation of 150 feet above the ram?

Solution—The ratio of power head to pumping head is 1:15; hence the efficiency of the ram will be 65 per cent. Using Chart No. 1, start with 30 gallons per minute, read up to 10-foot power head, then across to 150-foot discharge head, and down to 65 per cent. efficiency; following across to the left, the gallons of water delivered per hour will be found to equal 78.

The length of the supply pipe is very important as it governs the admission of air which is necessary for the efficient operation of the ram. Some manufacturers claim that the length of this pipe varies with the ratio of the power head to the pumping head. Successful installations show that the best results are obtained for small rams when the length of the supply pipe is four times the vertical fall, corresponding to an angle of 14.5 degrees with the horizontal, and five times the vertical fall, corresponding to an angle of 11.5 degrees with the horizontal, for larger rams.

The supply pipe should be of the same diameter throughout its length and without bends. The delivery pipe may be of any size suitable for carrying the water.

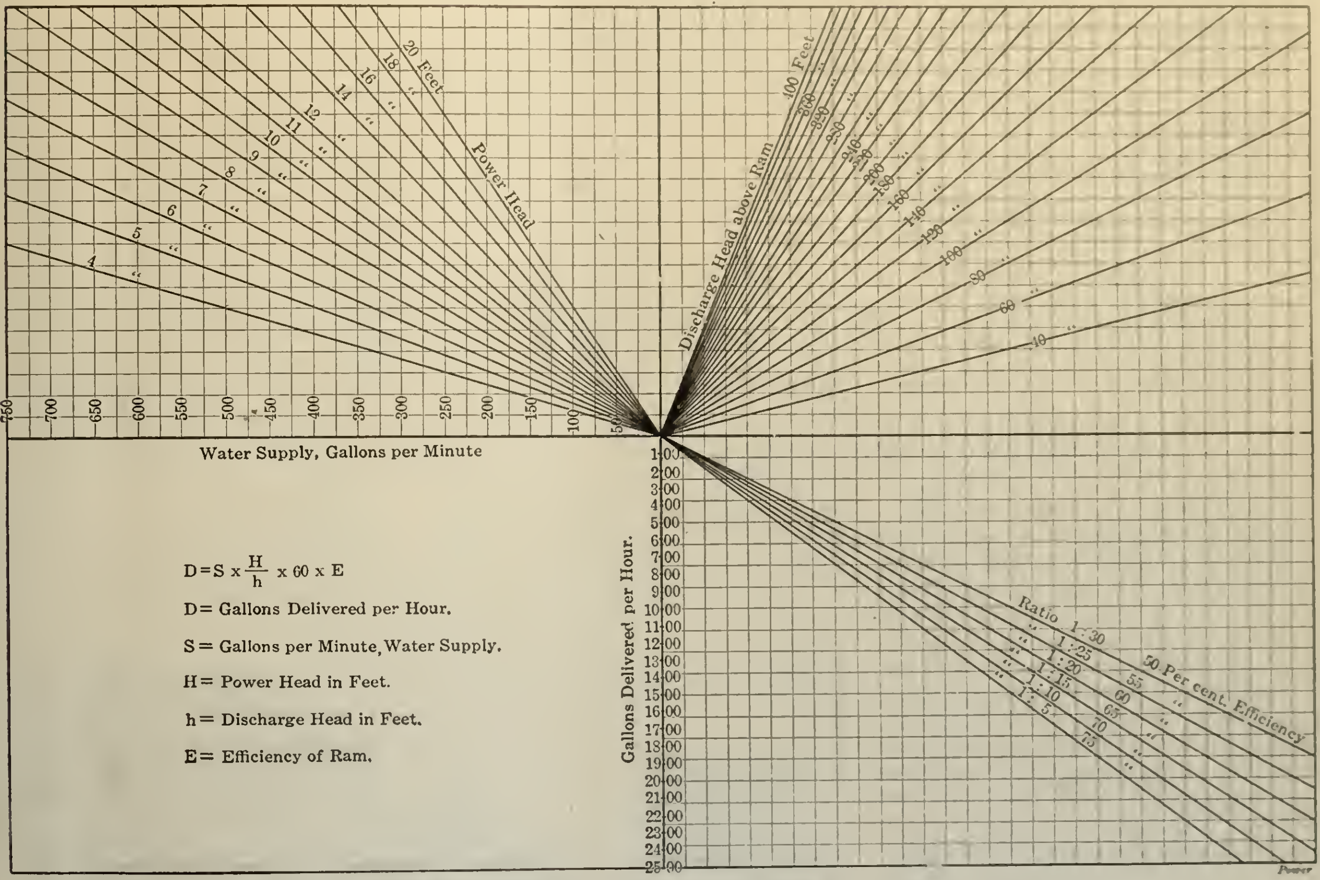


Fig. 2. Capacity of Hydraulic Rams up to 2500 Gallons Per Hour

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

The Electric Elevator

By William Baxter, Jr.

Elektron System with Mechanical Control

All the electric elevators described in previous articles belong to the type known as single-speed machines, which, after being brought up to speed run at substantially that speed constantly. Differences in load will make a slight difference in velocity but it is so slight as to be hardly noticeable. Electric elevators are also made to run at different speeds, and when so made are designated as "two-speed" or "three-speed," or as "variable speed" if they are arranged so as to run at any of several speeds, say, eight or ten. At the present time most of the elevators that are built for car speeds of 200 feet per minute or more are of the variable-speed type. The variations in speed are obtained in two ways, and both arrangements are very simple so far as the construction is concerned, but the principles involved in one of these methods are rather complicated, and I will not undertake to explain this method until machines in which it is used have been described.

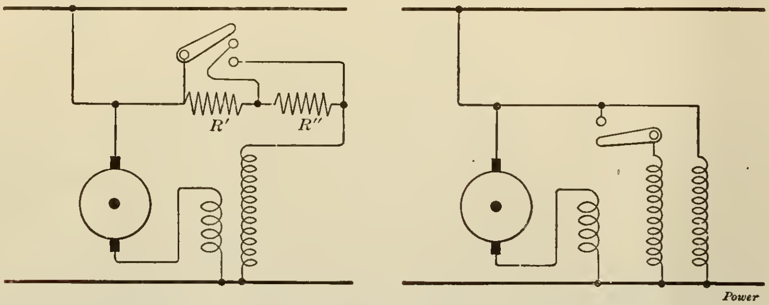


Fig. 112. Fig. 113.

Methods for Varying Speed by Varying Field Strength

The other type operates on a principle that can be easily explained, and this one is considered here. The principle consists in varying the motor speed by changing the strength of its field excitation, and this change is accomplished in either of two ways, which are illustrated in the simple diagrams, Figs. 112 and 113. In Fig. 112, it will be noticed that there are two resistors R' and R'' connected in series with the shunt field winding. If these are short-circuited by means of the switch, the current that will flow through the shunt field coils will be stronger, of course, than if the resistors are in the circuit, and the strength of the field magnet will be greater than if the current is weakened by leaving the resistors in circuit; as a result, the speed of the motor will be low. Therefore, when the resistors are both short-circuited, the motor will run at its lowest speed. If the short-circuit around the resistor R'' is opened, so that the field current has to pass through this resistance, its strength will be reduced and, consequently, the field strength will be reduced and the motor speed will be increased. Opening the switch, entirely, as represented, so that both resistors are cut into the circuit, will still further increase the motor speed. This arrangement gives a three-speed system, and if only one resistor were used, the motor could be run at either of two speeds; with nine resistors and a controller made so as to cut these into the circuit one at a time, ten different speeds can be obtained, and then we will have what is called a "variable-speed" machine.

The speed variation that can be obtained in this way is not very great in practice, although it might seem that it could be made anything desired. To increase the speed range, the size of the motor must be increased, and while a range of two or three to one can be obtained without greatly increasing the size of the motor, a range of six to one would require a motor so large as to increase the cost of the elevator machine considerably.

Fig. 113 shows a way of varying the field strength that is just as effective as that of Fig. 112, but it is not so generally used. In this arrangement, there are two shunt field windings, one of which is the regular winding and the other an extra winding to be cut in or out for changing the speed. When the regular winding is in circuit alone, the motor speed will be greater than with both field windings connected, because the second winding strengthens the field and reduces the speed; if a third winding is added and current is passed through it, the motor speed will be still further reduced. One advantage of this arrangement is that as there are two or more field windings, there is very little danger of the field excitation suddenly dropping down to

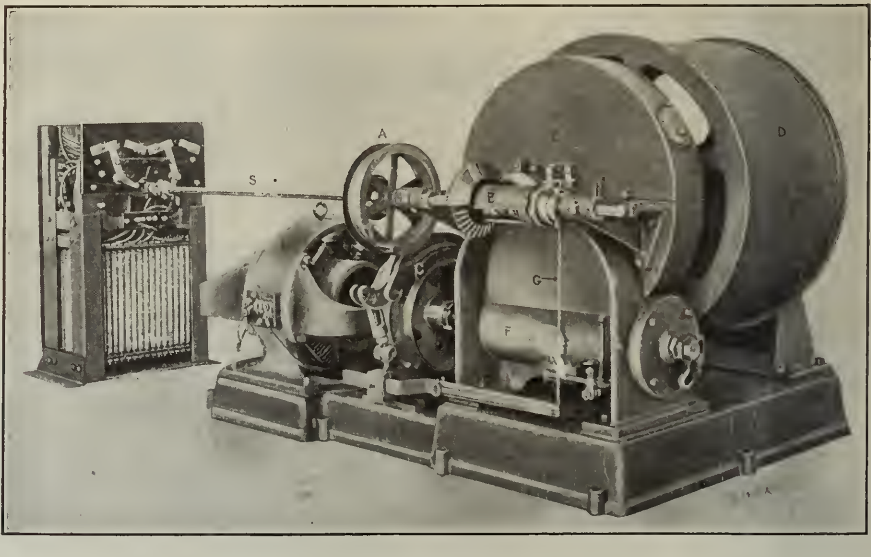


Fig. 114. Elektron Elevator with Mechanical Control and Automatic Acceleration

nothing, as it would if there were only one winding and the circuit of that one should be accidentally opened. With one field winding permanently connected, as in single-speed machines, there is less danger of its becoming open-circuited than when resistances are connected in circuit with it, as in Fig. 112. For this reason multiple-speed motors are sometimes made with two field windings connected as in Fig. 113, and resistors connected in circuit with one of these windings. This provides one field winding the circuit connections of which are never disturbed, and there is little danger of losing excitation.

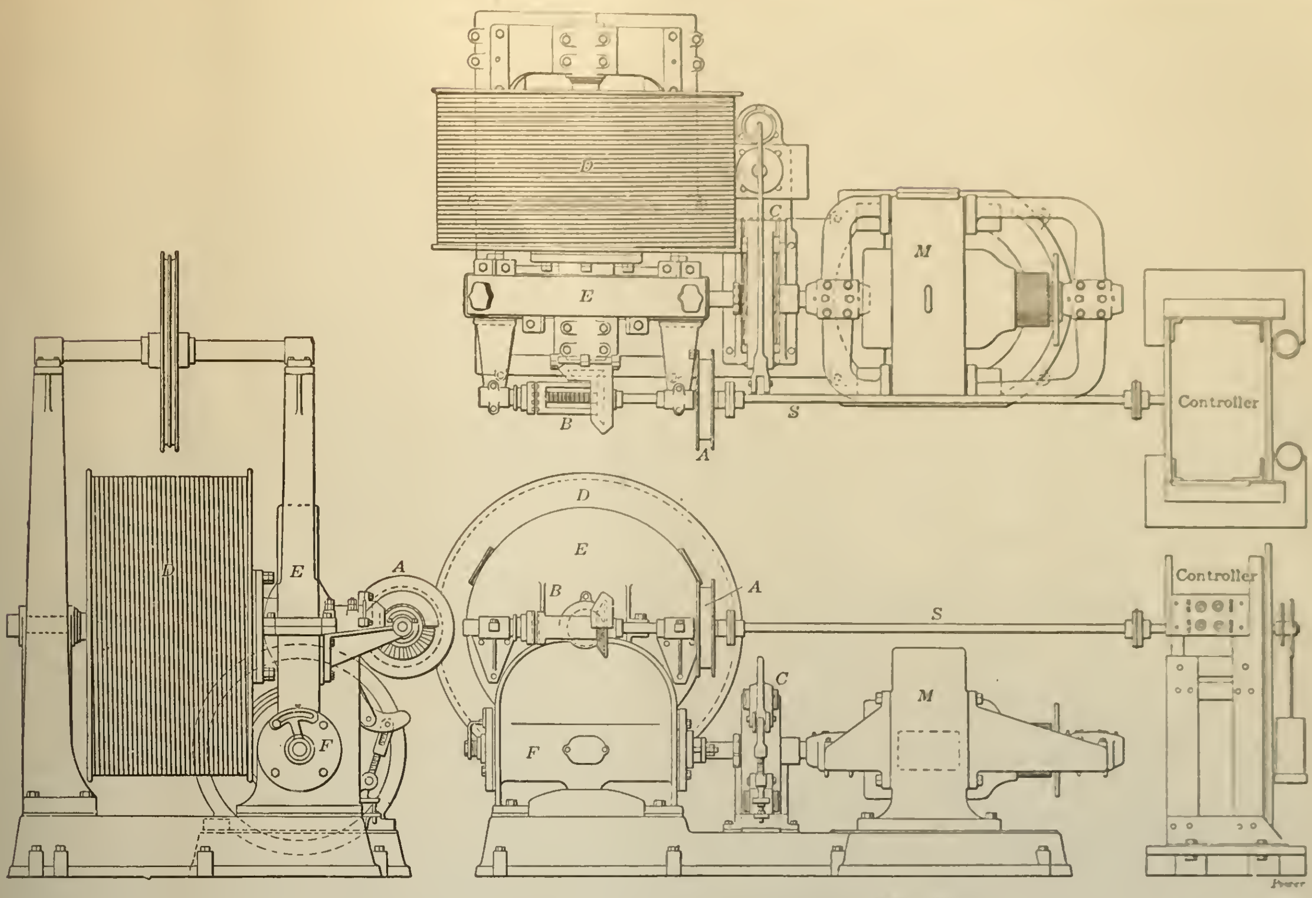


FIG. 115. PLAN, SIDE AND END VIEWS OF ELEKTRON MACHINE

The Elektron mechanical-control elevator shown in Fig. 114 is of the variable-speed type, and its field winding is connected in the manner shown in Fig. 112, with the exception that the speed-controlling resistor, instead of being divided into two sections R' and R'', is divided into a large number of sections, putting the machine in the variable-speed class. Before taking up the controller construction it will be better to describe some of the other details of the machine in which it differs from other machines previously described.

As may be noticed in Fig. 114, the operating sheave *A* is mounted upon a shaft *S* that carries the stop-motion mechanism at one end, while the other end extends to the controller and operates the several switches of the latter directly. The rod *G* is connected with the slack-cable stop so as to throw the brake on if the lifting cables should slack up, and at the same time move a clutch that causes the stop motion to rotate and turn the shaft *S*, thereby moving the controller switches back to the "off" position. The brake *C* is of the magnet-operated type. The general location of the several parts of the machine can be more fully seen in the drawing, Fig. 115, which gives an end elevation looking from the drum end of the machine, a side elevation corresponding to the view in Fig. 114, and a plan view. At the end of the worm casing *F* there is an arm with a long circular slot in its end. This is an adjusting device to take up any slack that there may be in the thrust bearings at the ends of the worm. This view also shows the frames extending upward from the machine to hold the shaft upon which the sliding sheave travels as the ropes wind and unwind on the drum. This is not shown in Fig. 114.

The construction of the brake of this machine is shown in Fig. 116. The magnet *M* that releases the brake is made so that when energized it lifts the plunger upward, carrying with it the lever *D* and thereby raising the shoes *B*, B' from the brake wheel *A*. The connecting rod D' is made so the distance between the brake wheel and the shoes can be adjusted and the screw D'' is provided to hold the shoes concentric with the wheel. The shoes have leather facings A'. At *F* is a dashpot to retard the motion of the shoes when the brake is applied and prevent too sudden a checking of the elevator. When the magnet *M* is deenergized, the weight of the lever *D* and the other parts hanging on it causes the shoes to close up and grip the brake wheel *A*.

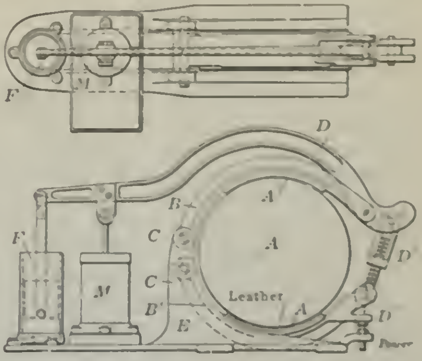


FIG. 116. BRAKE MECHANISM

The controller shown in Fig. 114 carries a reversing switch on the side shown in the illustration, a starting switch within the controller box, and the speed-controlling switches on the back of the controller box. The starting switch is actuated directly by the shaft *S*, but the other switches are moved by cams mounted on the shaft. The reversing switch is so made that it is at all times in one or the other of the running positions, as may be seen from Fig. 114; hence if the elevator is stopped and then started up again in the same direction as before, the reversing switch is not moved. It is only when the direction of rotation of the motor is reversed that the reversing switch is moved.

The construction and operation of the reversing switch can be better understood from Fig. 117. Upon the shaft *S* is mounted a casting *H* that carries two cams *D* and *D'*. These cams are made of fiber, to insulate the reversing-switch levers *A* and *A'* from the shaft *S*. The cams *D, D'* are not in line, the cam *D'* being nearer the observer than the other. The drawing renders the action of these cams somewhat confusing from the fact that it looks as if the cam *D* were holding the lever *F'* in the position shown by pressing against the roller on the end of the lever. As a matter of fact, however, the roller on the end of *F'* is not in line with the cam *D*, but is in line with the cam *D'*. The lever *F'* was moved into the position shown by the cam *D'*. If the shaft *S* were rotated counter-clockwise, the edge *d'* of the cam *D'* would strike the roller on the end of *F'* and move the latter into the position shown, throwing the switch blades *A* and *A'* into the contact clips *C''* and *F* respectively. If now the shaft were rotated clockwise to the position shown, the friction of the clips *C''* and *F* will hold the levers in the position in which they are drawn. Such a movement, however, would open the starting switch and stop the elevator. If the operator desired to start up running in the same direction, he would again turn the shaft *S* counter-clockwise and close the starting switch without disturbing the reversing switch, but if he desired to start the elevator in the opposite direction, he would turn the shaft *S* clockwise until the edge *d* of the cam *D* struck the roller on the end of the lever *C'* and thereby shifted the reversing switch blades *A A'* over to the opposite side into the other clips *F* and *C''*. The two contact clips *C''* are connected to each other electrically, and the clips *F* are also connected. The clips *F* are connected to one of the armature terminals and the clips *C''* are connected to the other armature terminal, through the starting resistance box. The blades *A* and *A'* are connected to the leads from the supply circuit. Consequently, throwing the blades *A A'* from one side to the other reverses the connections between the motor armature and the line.

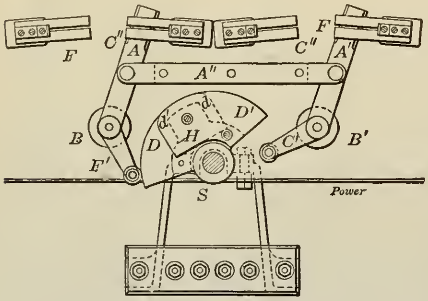


FIG. 117. REVERSING SWITCH

Smoothing Commutators

By James Roche

I have often read in POWER of persons grinding down commutators with sandpaper or homemade truing devices, or turning them in a lathe. The latter way is perhaps the only accurate one and is therefore the best, but it is often impracticable on account of the time taken and labor oftentimes needed.

I have found it very convenient and satisfactory to use a piece of grindstone for the purpose. One face of the block is concaved to fit, approximately, the curvature of the commutator, and this curved face is simply pressed against the commutator while the armature revolves. The stone will cut quickly and soon smooth the brush tread of the commutator. Of course, if the commutator is out of true, there is no way to true it except in a lathe or with a truing tool.

A block of grindstone 5x9 inches and 4½ inches thick is in regular use in our station for smoothing the commutators of two railway machines, which are 26 inches in diameter and 8 inches face, a motor-generator commutator 16 inches in diameter and 6 inches face, and for the collector rings on two Bullock and two Fort Wayne alternators. A smaller stone is used for three small commutators.

A spark is hardly ever seen on any of the six commutators, except in case of a sudden overload on the railway machines. All the commutators look like new and as if they had never been touched, and most of them have been in service about seven years in a plant operated twenty-four hours a day. We keep some brushes soaking in oil all the time and keep one of these in each set of brushes, staggering them around the commutator so as to engage the whole face, which keeps it well lubricated.

LETTERS

Care of Rotary Converter Rings

Mr. Eberhardt's question regarding the care of rotary-converter collector rings in the May 17 issue reminded me that there was once a time when my daily bread was earned by making myself generally useful around a substation with six 1500-kilowatt rotary converters. On ordinary loads we found that vaseline applied with a felt swab about once an hour to the commutator and once in two hours to the collector rings gave the best results. When the machines were loaded above the rated capacity it was necessary to grease them a little oftener.

Each week the commutator brushes were carefully inspected and any that were pitted or worn due to sparking were brought into shape again by placing a piece of sandpaper between the commutator and brush and drawing it back and forth, following the curve of the commutator, till the brush was smooth.

At the same time the collector brushes were taken off and thoroughly cleaned and then set back again so as to take care of any unequal wear.

W. L. DURAND.

Washington, D. C.

Motors in Dusty Places

Referring to that part of H. T. Dean's article on "An Overloaded Century Motor" which relates to trouble from wood dust collecting in the sleeve of a pattern-shop motor, I fail to see the necessity of taking this machine down every two or three months to clean out the dust. A far more effective method is to give the motor a good blowing out, preferably with compressed air, at least once a week. Where air is not available, a good hand bellows, vigorously operated, will do fairly well. I should feel exceedingly reluctant to let a commutator type of motor, whether direct-current or single-phase alternating-current, run for two or three months under an accumulation of dust in a wood-working mill, owing to the inflammable nature of this fine dust and the liability of a spark from the commutator setting it off. I have always insisted that motors be cleaned not less than once a week, generally at the end of the calendar week.

A better practice still, where commutator-type motors are purchased specially for wood-mill service, is to buy inclosed motors. Where open-type motors only are available, the writer has got around the difficulty by making two galvanized-iron caps or shields fit the ends of the motor frame, each shield having an oval or oblong hole cut in it for ventilation; the hole was covered with a brass screen or bolt cloth of rather fine mesh so as to exclude the dust. The accompanying sketch illustrates this arrangement. The shield on the pulley end has to be made in two parts, unless the pulley can be taken off without too much trouble and the cap slipped on over the shaft.

On some types of motors, some difficulty is experienced in fastening this dust shield securely, owing to construction of the frame. On most Wagner single-phase machines, however, the shields can be held on by lugs under the yoke-bolt nuts.

I have used this method very effectively on wood-mill motors, as well as on direct-current machines operating machine tools where there was a liability of metal chips or borings and oil getting into the motor and giving trouble.

Taking apart and reassembling the throw-out mechanism on the Wagner and Century motors is usually such a tedious and unpleasant job that any method which will obviate the necessity of doing so should be welcome.

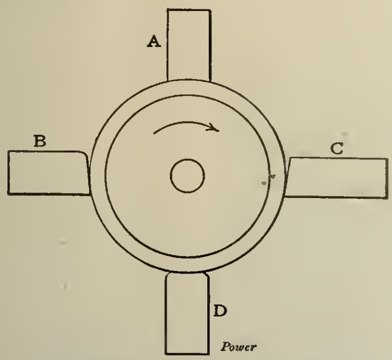
A. P. H. SAUL.

Buffalo, N. Y.

Mr. Lindsey's Burned Brushes

Here is another guess as to the cause of the brushes burning as stated by Mr. Lindsey in the May 31 issue of POWER.

The trouble may be due to any one or a combination of the following conditions: Brush holders unequal distances apart or not parallel with the commutator. That is, the holders at the armature end of commutator may be, say, 14 inches apart while the opposite ends are 13⅞ inches apart.



ONE CORRECT AND THREE INCORRECT BRUSH FACES

The use of too much oil or compound on the commutator.

The brushes may not fit the commutator properly. When fitting brushes to a commutator the sandpaper should be pulled in the direction of rotation, not back and forth. Then, too, if the brushes are quite loose in the holders they should be held firm and as near as possible in the position they occupy when the machine is in operation. The brushes should fit the commutator as at *A*, Fig. 1; the other three are wrong. However, I have seen brushes in one or all three of the conditions represented by *B*, *C*, *D*, and if the machine is carrying anywhere near its rated load, there is trouble about like Mr. Lindsey's.

A. K. VRADENBURGH.

Albany, N. Y.

Mr. Lindsey's brushes may be sparking a little because of vibration. If his machine is belted to the engine, the vibration may be caused by a thick spot in the belt, and if direct connected to the engine, lost motion in the crank-pin boxes may cause the vibration.

HERBERT B. BRAND.

Brooklyn, N. Y.

The excessive sparking mentioned by S. G. Lindsey is undoubtedly due to a lack of balance or adjustment somewhere. Machines in normal condition, especially generators of that size, do not act that way unless there is a "screw loose." If I had charge of that machine, my first move would be to see whether the brush holders were spaced equidistantly on the commutator. What makes me suspicious in this respect is the statement that 28 brushes were burned on a six-hour run, indicating that at least some of the brushes are not in their proper relation to the neutral point. The best way to find out whether the brushes bear on the commutator at equidistant points is to raise all brushes and fit a piece of tough manila paper, the width of the commutator, around it, the length of the paper being such that the ends will just meet. Then take a pair of compasses and lay off on the paper, laid out flat, a number of points equal to the number of brush holders. Draw lines squarely across the paper at these points, replace on the commutator and fasten down smoothly with string or a bit of paste under the ends.

Let down the brushes and note if all the brushes "toe the mark." If not, you have in all probability found the trouble, and the brush holders should be adjusted until all brushes line up true to the marks. If, however, the spacing is found all right, and, assuming that shifting the rocker ring to find the neutral point has already been tried, the next move will be to investigate the polarity of the interpoles. As stated in the editorial note, the polarity of each interpole must be the same as that of the main pole next ahead in the direction of rotation. These can be tested out very readily with a pocket compass, which can be bought for 10 cents. However, this latter trouble is rather a remote possibility, but should it exist the polarity of the interpole may be corrected by merely transposing the connections of the coil on it.

Another probable cause for sparking is improperly fitted brushes; they may not touch the commutator across their entire width or face. I have known cases where brushes apparently bore evenly, but upon close examination it was found that only the toe of the brush was in actual contact; consequently the contact surface of the brush was reduced to such a degree that when a heavy load came on, the brush could not carry the current without undue heating, causing excessive sparking and burning of the brush.

I am opposed to commutator lubrication on general principles, and would not permit its use on commutators under my care except in emergencies. There are some rare cases, however, where it may be beneficial. In such cases I would recommend vaseline, but it must be used very sparingly, applied with a piece of canvas or cotton cloth slightly moistened with the vaseline. This treatment sometimes helps to form the chocolate-brown glaze so desirable on a well kept commutator. After this glaze is obtained, no more "dope" of any kind should be used. If graphitized carbon brushes are used, the commutator will take care of itself. That has been my experience.

A. P. H. SAUL.

Buffalo, N. Y.

Identifying the Conductors in a Cable

Mr. Ryan's article on locating faults in circuits, in the issue of May 24, suggested to me that perhaps a method for identifying conductors in a cable already laid, without assistance and with the least possible amount of walking, might be of service to some of the readers of POWER.

Take for example, a 10-conductor cable one-half a mile long. To identify all of the conductors by means of the galvanometer and battery by grounding one end of the conductor would call for a lot of walking back and forth. An easier method would be as follows: At the station end, connect the conductors in groups, each group containing one more wire than the preceding group. Now go to the distant end and with the galvanometer and battery identify the wires in the various groups.

Make a diagram similar to the one here shown. Number the conductors from one to ten and proceed somewhat as follows: Connect No. 1 to No. 9 in group No. 4; No. 2 to No. 4 in group 3; No. 3 to No. 8 in group 4, etc., taking care not to connect more than one conductor of any one group to conductors of any other one group. Indicate on the diagram the different connections and return to the station end where by means of the sketch and the galvanometer, after disconnecting the ends of the various groups, the connections as made at the distant end can be traced out and the station end numbered to correspond with the far end. For example, a wire in group 2 is connected to a wire in group 3. These can be identified and by reference to the sketch the wire in group 2 is seen to be No. 2 and the wire in group 3 is No. 4, and so on.

This method can be used on a cable with any number of conductors and dispenses with the services of an assistant and the necessary means of communication between ends. I would suggest the use of a portable voltmeter, if one is available, instead of the galvanometer.

A. C. KERR.

Ft. Monroe, Va.

Gas Power Department

A New Vertical Engine

The accompanying engravings illustrate the principal constructional features of the Turner-Fricke gas engine, which is relatively a new comer in the American field. There are no radical departures from modern practice in the design, the aim having been evidently toward improving details of construction rather than the exploitation of any personal hobbies.

As might be inferred from close inspection of Fig. 1, the main framework of the engine consists of a box base and a crank case bolted together, with the cylinders individually bolted to the crank case. The base carries four pillow-blocks on transverse yokes, into which the crankshaft bearings are set, as represented in Fig. 2. The base is ribbed inside and the pillow-blocks are double-ribbed, obviously for the purpose of securing strength without excessive weight. The lower shell of each journal box rests on a massive wedge which is adjusted by means of bolts through opposite sides of the pillow-block jaw (see Fig. 4) to bring the boxes into alinement vertically; they are kept in lateral alinement by the jaws in which they are mounted. These adjusting bolts are reached through hand holes in the crank case, of which there are three, opposite the cranks, in each side of the case, as shown in Fig. 3. The pillow-block caps are fitted to the upper ends of the jaws in absolute line with each other, so that it is necessary only to put the shaft into contact with the caps in order to have it accurately lined up; the caps are lined with bearing metal, like the shells, to permit this contact with the shaft.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

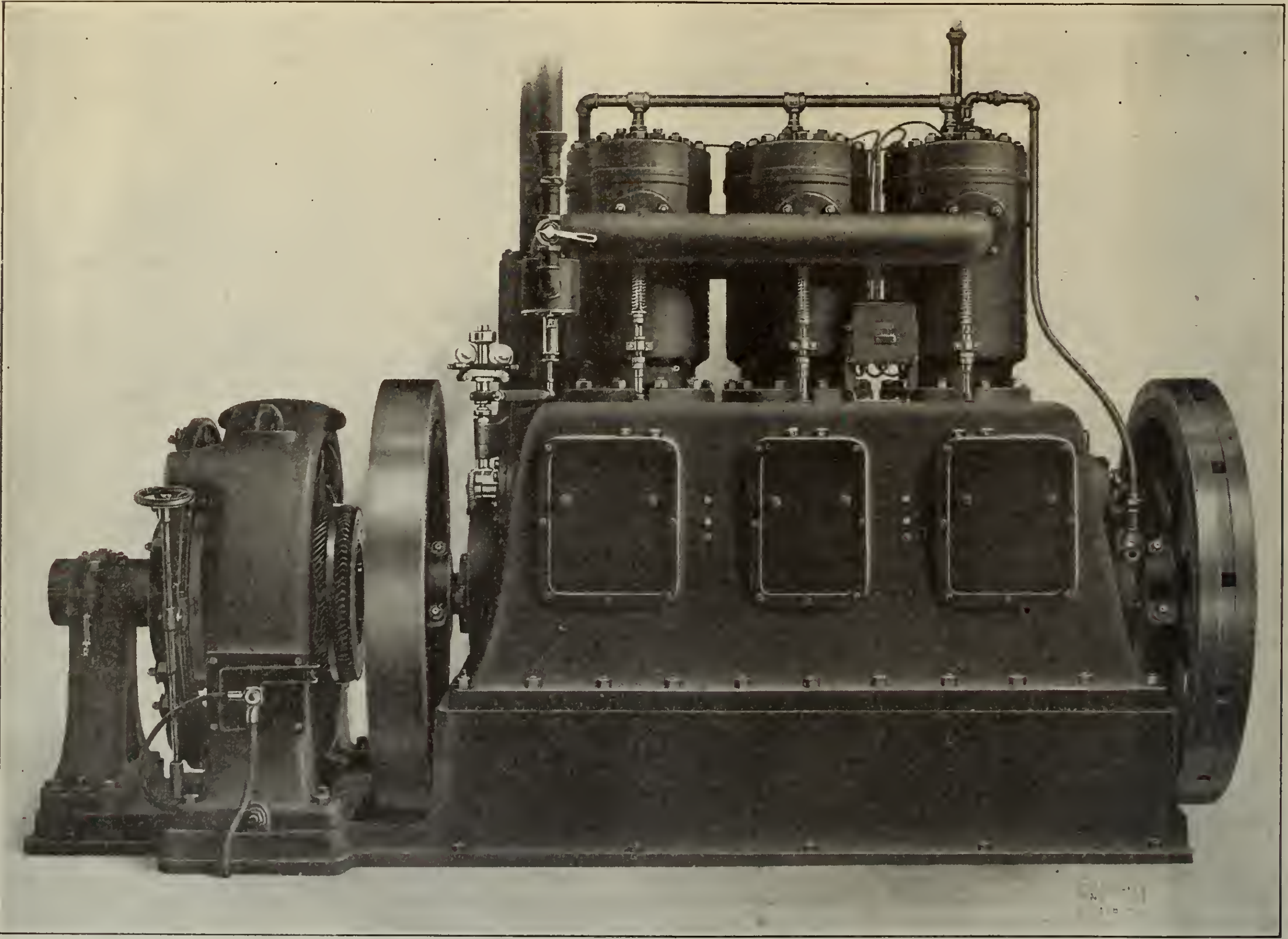


Fig. 1. Turner-Fricke Gas Engine Direct Connected to a Direct-current Generator

The engine operates on the four-stroke cycle, and is therefore equipped with the usual half-speed cam shaft for valve operation—or, rather, with two such shafts, the inlet valves being on one side and the exhaust valves on the other side of the cylinders. The cam shafts are driven from the crank shaft through two-to-one spur gears, which are visible in Fig. 3, and each cam shaft is removable from the crank case, without disturbing anything else, upon unbolting the intermediate journal boxes and taking off one of the end boxes; the latter are cast with flanges covering openings in the crank case large enough to pass the cams.

The valve push rods terminate in the crank case at rocker arms, each carrying a roller in the path of the corresponding cam. Fig. 4 shows this construction and the arrangement of the valves and other internal parts corresponding to one cylinder. The valves work on removable seats but the complete removable valve cages frequently employed by engine builders are not used, for two reasons: it is easier to cool the cage effectively when it is integral with the main jacket casting than when it is a separate structure set into a pocket, and by providing removable valve-stem sleeves, seats and plugged openings above the seats, all of the operating advantages of removable cages are obtained. Each valve cage is completely surrounded by the jacket water up to the pipe connection. The upper end of each push rod is equipped with means for adjusting the buffer with relation to the end of the valve stem.

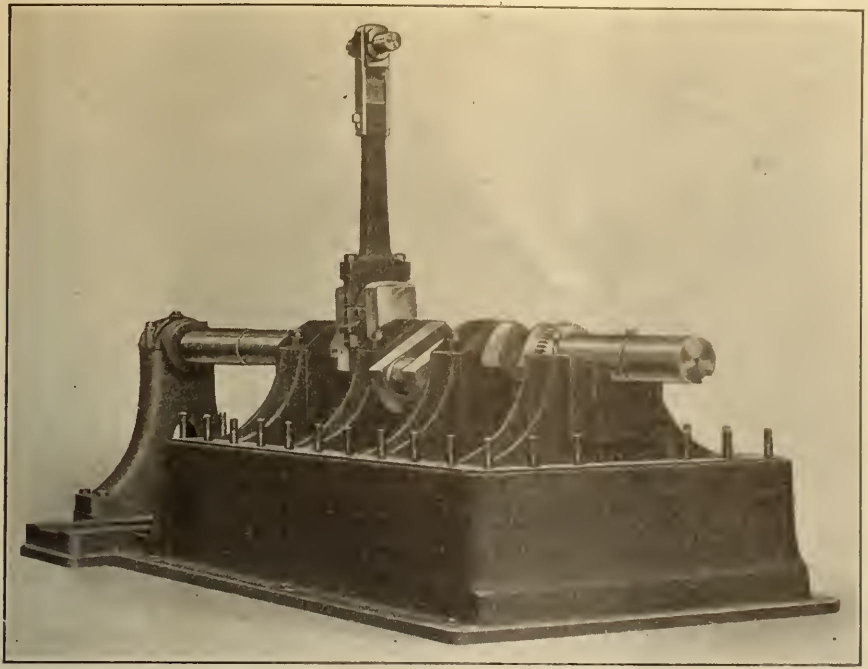


Fig. 2. Base and Crank Shaft of Turner-Fricke Engine

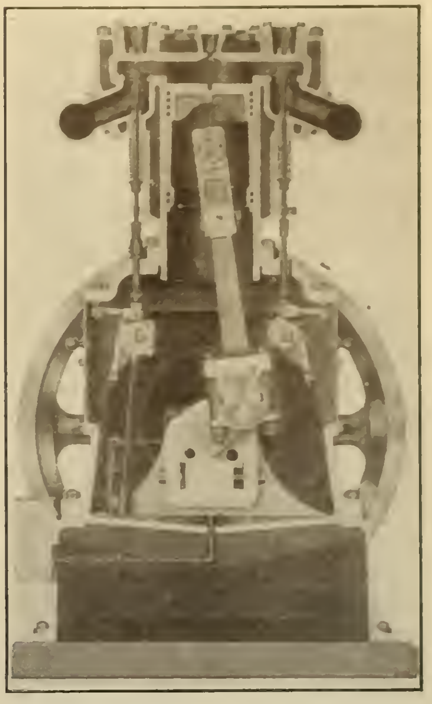


Fig. 4. Vertical Section through Middle Cylinder

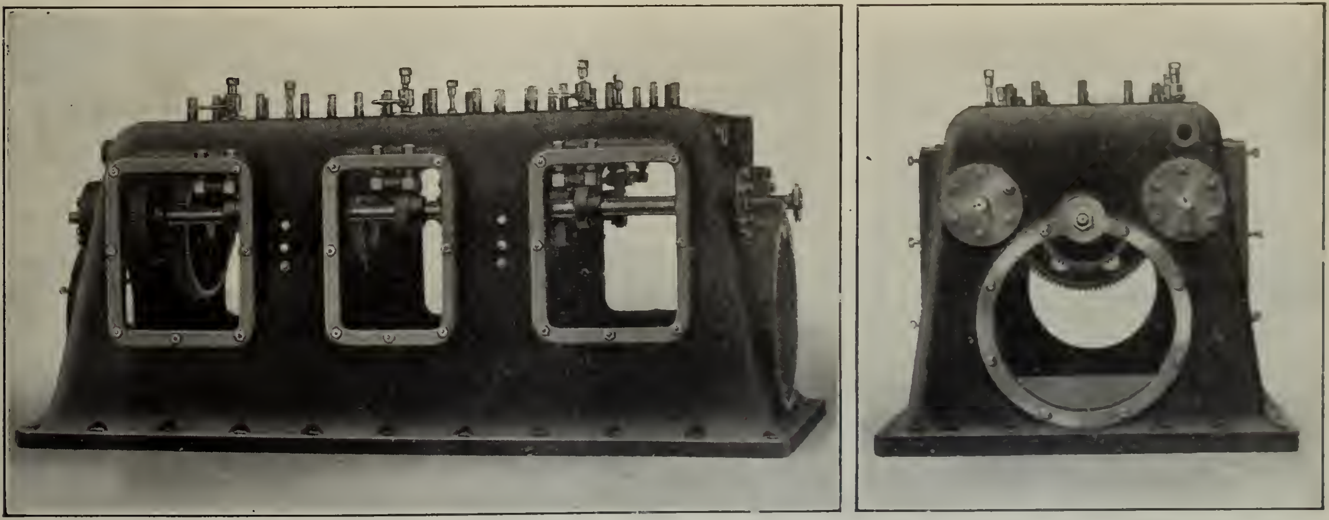


Fig. 3. Crank Case, with Both Cam Shafts in Place

The governor is of the simple flyball type, driven through bevel gears from the end of the inlet cam shaft, as shown in Figs. 1 and 5; it controls the port openings of the mixing valve, which is of the multiported piston or "circular gridiron" type. The proportion of air to gas is adjusted by means of the hand lever, and when once set it remains unchanged by the mixing valve. The action of the valve, under the influence of the governor, is to throttle the mixture according to the load requirements.

The arrangement for adjusting the mixture ratio is unusual and ingenious. The mixing valve *V*, Fig. 6, which is a drum with closed ends, is mounted rigidly on its stem but the stem is separate from the push rod *R* by means of which the governor lifts the valve. Therefore the valve is free to rotate so far as the governor connections are concerned. The bottom head, *H*, of the valve chamber is held on by a circular flange *F* which allows the head to be rotated on its seat; a lever *L*, pivotally attached to the cylindrical spring housing *B*, carries a pin *C* which extends through the chest head *H* into the valve drum; by swinging the handle *L* one way or the other, the valve drum is twisted around so as to increase or decrease the area of the gas inlet openings *G*. The air ports *A* in the valve drum are longer than the corresponding ports in the chest wall, and the valve can therefore be twisted far enough to shut the gas ports entirely without affecting the area of the air ports. The thumb nut on the end of the pin *C* clamps the chest head to the flange *F* in whatever position the valve may be set.

The ignition system is of the high-tension or "jump-spark" class, similar to that commonly used on automobile engines. The spark plug of each cylinder is set in the center of the cylinder head, as shown in Fig. 4, but there is no pocket around the plug on the inside of the cylinder.

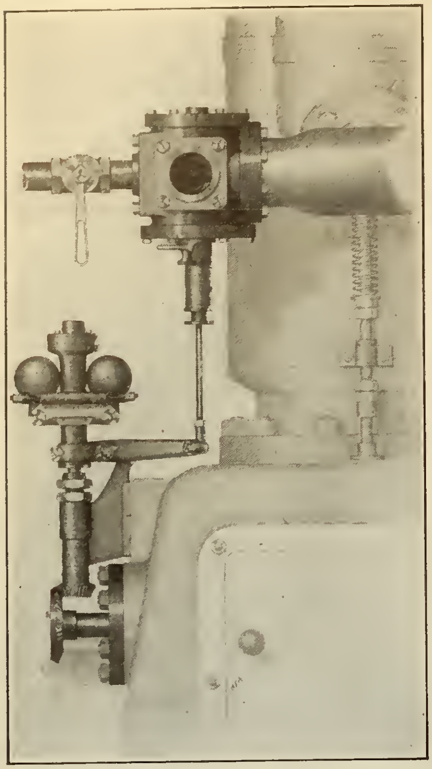


FIG. 5. MIXING VALVE AND GOVERNOR

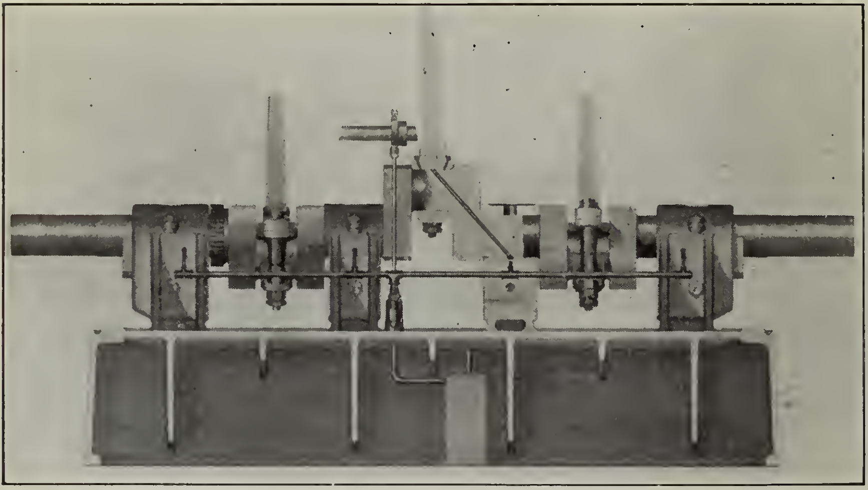


FIG. 7. LUBRICATING SYSTEM FOR MAIN BEARINGS AND CRANK PINS

The pistons are lubricated by the splashing of oil when the cranks dip into oil in the base, in the customary manner, but this is not relied upon to lubricate the crank pins and main bearings. A small oil pump driven by an eccentric on one of the cam shafts (Fig. 4) delivers oil by force feed to the main bearings and crank pins, which are channeled, as indicated in Fig. 7, so that the oil can pass freely between each of the pins and the two adjacent bearings. The delivery-pipe system is also depicted clearly in this illustration. The pump is shown just above the surface of the base, to the left of the center; it is more clearly represented in Fig. 4.

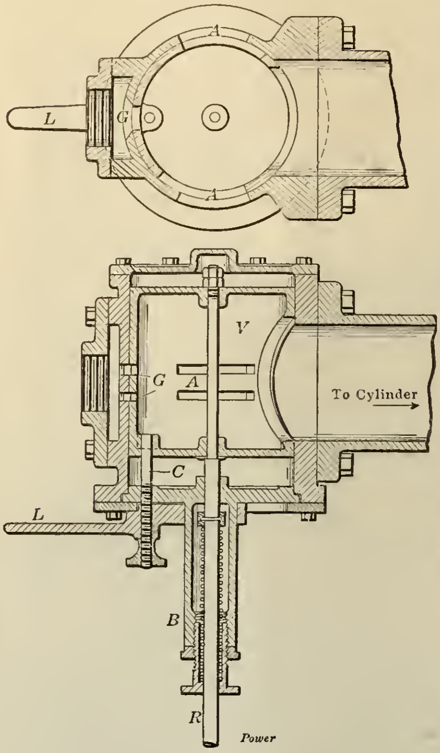


FIG. 6. MIXING VALVE SECTIONS

The engine is started by compressed air through a valve which is put into commission by the admission of air from a storage tank to the valve. The valve is operated by a cam on the end of the inlet cam shaft, as represented in Fig. 8. When the air supply is turned off, the valve drops out of action.

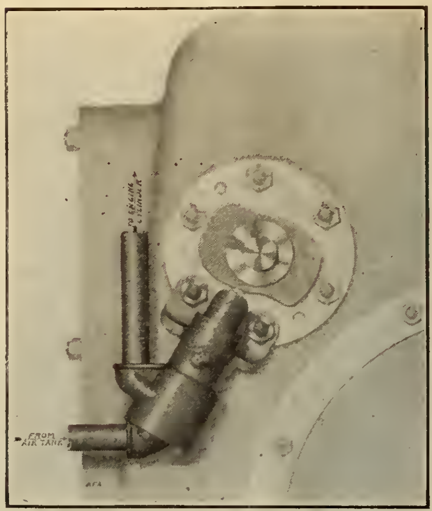


FIG. 8. STARTING CAM AND VALVE

Another Case of Mere Common Sense

The articles by Frank E. Booth and W. L. James telling about troubles with internal-combustion engines which proved too deep for the local engineers remind me of a case in which my firm sent a high-priced man 200 miles to do a ten second job.

A 10-horsepower horizontal oil engine was provided with a fuel tank in the base plate from which the oil was pumped up to the lamp and vaporizer measure. The engine gave the local force much concern by resolutely refusing to run for more than two or three minutes at a time. They could not get enough oil at the measure so they carefully cleaned the pump-ball valves, blew through the pipes, repacked the plunger, tightened a nut here and slacked one there, to no purpose. He rang up the works and our man went down. The first thing he did, after starting the engine up and having it stop after the regulation three minutes, was to apply his finger nail under fine-wire gauze filter at the bottom of the pump intake and scrap away the waste and filth that had got there by the carelessness of the operator in not pouring the fuel into the tank through the gauze strainer provided for that purpose.

Mr. James speaks truly when he says that this is the sort of talent which is getting gas and oil motors a bad name for "cussedness."

JOHN S. LEESE.
Manchester, England.

Readers with Something to Say

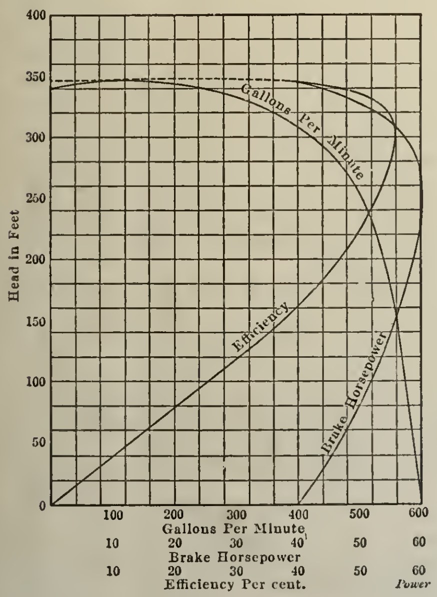
Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Centrifugal Pump Characteristics

The accompanying curves were plotted from data obtained during the test of a 4-inch three-stage centrifugal pump running at a constant speed of 1000 revolutions per minute, and show some interesting characteristics.

The quantity discharged is a maximum at zero head. As the head pumped against increases, the quantity discharged decreases, at first slowly, but at an increasing rate as the head increases. Near the maximum head the quantity pumped decreases very rapidly and reaches zero quantity at the maximum head. Above this maximum head no water can be pumped.

The brake horsepower is a minimum at a head above the maximum head against which the pump will discharge. As the head decreases the pump starts pumping and the brake horsepower rises at once to a value approximately two-thirds of the maximum value. As the head decreases the brake horsepower increases rapidly to its maximum value at approximately three-fourths of the maximum head. Reducing the head further gradually decreases the brake horsepower until at zero head it is again approximately two-thirds.



CURVES SHOWING CHARACTERISTICS

The efficiency which at zero head is zero, rises gradually to its maximum value at about seven-eighths of the maximum head. Above this the efficiency drops off rapidly to the maximum head when, of course, it falls to zero.

These qualities of the centrifugal pump are extremely valuable, especially when electric motor drive is used. They are of the utmost importance when the pump is used for fire purposes, for the breaking of a pipe or hose line does not render the motor liable to burn out or shut down at a critical moment.

WILLIAM SANFORD.

New York City.

A Remedy for Priming

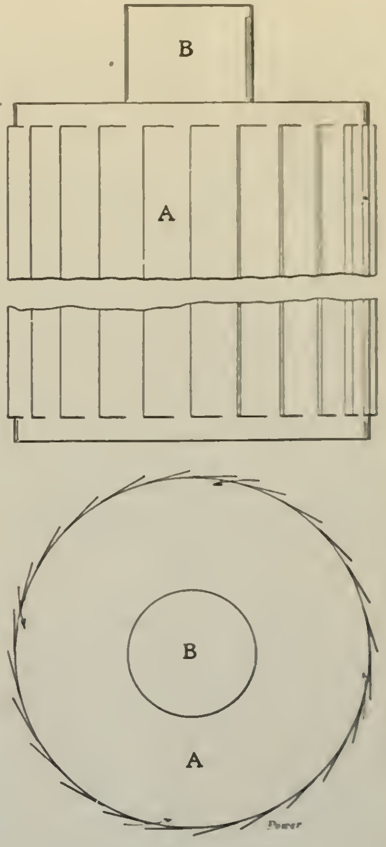
The writer experienced considerable annoyance from priming in the boilers and water being carried over into the engines, until the following scheme was devised, after which all trouble ceased.

The illustration shows the means employed for this purpose. *A* is a cylinder of heavy sheet metal, open at the bottom (although it may be closed if preferred), and at the top is a hole slightly smaller than the steam pipe where it enters the boiler. A flange *B* is soldered to the top of the cylinder for insertion into the steam pipe. Longitudinal slits about 2 inches apart are cut in the metal as shown and are bent outward, forming tangential strips with openings between them of a width of from ⅛ to ¼ inch, according to the size of the separator and the amount of steam which is to pass through.

The apparatus is placed in the boiler with the bottom resting on the flues and the flange *B* projecting into the steam pipe. Its operation is as follows: Owing to the tangential disposition of the long and narrow openings, the steam flows into the cylinder in the direction indicated by the arrows and is thus given a rotary movement at high velocity. This motion causes the particles of water and solid matter to fly outward by centrifugal force, and these coming in contact with the walls of the cylinder adhere to the latter and run back into the water, while the steam, being lighter, passes toward the center and out through the flange *B*.

The flange *B* should be several inches long so as to extend a considerable distance into the steam pipe. The length and diameter of the cylinder *A* must be determined by the size of the manhole and the distance of the steam outlet above the top row of flues. It has been the writer's experience that a very small cylinder will take care of a large volume of steam. In the plant using this device the engine is 26x48 inches in size and runs at 76 revolutions per minute. The separator is 12x48 inches and the openings are ¼ inch in width and 2 inches apart, giving an effective area of nearly 200 square inches. Care must be taken to provide a sufficiently large area for the steam to enter. If the bottom of the cylinder be left open it must be below the water line, otherwise the steam will enter at the bottom and the effect will be lost.

This type of separator has an advantage over those located in the pipe lines, in that it is better to separate the water from the steam before it leaves the boiler, for the obvious reason that heat which would otherwise pass out with the water is thus retained. The use of this scheme has shown a perceptible improvement in the economy of the plant in which it is used and its trial is heartily recommended.



STEAM SEPARATOR

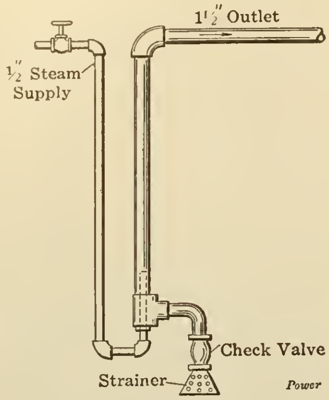
T. TWISTER.

Milwaukee, Wis.

A Homemade Ejector

The accompanying sketch shows a simple homemade ejector which I used successfully to pump out a well with a lift of slightly over 15 feet.

To start the ejector, the steam-supply valve is first opened wide, allowing steam to pass up through the outlet pipe. This creates a suction sufficient to lift the water about 5 feet. The check valve then closes due to the weight of the 5-foot column and does not open again until part of this water has been forced up into the outlet pipe and discharged. This operation causes a pulsation about every second, and a steady flow is obtained.



Ejector

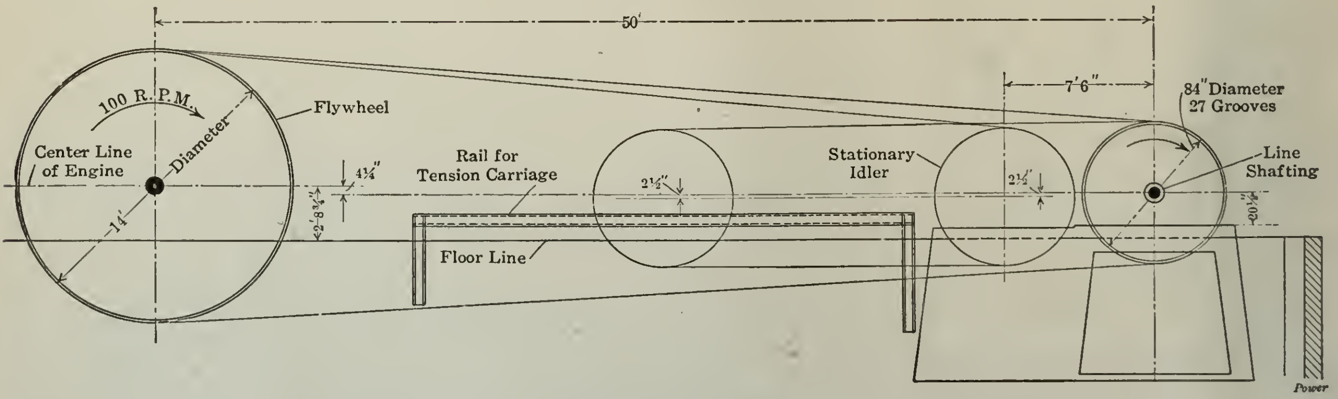
A. C. Harrison.

Jersey City, N. J.

Troubles with the Tension Carriage

The accompanying sketch shows the arrangement of rope drive in our plant.

We have four 500-horsepower Allis-Chalmers engines, noncondensing and each having a flywheel 14 feet in diameter with 13½ grooves for 1½-inch rope, so arranged that another wheel with the same number of grooves can be added when the engines are compounded. The speed is 100 revolutions per minute and the power transmission is connected to an 84-inch sheave upon a line shaft which can connect two engines together by means of a jaw clutch whenever the conditions of load warrant. The distance between line shaft and engine is 60 feet for one pair of engines and 50 feet for the other pair.



System of Rope Drive

Is the arrangement of the tension carriage and the stationary idler, as shown in the sketch, correct? The tension carriage will not automatically take up the slack in the rope, although there has been as high as 1400 pounds counterweight applied, and it has been found necessary to use block and tackle in addition to the counterweight to pull back the tension carriage whenever the rope becomes too slack.

It is the writer's opinion that the tension carriage ought to be placed next to the sheave on the line shaft instead of the stationary idler. However, there is considerable difference of opinion in this respect. Advice from those who may have had experience with similar conditions or who wish to express themselves would be appreciated.

T. C. Thompson.

Chanute, Kan.

Location of Boiler Feed Pipe

Where the feed water should enter a return-tubular boiler and how it should be piped inside are questions which have almost as many answers as there are engineers.

Many advocate entering at one side of the front head above the tubes, then running back at least two-thirds the length of the boiler and across to the other side with an elbow delivering the water between the shell and the outside tubes. The advantage claimed for this method is that the water is heated and prevents unequal expansion of the boiler parts. The economical effect would be "Robbing Peter to pay Paul," and at the best only a few degrees rise in temperature can be secured with only 10 or 12 feet of pipe. This method has a very serious defect, due to the difficulty in cleaning the inside pipes when they become filled with scale, and although it is argued by the advocates of this method that the sediment following the feed water is deposited farthest from the furnace, this would be true only in the case of sand, the usual deposits in the feed water being carried by the circulation inside the boiler and may be left at either end.

A great many boilers are fed through the blowoff. In so far as unequal expansion is concerned this method does not, like the preceding one, impinge the incoming water against the boiler shell, but it does materially assist in the stratification of the different temperatures, and this has a tendency to bow the shell. Nor does it in any way prevent deposits over the furnace. However, in this method the blowoff is protected from being burned, and with a tee located at the entrance to the boiler, the pipe can be cleaned without the necessity of removal.

Again, boilers are fed through the top with a delivery pipe extending down between the tubes to the bottom of the shell. This seems to be the poorest method of all. It causes stratification of temperatures and drives the feed water against the shell with the greatest possible force, sometimes reducing the temperature of the shell and setting up local strains.

This is illustrated in a case observed by the writer, where the feed water was used without being previously heated and in winter its temperature was almost to the freezing point. The boiler had a circumferential crack about 4 inches long over the furnace, caused by improperly driving up a bag, and was plugged with copper rivets. The fireman could not keep an even water level and when it became low, was accustomed to speed the pump to its full capacity until the proper level was gained. The crack was thus kept leaking all the time and the rivets had to be tightened every time the boiler was cleaned.

Another fireman, who was careful to hold the pump at a reasonable speed at all times, took charge and for a period of nine months the crack did not leak. Evidently under the care of the first fireman the shell stood some serious strains which would have been greatly reduced by feeding either through the front head and across to the side or by feeding through the blowoff.

Some boilers are fed through the front head at one side, and below the top row of tubes. It is about the same as the first method noted, except that the water is not heated in the least before it combines with water already inside the boilers.

Still another method is to feed through the top with the feed pipe reaching just below the lowest water level. A tee is often placed on the end of the feed pipe so as to deliver the water in each direction, and in some instances pipes, drilled full of holes, are screwed into the tee. This insures a thorough mixture of the feed water with that already in the boiler before it reaches any part of the shell. In other cases a pan or plate is set directly under the pipe to bring about the mixing of the water. This seems to be the most reasonable way to deliver the feed water into a return-tubular boiler.

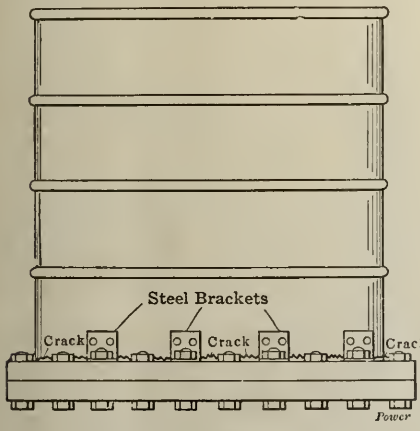
J. R. Early.

Columbus, O.

A Repaired Cylinder

The sketch shows a 90-inch compress cylinder which I patched 12 years ago and which is still in operation, and as strong as ever.

This cylinder had a crack extended all the way around just above the flange and caused, as I soon discovered, by tightening too severely on the flange bolts. The crack on one side extended nearly through the metal and the engineer in charge thought that a new cylinder was necessary and that the old one could not be patched.



Repair to 90-inch Cylinder

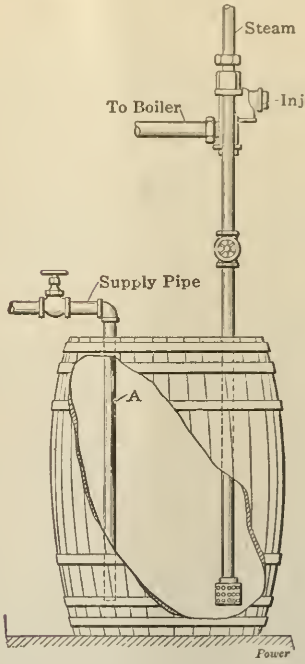
After satisfying myself that the crack had not been caused by the natural pressure which the cylinder was subjected to, and that it had not quite extended through the metal, I decided that it could be repaired. Accordingly I had twelve steel brackets made as shown, allowing the cylinder bolts to pass through the lower flange of the brackets, the other flanges being bolted to the side of the cylinder with ⅞-inch tap bolts. One of these brackets was fitted into every alternate pocket, and when bolted down made a substantial and satisfactory job.

T. K. McCarter.

Temple, Texas.

Trouble with an Injector

A few years ago while operating a steam shovel, I experienced trouble with a Penberthy injector in taking water from a barrel, as per sketch. The lift was about 4 feet and the supply pipe ran nearly to the bottom of the barrel. After starting the injector, I made it a practice to turn on the water, so as to keep the barrel full, but on so doing, the injector invariably refused to work.



Injector Drawing Water from Barrel

I concluded that the reason for this lay in the fact that the water entering at such a low point in the barrel, set up a swirling motion. I accordingly cut the pipe at the point *A* and did not have any more trouble with the injector.

Coram, Cal. J. H. Stretton.

Foreign Labor in Steam Plants

Sufficient help, both trained and untrained, is absolutely necessary for the performance or completion of any engineering work. Ordinarily, skilled labor is easily obtained and we find most employers ready to make any reasonable concession in order to obtain the services of men of ability. On the other hand, unskilled labor is absolutely necessary to a greater or less extent in most undertakings, and is permissible from a financial standpoint, so far as the rough end of the work is concerned.

We are being confronted daily with the proposition from the employer, that a certain amount of cheap labor, even in places requiring intelligence, is absolutely necessary in order to keep costs down and thus remain in business. Truly, we have no criticism to offer to any concern or individual for getting things done cheaply, but in the majority of cases is the work done intelligently? Having been up against this proposition for a number of years, the writer has reached the conclusion that the subject is at least worthy of discussion.

In order that any steam plant may operate satisfactorily and safely, men must be employed on whom dependence can be placed. Are we able to draw this grade of material from among this cheap class of laborers? Can the average foreign laborer be depended upon? Do enough of them, even after several years of continuous service, exhibit a sufficient degree of intelligence to make their retention an economical and safe proposition? Do they, as a rule, exhibit that sober, trustworthy and dependable disposition that is so necessary in the operation of a steam plant? What conclusion can we reach? To each question we might answer both *yes* and *no*, with equal candor and truthfulness. They are human and consequently are subject to all the failings of the human race in general, but are they not more subject to these failings than the average untrained American?

It is not the writer's desire to discriminate unnecessarily against persons of other nationalities in any class of work. Some of our greatest engineering achievements are of foreign origin and to these we join in hearty tribute. It is not of this class that I speak. We are called upon to maintain the profession in its high standard, and can it be done under the existing conditions? I have positive knowledge of concerns employing foreign laborers who in the course of a few years have showed enough reliability and intelligence to be placed in charge of large boiler plants. Again, I have had under personal observation, men of this class for several years and find that with the best of treatment and careful instruction, while entirely satisfactory so far as the rough work is concerned in firing, yet they cannot be trusted to regulate the water in the boilers.

Those of you, who for years have been in charge of large plants and have had these fellows to contend with, do you find it a source of comfort and profit? Or addressing the manufacturer and employer in general from a financial standpoint, does it pay and should we be asked to tolerate it? Are they really cheap at any price?

Safety first, economy second. To what extent can we tolerate these conditions and win?

T. M. Sterling.

Middle Branch, O.

Setting Link Driven Valves

In setting Corliss valves, before thinking of lead, compression, etc., we first "square the valves." In setting a common slide valve, the same procedure should be followed. First, the eccentric is placed at mid-stroke; to do this, turn the full side of the eccentric to the extreme inner position and place a mark on the valve stem the width of a rule from the stuffing-box gland. Then turn to the extreme outer position and mark stem as before. Bisect the distance between these two marks and turn the eccentric till this middle mark just touches the rule, and fasten the eccentric in that position; which is just at mid-stroke. Now adjust the valve rod or stem till valve is in mid-position—that is, lapping each port equally. The valve is now squared. All that remains to be done is to put the engine on one center and revolve the eccentric until the proper lead is secured.

Now I should adapt this method for link-driven valves as follows: First, disregarding the position of the crank, drop the link and find the mid-position for the ahead eccentric as outlined above, and fasten the eccentric. Next, lift the link and set the astern eccentric in the same way and fasten it. The relative position is of no account for the present. One eccentric may be up and the other down, or both may be up, or both down. With the link up, adjust the valve to mid-position, thus squaring it with the astern eccentric. Drop the link and see if the valve squares with the ahead eccentric. If it does, the valve is properly squared; if not, adjustments are necessary in the length of eccentric rods. Let these be made and the link raised and lowered again, to test the work. The valve is now squared with both eccentrics. Place the engine on either center and pull the link all the way up and turn the ahead eccentric in the proper direction until the desired lead is secured. Fasten the eccentric there. Now drop link all the way down and advance the astern eccentric to secure the same amount of lead, and fasten it. If the work has been carefully done, the valves are now set. But to test our work, turn the engine to the other center and measure the lead for each position of the link. If a mistake has been made, it may be rectified now by making half the correction on the eccentric rod and half by moving the eccentric.

For instance, suppose we have set the valve on the head end with 1/32-inch lead. Upon testing, the astern position is correct, but in the ahead position we have 1/32-inch lap instead of lead. Any adjustment upon the valve stem would upset the correct setting for the astern position, so we must leave the stem alone. Following the rule, we will adjust the eccentric rod. The total error is 1/16 inch, so by lengthening the eccentric rod the valve is moved 1/32 inch toward the head end. This will bring it in line with the port. It will also obliterate all lead on the head end. Now complete our rule and correct the other half of the error by advancing the eccentric until the valve moves 1/32 inch. This same movement will restore the lead on the head end.

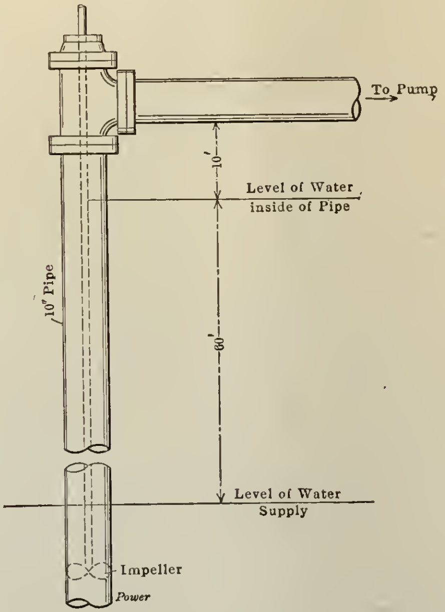
The advantage of this method is that it saves a lot of labor in turning over the engine; also, there is no guessing at adjustments. Whenever adjustment is to be made the precise amount is known. It is the simplest, shortest and best method for a simple slide-valve engine, and, although never having used it on a reversing engine, I cannot see why it is not just as applicable to the latter as to the former.

William E. Dixon.

Hudson, Mass.

Will the Pump Raise the Water?

Will some reader of Power kindly enlighten me upon the following question: Referring to the accompanying sketch, if the water in the pipe is raised to a hight of 60 feet above that of the surrounding water by an impeller or other mechanical means, will a pump situated 10 feet higher, create enough suction to raise the water the remaining distance?



Supply Pipe

W. F. Callister.

Memphis, Tenn.

Should Superheaters Be Used?

We are investigating the advisability of using steam superheaters in several of our small boiler plants. These plants each consist of from two to four 72-inch by 18-foot return-tubular boilers carrying 125 pounds pressure. Much of the steam generated is carried a distance of from 900 to 1500 feet, and we believe that by superheating enough to give us dry steam at the end of the long lines we could save considerable. We would be very glad to hear from some of the readers of Power who have had experience with such installations, giving, if possible, a sketch or description of the superheaters used.

A. B. Holley.

Virginia, Minn.

Are the Rim Bolts in Tension or in Shear?

At a plant in this locality a new driving pulley is being installed and is causing much discussion. It is 13 feet in diameter with a 72-inch face and is built up in halves with arms as shown in the sketch. The wooden rim is to be placed after the wheel is in position and will be held to each of the arms by four 1½-inch bolts.

The controversy which has arisen among the engineers is in regard to the stress on these rim bolts. Some say that the bolts will be in shear while others say that they will be in tension. Will some reader of Power kindly settle the question.

Patrick Mulham.

Adams, Mass.

Do Factors of Safety Increase with Age?

It is claimed by some authorities that the factor of safety increases with the age of a boiler. Such an increase would often prove very unsatisfactory to steam users by cutting down the efficiency of their plants, and often making the load too much for the engines.

What I would like to know is, "how much authentic evidence is there to be had in support of the theory that boiler material weakens or becomes fatigued with age?" Of course, by this I do not mean weakening caused by the material becoming thinner.

Has it been proved that the temperature changes to which a boiler is subjected, have any effect upon the strength of the steel? Or, do continual variations in stresses below the elastic limit produce cracks or weaken the material? I should like to see the theories expounded upon this point.

I should not be at all surprised to find that the statistics on boiler explosions show that more old boilers explode, even where inspected, than do new ones, but is this not largely due to other defects than a change in the strength of the steel?

However, I am not opposing the theory but feel that it should be quite well established before any legislation is based upon it.

H. H. Hastings.

St. Louis, Mo.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Compression

The interesting article by William Westerfield in the April 26 number prompts me to submit a few experiences.

While it is true that diagrams of certain character indicate certain conditions in the cylinder, it must not be forgotten that of the various types of engines or of valve gears each has a characteristic diagram. A single-valve engine yields, normally, a different diagram than does a four-valve. Yet, under certain conditions, the diagrams of two engines of dissimilar type might closely resemble each other. In one case the economy might be high, and in the other it might be very low.

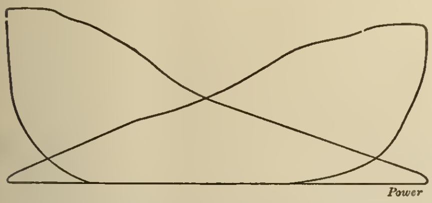


Fig. 1. Diagrams from Four-valve Engine

The diagrams in Fig. 1, which were taken from a four-valve engine of high economy, resemble somewhat those of Mr. Westerfield's Fig. 3. The engine from which my diagrams were taken is 8x10 inches in size and operates at a speed of 275 revolutions per minute, which gives a piston speed of 458 feet per minute. It is evident from this that this engine is not making its good showing on account of high piston speed. The explanation lies, however, in the fact that it is fitted with Corliss type valves, which are so fitted in the cylinder that there is but 2½ per cent. clearance.

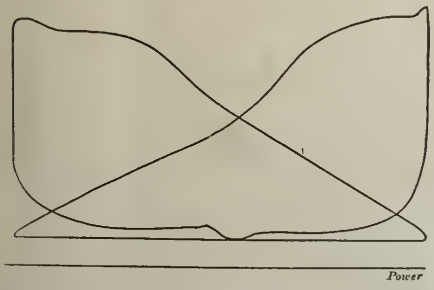


Fig. 2. Diagrams from Vertical Cross-compound Engine

In Fig. 2, I present diagrams from a vertical, cross-compound engine. The exhaust valves are of the Corliss type while the admission valves are of a multiple-ported plate type. These valves give a very low percentage of clearance. The cylinder is 16x36 inches in size. The engine operates at 160 revolutions per minute under a steam pressure of 160 pounds. This engine shows remarkably good economy. I have not at hand the figures on the water rate, but the coal consumption is only 1.72 per horsepower per hour.

The diagrams shown in Fig. 3 are from an original George H. Corliss engine. They do no discredit to the design. The engine is 22x48 inches in size and runs at a speed of 86 revolutions per minute. The compression is moderately low and the economy good.

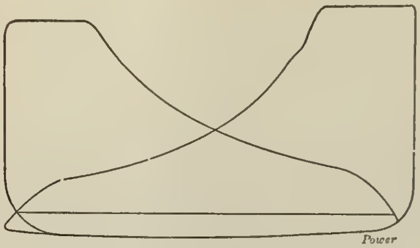


Fig. 3. Diagrams from Corliss Engine

In general I have found that with a heavy, varying load too much compression and too quick admission do not produce good economy.

C. R. McGahey.

Sheffield, Ala.

Boiler Blowoff

In the May 31 issue, Milton Heglin replies to G. E. Miles in regard to the above saying that scale lodges under the perforated blowoff pipe, causing bagging and bulging of the sheets in time.

The pipe does not take up much space on the bottom of the boiler, and there is little chance for scale to lodge under it, provided the boiler is not run too long without cleaning. A boiler equipped with such a blowoff would not scale up so quickly as the mud is carried off better. The idea of having just a common blowoff connection is absurd, as such a small hole, 1½ to 3 inches, according to the size of the boiler, will only carry off such mud as settles in or immediately around it.

In my opinion the proper time to blow off a boiler is in the morning before firing up or, as Mr. Miles says, when the fires are banked, as it is then that the water is at rest in the boiler. When the boiler is steaming, the mud is in circulation with the water and blowing off is of little use, unless all of the water is blown out, which, of course, is impracticable.

Frank Gartman.

Sheboygan, Wis.

Why Didn't They Explode?

The first paragraph on the first page of Power for June 7 is a whole lot said in the nearest way through.

As an inspector of steam boilers with a number of years' experience to my credit, or discredit as the circumstances may warrant, I have often been compelled to admit the truth of the statement that sometimes a boiler will explode for no apparent reason. Again, we can find no good reason why they do not explode.

I was, very recently, called upon to inspect a new "risk" in the mountains of Tennessee. It was a very out-of-the-way place, to reach which I first boarded the "Dixie Flyer," rode a few dozen miles, then took a combination freight and passenger train, rode a dozen or so miles further, then descended to a push car and after a mile or so of this I was confronted with the method of transportation in vogue in Adam's time, so I picked up my grip and tramped a mile further up into the hills to a quarry where the proprietor had a second- (or more) hand boiler on a flat car awaiting my arrival. The first view of the boiler I had was a full-length one with its bottom toward me. I stood still and counted nine separate and distinct patches and remarked on them to the prospective buyer. He replied that he thought those were patches but was not sure. Further and more intimate examination proved the boiler to be practically worthless as a steam generator; it had nearly all of the boiler diseases.

After I had emphatically expressed my disapproval of this boiler the proprietor suggested that I go down to the boiler house and take a look at the boiler then in use but which it was proposed to replace with the boiler I had just inspected. On arriving at the boiler in service, I first noted that the steam gage registered 90 pounds, and as the gage looked new, I surmised that it was somewhere near correct. The water column and gage glass looked all right and seemed to be properly connected. The flue cap was opened; it was a horizontal return-tube boiler, 60 inches by 16 feet in size. I noticed that nearly all of the tubes were leaking at their front end. The engineer (?) stated that it was impossible to keep them tight because the head was not over an eighth of an inch thick, due to corrosion. The attendant then asked if I would like to look in the furnace and opened the furnace door. On looking in I noticed quite a leak about the center of the first sheet over the fire. As close an observation as it was possible to make revealed the information that there was a longitudinal crack in the sheet about 6 inches long, through which the water was coming in a stream. By questioning the attendant I verified my opinion as to the dimensions of the fracture; then looking at my watch I remarked that I would have to hurry if I caught my train.

Now, in this case there was utter ignorance on the part of all connected with the plant as to the danger they were in. Their idea was that as the boiler was very weak and already had numerous leaks, nothing could happen to it which would be very serious. I am expecting any day to hear that there are a number of the employees of that quarry being picked up in baskets and distributed around to those of their families who care to take the trouble to inter infinitesimal portions of their kinsmen.

My position was such that I could not expand on the merits and demerits of the situation to those intimately exposed to the danger. I did that once in the case of a locomotive and because everybody boycotted that particular boiler on account of what I had said, the owner returned all his boiler policies for cancelation, at the same time exuding the information that he never more wanted a meddling boiler inspector around interfering with his business. Of course, my boss fell on me like a thousand bricks for saying anything to the employees, and the boss was, from a business standpoint, right, but I felt that I owed it to the men who were operating the locomotive to warn them to be ready for a trip to the beautiful shore unless they desisted in their efforts to operate that boiler. Since that time I have grown hardhearted, and besides, I have to look out for my own bread and butter, and I take pains to keep a close mouth to all except the highest in authority; a few more returned policies might cause me to have to look for a new job.

Any boiler inspector can tell that there are often occasions when he meets with circumstances where loss of human life and much property are among the probabilities, judging by the condition of boilers, and we often feel constrained to overstep our instructions and warn the men who would be most likely to suffer personal injury. It is needless to say that in the majority of instances in which a report of such conditions is made to the owner or manager it never gets any further because of the fact that it might interfere with the output of the plant. I once, late on a Sunday afternoon, found a boiler in a sawmill plant in which all the braces in the rear head above the tubes were broken, a crack extended all the way round the segment in the knuckle of the flange, the crack being visible only from the inside. I called the engineer around and sent out for the superintendent who came and took a look for himself and then called the manager in. The latter official said he could not shut down for repairs at that time and asked me for permission to operate the boiler for a few days until some urgent orders could be filled; of course, I could not assent, and was told that the boiler would be fired up next morning as usual; so I immediately made tracks for the telegraph office and suspended the policy on that boiler. The boiler was operated for a week and nothing happened.

I am constrained to believe with the editor that in such instances as the above a watchful Providence alone prevents disaster.

A. C. Terlene.

Chattanooga, Tenn.

Connecting Rod Crosshead Joint

In reading Mr. Blackwell's communication in the May 17 issue, I am reminded of the experience a friend had with the screwed piston rod and lock nut. A 22x48-inch Corliss engine was fitted in the above manner and a tram supplied by the makers to adjust the rod to the proper depth in the crosshead. A pound developed in the cylinder and investigation showed the jam nut to be perfectly tight, but an attempted application of the tram disclosed the fact that the tram mark was not in sight. By slacking the nut and turning the rod around till the tram mark was visible it was discovered that the rod had turned in the threads the length of the clearance, allowing the piston to strike the head.

Subsequently, I had an experience with a 16x36-inch Corliss engine which was equipped as above and had a piston with a single ring which had worn a shoulder in the cylinder. A continuous snapping of the ring at one end of the stroke one morning led to an investigation and the same condition was found as in the former case.

These instances prove that a piston rod may turn in the crosshead in spite of a tight lock nut.

In this type of connection the nut, being drawn up, clamps one face of the threads of the rod against one face of the threads in the crosshead, the thrust on the out stroke being taken by the nut while the pull on the in stroke is sustained by the threads in the crosshead. This is an alternation of strains conducive to looseness, while in the split crosshead the threads of the crosshead and rod are tightly wedged into each other, both faces of the threads having equal bearing. In addition, when the pulling apart strain exerted on the rod by the tightened jam nut in addition to the working strain is considered, the split crosshead seems to be the most correct in mechanical principle.

F. C. Holly.

Yazoo City, Miss.

Boiler Accidents

The importance of piping a boiler so that nothing can get into it except that which is intended to go in, is often overlooked, as was called to mind by the article, in Power of May 24, "Soap from Boiler Stops Engine." Fortunately nothing of a disastrous nature resulted on that occasion, but to bring the subject more forcibly to mind I will relate two instances which were disastrous.

A Stirling boiler at Sherman, Cal., several years ago, scattered its various parts over the surrounding country because of an endeavor to make steam from heavy crude oil by putting the oil in the boiler instead of in the firebox. The boiler was equipped to burn crude oil and was arranged to blow the sediment from the oil line with steam. By mistake, the valve was left open or leaked, and the boiler filled with crude oil which carbonized on the shell and tubes. The intensity of the fire heated them to redness, the bottom drum gave way and there was a pyrotechnic exhibition which rivaled Halley's comet as the parts of the boiler, blazing with oil, went flying through the air.

At Sandwich, Ill., about seven years ago, there was installed a return-tube boiler 72 inches by 18 feet in size and having a triple-riveted butt joint. The plant had been shut down for two weeks when one of the boilers exploded, wrecking the plant, killing one man and injuring several others. There had been no fire under the boiler for two weeks. The only explanation was that a valve on the ammonia system either leaked or was left open and the ammonia was allowed to get into the boiler with disastrous results.

While it is well to keep in mind the importance of keeping things from entering the boiler which have no business there, it is of as much importance to be able to get them out when they get there in the ordinary course of events. There are two return-tubular boilers in a plant near Whittier, Cal., which were operated for about five years without having a blowoff pipe attached. The space between the tubes became packed solid with scale. The tubes had been burned and rolled until the tube holes in the heads were a half inch larger in diameter than the tubes; the heads on one boiler were bulged nearly 2 inches, one being cracked; the shells bagged. New tubes were recently put in with the space around the ends filled with sheet copper and rolled; the bags were driven back. The boilermaker insisted upon putting on a blowoff. Those boilers are being run today—death traps. The boy in charge fired one of those boilers while the boilermaker was there, before it was repaired, and at noon put out his fire, shut off all of the valves, leaving 80 pounds gage pressure in the boiler. When the boilermaker returned from dinner the gage registered 120 pounds. There was a hasty exit of the boilermaker. The boy is not to be blamed for he could not understand how the heated furnace without fire could raise steam. I cannot understand why the boiler did not blow up.

There was, until recently, a small laundry operating near here with the safety valve hanging on a nail in the brickwork of the boiler setting. When asked why it was not in its place on the boiler, the "engineer" said it wasted too much steam so he took it off.

Yet we have no State or national boiler-inspection laws!

FRED W. AXE.

Los Angeles, Cal.

Air Chambers on Pumps

Like Harold James, in the May 17 issue, I cannot understand why more manufacturers do not fit their pumps with air chambers. Again, of those that do, why do some put the air chamber on the discharge, while others put it on the suction? Several years of experience with pumps in the mines of northern Michigan have convinced me that the air chamber is a very important part of a pump. I have had to do with a large assortment of pumps, including single- and double-acting pumps; chest and pot forms, piston or water block, and plunger types; steam, gasolene, air and belt driven; and have yet to see a reciprocating pump that could not be improved, in regards to both action and economy, by a properly designed and located air chamber.

Some engineers and manufacturers will tell you, "Why, we put a relief valve on our pumps and can set it to relieve the discharge of any excessive pressure." You bet they can. But the "Old Man" pays the coal bill.

There is excessive pressure at the start of every stroke that a pump makes. The more speed a pump develops, the more excessive is the stroke-start pressure. Put a suitable size of relief valve on a pump running at its rated speed, and set it low enough to relieve the sudden high stroke-start pressure (which causes water hammer in the valve chambers and pipe fitting) to the same extent that an air chamber would, and it will be found to be discharging almost, if not quite, continuously. It costs money to force water through the resistance of a relief valve. Measure the discharge of a relief valve set to eliminate the water hammer, and find just what that method costs. Then compare this cost, for a period equal to the life of an air chamber, with the cost of a chamber and you will have dollars and cents in favor of the air chamber in an amount that will surprise you, as lots of things do when considered in a financial way.

It is well to consider any mechanical problem, first with regard to safety and second from a dollars-and-cents point of view.

Use a relief valve on any pressure system, by all means, but set it as a safety device and not to relieve part of the working pressure in order to secure smooth running.

By reducing the maximum discharge pressure the air chamber saves money in packing. By running with looser packing there is less friction on the plungers. By minimizing or doing away with water hammer, money is saved in the maintenance of pipe lines and pumps, especially pump valves and valve springs.

Where the water supply is below the level of the pump, and especially when the water is cold, the hammer in the suction pipe is considerable unless the pipe and fittings are of excessively large size.

Although the closer an air chamber is placed to the discharge valves, the more efficient it is, it will give good results in other positions.

If you have a feed-water system that is noisy, or seems determined to leak and blow gaskets, try an air chamber on the pump.

WILFRED E. BERTRAND.

Upper Darby, Penn.

Central Station versus Isolated Plant

In POWER for May 10, W. E. Crane relates his experiences with exhaust-steam heating and states that it was found to be cheaper to use live steam for heating and run condensing, and then states that under such conditions it would be cheaper to buy current from a central station. I am surprised to hear of such a coincident. It would certainly appear plausible, if the exhaust from the engines was entirely made use of to heat the buildings and none of it was exhausted into the atmosphere, that nothing would be lost by such a procedure; also, if we were to change over and run condensing, the same amount of steam would have to be used for live-steam heating as was used for running the engine when running noncondensing. So, in such a case, it would be necessary not only to evaporate enough water to run condensing but also to evaporate enough to run noncondensing. If only a portion of the exhaust were made use of and the balance exhausted into the atmosphere, then under those conditions it might be cheaper to run condensing and use live steam for heating, but whether it would be cheaper to buy current or not would be dependent upon being able to dispense with the greater part of the regular operating force, which is seldom the case as competent men are required to look after the machinery, etc., such as elevators, motors, lights, boilers and piping in case of an office building or hotel.

This section of the country (the Miami valley) is full of paper mills, and not in a single case do the paper machines dry paper with live steam but the exhaust from the engines driving the paper machines and Jordons is used. In mills making a heavy sheet such as boxboard you will find not only the engines driving the paper machines running noncondensing but the main engine as well, which furnishes power for the beaters, etc.

It has been considered good practice to make a pound of paper with a pound of coal in this locality. Recently a mill has been started which is considered to be the model paper mill of the country. This mill runs entirely noncondensing; even the electric-lighting engine exhausts into the main exhaust together with the other engines. In this mill they are making a pound of paper with two-thirds of a pound of coal, which is considered to be extraordinary.

Several mills in the valley have experimented to their sorrow with a central generating plant and motor drive throughout. All of these mills have abandoned the practice.

Mr. Crane speaks of an incident in an office building in which they found it cheaper to buy current and heat with low-pressure steam. I am intimately acquainted with the conditions existing in the larger hotels and office buildings of a neighboring city of some 400,000 inhabitants. In not one single instance have any of them found it cheaper to buy current, nor have any of them been sufficiently suffocated by the arguments set forth by the clever central-station solicitors and power experts, nor bulldozed by the smoke inspector, to a sufficient extent to abandon their private plants and buy current from the central station. I do not wish to imply that they are a bull-headed lot and not willing to be shown, for they have been shown. I know of some hotel people who used central service at 1¾ cents per kilowatt, furnishing their own heat. They finally got wise and put in their own plant and found that they did not burn any more coal during the six months of the heating season than when buying current, also that the central station would have to supply current at three-quarters of a cent per kilowatt the year round to compete.

The managers of a large office building found that very little additional coal was required during the heating season when operating their own plant as compared to conditions when buying current. In both cases just related, the labor was identically the same as when buying current.

Charles W. Naylor in Power for May 3 makes the following assertions: "The central station captured the plant by offering a low rate."

How is it possible for the central station to supply current cheaper than the isolated plant, when the vast amount of money invested in transmission lines and underground conduits is considered, also the extent to which the stock is watered and the profits that must be made to pay only a very modest dividend?

"The service has been practically uninterrupted and almost perfectly satisfactory."

Is this saying anything that cannot be said of a modern isolated plant under able and efficient management?

"The central station is lowering the cost of current generation rapidly."

Since when have all the improvements in economy been created solely for the central station? Is it not a fact that the generation cost is only a very small part of the cost of the delivered current, even if sold at the actual cost of production, every item being considered?

Mr. Naylor asks, "Will the isolated plant be able to withstand the onslaught of the central station?" to which I very affirmatively say yes, provided the engineers in charge know exactly how much it costs them to furnish current and heat and elevate themselves to a position of due respect.

Mr. Naylor says, further, that owing to the high land value and the fact that coal must be hauled two or three miles by wagon, also that the sidewalks must be kept clear for patrons' carriages, central-station service is cheaper when it is considered that the noise, dirt, danger and labor of the isolated plant are abolished. If this is so, why do so many of the most recently erected hotels and office buildings have their own plants? Also, why do not the alleged dirt and noise prevent the installation of private plants in such instances in which the most discriminating traveling public are to be dealt with?

As for the sidewalks, I am at a loss to understand how Mr. Naylor's plant disposes of the ash and receives the coal for heating the buildings, which certainly is considerable in so large an establishment. Down in this "neck of the woods" this item has been found to require approximately the same quantity of coal as when using central-station service.

Mr. Naylor speaks of the coal being hauled two or three miles as a good reason for the abandonment of their plant. Why do so many of the large hotels and office buildings find it profitable to haul coal a like or even greater distance?

In conclusion, I would say that the isolated plants of the better class are not "captured" by the central station, for the reason that they can save the owners money. When one is, it is because of the high-class salesmanship (which must also be paid for by the "taken-in" isolated plant) used in hypnotizing the private-plant owners or managers. Again, in quite a number of cases the owners or managers hold stock in the central-station company.

Willard Bogart.

Middletown, O.

Operating Costs of Large Units

In the article in the May 31 issue on "Operating Costs of Large Units," the following statement is made in the third paragraph on page 983:

"With eight boilers of 500 square feet of heating surface each, making a total of 40,000 square feet, furnishing steam to one turbine developing 14,000 kilowatts, a kilowatt will be generated by

$$40{,}000 \div 14{,}000 = 2.85$$

square feet of heating surface, or a horsepower by 2.13 square feet, which is considerably less than the 10 square feet allowed by the ordinary boilermakers' rating."

It is undoubtedly true that under the conditions assumed a kilowatt will be generated for every 2.85 square feet of heating surface working in the boiler. Likewise, it is true that this corresponds to 2.13 square feet of surface in the boiler for every horsepower generated. However, this amount of surface per horsepower delivered at the switchboard cannot be compared with the ordinary boilermakers' rating of 10 square feet per boiler horsepower.

A few calculations will serve to show the amount of error. According to the ruling of the American Society of Mechanical Engineers, a boiler horsepower consists of the evaporation of 30 pounds of water at 70 pounds pressure from a feed temperature of 100 degrees Fahrenheit. This is equivalent to 33,305 B.t.u. per hour.

Assuming the same figures as are assumed in the latter part of the article, we find that the heat per pound of steam leaving the boiler is 1290 B.t.u. As we are not considering superheater surface it will be necessary to subtract from this the B.t.u. imparted in superheating. This will amount to

$$150 \text{ degrees} \times 0.60$$

the specific heat of superheated steam which would decrease the total heat per pound of steam to 1200 B.t.u.

Assuming now that the feed water to the boiler comes from an open heater at 210 degrees, we have the total heat imparted to the water per pound by the boiler-heating surface as

$$1200 - (210 - 32) = 1022 \text{ B.t.u.}$$

Dividing this into 33,305 B.t.u. per hour, which is equivalent to a boiler horsepower-hour, we get an amount of steam per boiler horsepower-hour for these conditions of 32.6 pounds per hour.

Using the figure 35,000 pounds per hour per boiler, we find that at this rate of driving, the boiler horsepower would be 35,000 divided by 32.6, which equals 1074 boiler horsepower. The square feet of heating surface per boiler are 5000, so that the number of square feet per boiler horsepower is 5000 divided by 1074, which equals 4.65 square feet. This figure can now be compared with the ordinary boilermakers' rating of 10 square feet per boiler horsepower. Roughly speaking, we would conclude that the boiler is being driven at about 100 per cent. over rating. This rate of driving is very often achieved for short periods, and where economizers are used the efficiency of the whole boiler plant is in no way impaired by the high rate of combustion.

This rate of driving corresponds to about 50 per cent. overload on the turbine, so that when the turbine is operated at its normal load of 14,000 kilowatts, the pounds of steam per boiler would be only 22,750 instead of 35,000 and the square feet of boiler surface per boiler horsepower developed would then be about 7.1 instead of 4.6. Thus, at full load on the turbine the rate of driving of the boilers is but about 40 per cent. over rating.

Paul A. Bancel.

New York City.

Receiver Pressure

I have taken great interest in the discussion on the proper compression in steam engines and receiver pressures. It appears to me that the proper compression or the amount of lead to give an engine is not the same for all engines, but must be regulated to suit each individual engine.

The receiver pressure also must be regulated. Each case has its own peculiarities. For instance, two engines are identical in all respects, made from the same patterns, machined by the same men and to the same scale. When sent out they are erected by two gangs of mechanics. The foreman of one gang must have accurate alinement and perfect work. The foreman of the other gang is not so particular, consequently the discrepancy is considerable. When steam is turned on and the engines tested, it is found that one requires more steam to perform the same amount of useful work. The extra steam is eaten up by inferior workmanship and excessive friction.

H. R. Blessing.

Philadelphia, Penn.

Modern Rotary Steam Engines

By Warren O. Rogers

Many types of rotary steam engine have been designed, but faulty mechanical construction, excessive steam consumption or a combination of both have prohibited their commercial use. Several makes of this type of prime mover are given attention in this article, and a study of their construction will aid the reader in forming an opinion as to their practicability.

In Fig. 1 the engine is shown mounted on a subbase, to which the high-pressure and low-pressure cylinders are secured. Each cylinder is placed midway between two bearings. The shaft passes through the side covers with a free clearance. A cylindrically shaped rotor piston is eccentrically attached to the shaft and the weight of its projecting periphery is counterbalanced, and each engine is counterbalanced within itself. The two cylinders are joined together, and the moving parts as a whole are balanced.

Fig. 2 is a sectional view through the center of the high-pressure cylinder. The shaft, where it passes through the side of the rotor casing, is made steam tight by packing boxes. The rotor piston is mounted on the shaft eccentrically, so that the projecting periphery of the rotor passes the inside wall of the casing with a clearance of less than one-thousandth of an inch on each side. This clearance space is made steam tight by a metallic strip of packing placed in a groove, and pressed outwardly by a flat spring.

Leakage of steam and excessive friction are prevented by a packing ring, contained within a packing ring. The overbalanced periphery is relieved of its extra weight by means of holes passing through the rotor from side to side, which connect depressions on each end of the rotor, and allow any leakage of steam to give equal pressure on both sides of the piston.

The hammer-shaped rider abutment is pivoted at the handle end by a hollow bearing passing through the side casings. This bearing is placed in the direct line of resistance to the steam pressure, with the rider portion between the entrance and the exhaust port. The face of the rider is made steam tight by a metallic packing strip pressed against its vibrating radial surface, by means of a flat spring.

Attached to the rider end is a hardened strip with a cylindrical toe. An intermediate metallic saddle plate, cast in a circular segment, is placed between the rider toe and the rotor. The saddle fits closely to the moving rotor, both being of the same radius, and is held in place by a radial arm hinged to the abutment bearing.

Attached to the handle of the rider is a pivoted pressure rod that enters the hollow end of a pressure piston which closely fits a cylindrical opening, in the outer space of which the steam pressure holds the entire mechanism against the periphery of the rotor.

The saddle has a tongue piece that projects into an opening of the entrance port and holds the rotor packing strip in place. It is far enough forward to form, while in operation, a leverage that prevents the back end of the saddle from being thrown up as the projecting end of the rotor piston revolves. The saddle glides over the surface of the revolving rotor with a rising and lowering rocking motion, while the rounded toe of the rider moves up and down with a constant pressure against the center of the rocking movement of the saddle. The saddle in turn is pressed against the rotor, thereby preventing the steam from taking the short passage from the entrance port to the exhaust port.

The port through which the steam enters is governed by a sliding revolver-barrel-shaped valve operated by an eccentric, which is attached to the main shaft outside the main bearings, thereby admitting steam once at every revolution of the shaft.

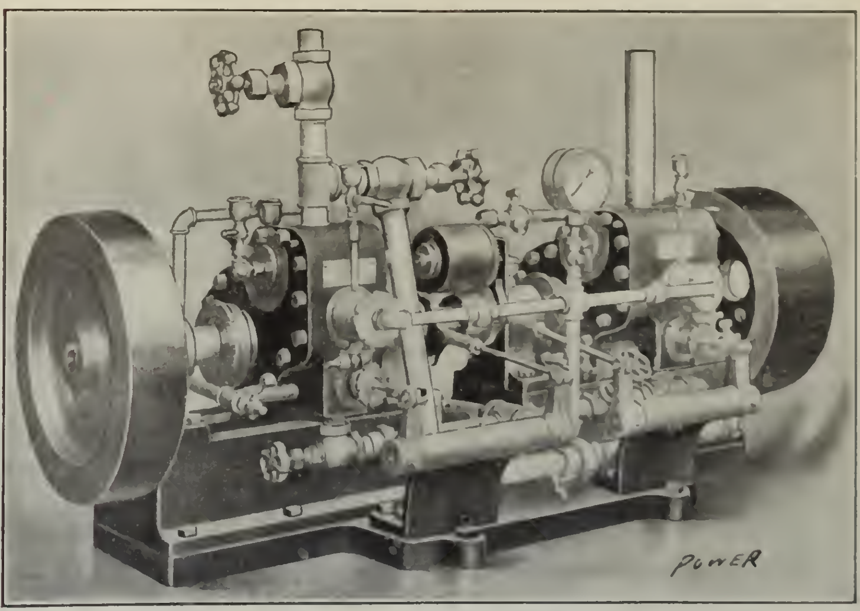


Fig. 1. Harriman Rotary Engine

This valve closely fits into a cylindrical inner casing having steam ports near the front edge of the valve. The valve has holes passing through it lengthwise, which allows some of the steam to pass through and thus equalize the pressure at both ends of the valve, giving it a free movement while in operation.

This casing is entirely surrounded by live steam under constant boiler pressure. The steam enters the steam chamber behind the rotor, just as its projecting end reaches the bottom of the entrance port. The steam, being prevented by the rider abutment and saddle from passing directly to the exhaust, exerts its pressure in the space behind the rotor piston and forces the piston around. The steam is cut off when the rotor piston is half way to the exhaust chamber, and the expansion of the steam continues to force the rotor to the exhaust chamber, while the momentum continues the revolution of the rotor to the entrance port again.

The steam already in the cylinder continues to expand, and forces the rotor piston around until the expanded steam is released at the exhaust chamber, and then passes to the low-pressure cylinder through the exhaust port. The rotor piston continues its revolution under the acquired momentum, and under the alternate force from the low-pressure cylinder, until it again reaches the bottom end of the inlet port where the steam is admitted

Having done its work in the high-pressure cylinder, the steam passes directly to the low-pressure cylinder, where it again repeats the cycle described for the high-pressure cylinder.

The interior parts of the engine are lubricated by oil entering with the live steam, and also through the hollow shafting that passes through the handle of the rider abutment. It passes down through holes in the handle of this rider and through the rider itself to the rounded steel toe which presses on the saddle. It also passes through the holes in the hinged arm piece to the saddle. Tests made by Edward F. Miller and Theodore H. Taft show the steam consumption of a Harriman 25-horsepower engine to be 31.02 pounds of steam per brake horsepower-hour. This engine is manufactured by the Harriman Engine Company, 58 State street, Boston, Mass.

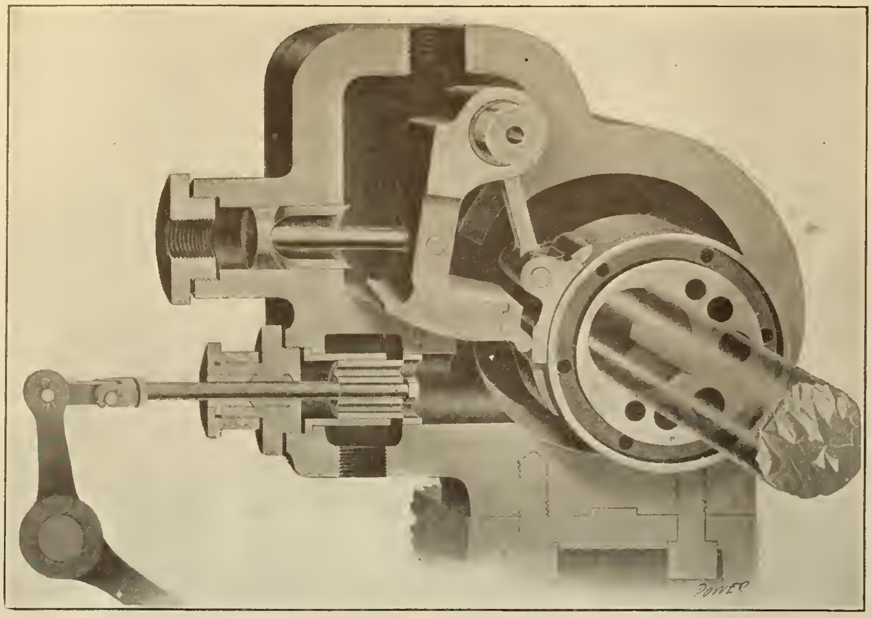


Fig. 2. Sectional View of Harriman Rotary Engine

Knowles Engine

This engine, Fig. 3, is the invention of John Knowles, Colorado building, Denver, Colo. It consists of a pair of engines arranged side by side, each being provided with two intersecting cylinders containing rotary pistons coöperatively arranged, and having a driving shaft arranged axially through both the upper and lower horizontal cylinders and pistons of both engines. The pistons and shaft rotate in unison. The engine is also provided with an adjustable rotary cutoff-valve mechanism that will permit the steam to be cut off at different predetermined points or parts of the revolution of the driving pistons in their respective cylinders.

In Fig. 4 is shown a vertical longitudinal sectional view of this engine. Fig. 5 is a transverse vertical sectional view through the rear cylinder, showing the pistons in the position they occupy relative to each other immediately preceding the admittance of steam to the cylinder. Fig. 6 is an outside end view of the engine, showing the valve-governing gear. Fig. 7 shows a sectional view of the steam-inlet shaft, the piston being removed, as at *A;* a transverse sectional view of the same at *B,* and a side and end view of the steam-inlet sleeve upon which the steam-inlet shaft revolves, at *C.* A view of the cutoff valve *D,* which is rotatively journaled in the steam-tight sleeve is also shown.

The operation of this engine is as follows: The steam enters the hub of the forward cylinder head around the sleeve *A,* Fig. 4, through the steam-inlet apertures *B,* into the interior of the rotary valve, from which the steam is discharged through the ports *D D,* Fig. 5, to the port *E* and *F,* of the shaft, Fig. 4, as it rotates. The steam then passes into the cylinder and pushes against the piston arms *G* of the pistons shown in Fig. 5. This causes the two pistons to revolve in their cylinders, the meshing of the gear teeth of the piston causing them to move in unison. The arms of the inlet ports of the valve and sleeve are such, relative to the oppositely proportioned parts of the shaft, that the upper pistons and their shafts have no dead centers when the valve is set to cut off at 70 per cent. of the piston's rotative stroke, as one or the other of the parts of the sleeve and valve are open to the ports of the shaft at all times.

The point of cutoff is controlled by means of an index gear wheel *H,* Fig. 6, which is secured to the outer end of the valve stem. An index plate *J* is secured to the front of the cylinder head. A slidable tooth bar *K* meshes with the gear *H.* The upper end of the sliding bar *K* is attached to a governor which automatically moves the sliding bar up and down as the speed of the engine increases or decreases. This movement is transmitted to the valve, giving it a semi-rotary motion, and in so doing determines the point of cutoff. This cutoff varies from 10 to 70 per cent. of the operative stroke of the engine.

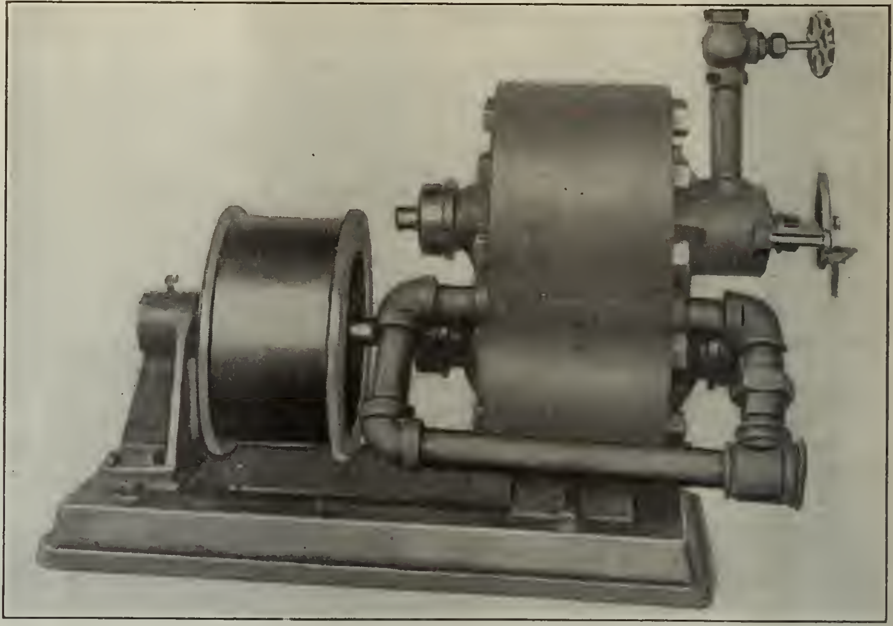


Fig. 3. Knowles Rotary Engine

Schmidt's Rotary Engine

A patent for a rotary steam engine has been granted to Rudolph F. Schmidt, 607 East Third street, Cincinnati, O. An end view of the engine, with the cover removed, is shown in Fig. 8. A long-hinged abutment which provides for swift and easy passage of the piston head under the abutment is made hollow on top in order to reduce the weight. The piston wheel is mounted on the main shaft and fills the center of the cylinder, leaving a shallow steam channel surrounding the wheel. The piston head is made in two sections, interlocked and curved on both ends, which allows a back and forward movement under the abutment without the aid of cams and levers.

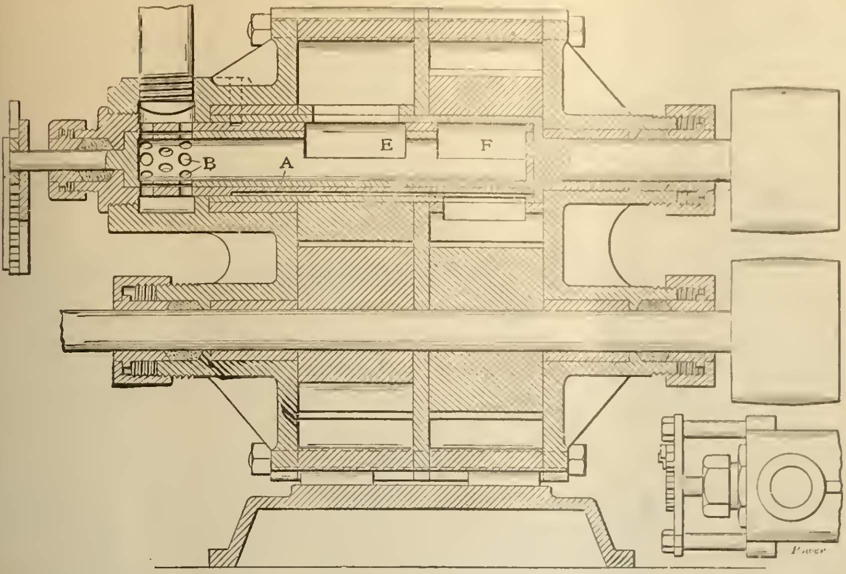


Fig. 4. Vertical Longitudinal Sectional View of Knowles Rotary Engine

Split packing rings, which are not supposed to give at high pressure, or cause great friction on the sides of the piston wheel, come level with the rim and partly extend into grooves in the sides of the cylinder, into which they are forced to prevent leakage, and partly into corresponding recesses of the piston wheel, thereby protecting the abutment from wear while the wheel is in motion.

An end spring plate facing against the abutment the full depth of the channel prevents entering steam from passing to the exhaust opening, and a safety spring plate above the abutment counteracts upon it to prevent injury to the engine. The exhaust is always open, thus providing a clear outlet. The drain pipe at the bottom of the casing is to drain the cylinder from condensation before starting the engine.

Fig. 9 illustrates a front sectional view of the cylinder which is composed of three parts: the main body, the cover or cylinder head and the base. The piston shows recesses on both sides of the wheel cover and side of casing, which are for packing rings. Interlocked gearings with set screws on a line with the packing ring and arranged for adjusting the ring evenly into the recesses of the piston wheel are also shown.

Fig. 10 illustrates the cover or cylinder head, showing the levers which counterbalance the weight of the abutment and thus lessens the strain and jar due to the swift contact with the cam. The eccentric for operating the steam valve, and special pin for the bell-crank levers are also shown.

The hinged lever or shifting device is for raising the abutment out of contact with the piston head while it is in motion, which is essential when reversing the engine or the position of the cam wheel.

Each cylinder is equipped as a complete engine, but to eliminate a dead-center position with cutoff at less than one-half stroke, three cylinders are used; with cutoff at one-third stroke, four cylinders, and for full expansion of high-pressure steam, five cylinders are used.

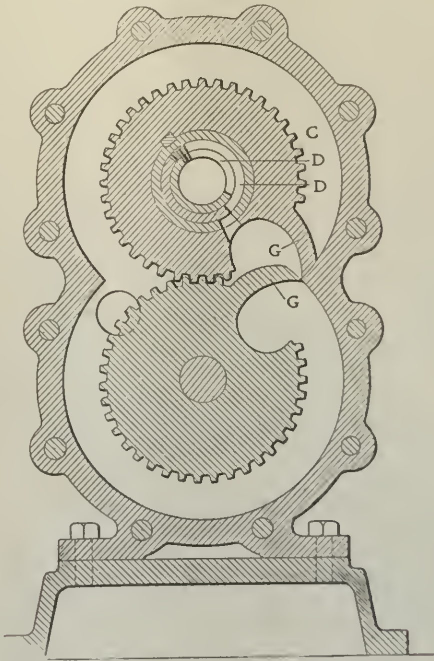


Fig. 5. Transverse Sectional View Knowles Engine

The long abutment makes a swift passage of the piston head possible and prevents the steam from passing to the exhaust outlet. The piston head is fastened to the piston wheel, which is keyed to the main shaft upon which, on the outside of the casing, are cam wheel and levers which are attached to the pin of the abutment for the purpose of raising the abutment in immediate advance of the piston head and thus prevent violent contact. There is also an eccentric to operate the valve. The valve does not connect with the exhaust outlet, and allows a cutoff of steam at any point of piston travel, thus utilizing the expansion of the steam. The entering steam, encountering the resistance of the abutment, advances the piston head, and after cutoff the steam is released when the piston passes under the exhaust opening.

A single-cylinder engine of this class requires a flywheel to bring the piston head, when stopped, to a port opening, but with a duplication of cylinders on one shaft, one piston head on each, and each piston wheel arranged to preserve balance and take steam, the dead center is eliminated.

Large port openings, in porportion to the piston surface, are provided, which allows a quick response to the load.

Englund Rotary Engine

According to *The Engineers' List,* this engine is the invention of Albert O. Englund, Wayne, Kan. In its general features, the engine comprises a fixed annular cylinder having a piston moving therein, which projects outward from a central shaft. Within the cylinder are one or more gates which, when closed, form abutments, or cylinder heads, and which, when open, permit the passage of the rotary piston and then close.

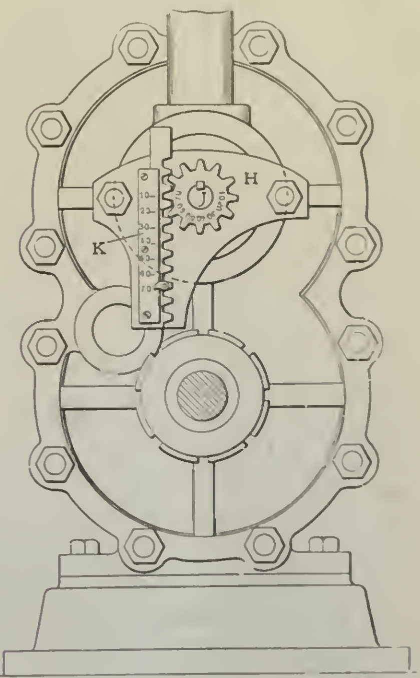


Fig. 6. End View of Knowles Engine

As will be seen from Fig. 11, the cylindrical cylinder shell is mounted on a base provided with suitable bearings for a central rotary shaft. Projecting outward from the shaft is a collar which carries the piston disk.

The cylinder is slotted on its inside circumference, and the collar on which the disk is mounted fills this slot while suitable packing rings prevent the passage of steam out through them.

At diametrically opposite points on the cylinder are located gate chambers, containing the gates.

Communicating with the gate chambers on either side of the gate are steam-inlet and exhaust valves. These are operated by rock shafts, see Fig. 12, which are actuated by inwardly projecting curved arms, having antifriction rollers at their ends. These are engaged by cams mounted to rotate with the central main shaft, but so mounted that they may be shifted with relation to each other to control the time of opening and closing the inlet and exhaust ports. The lift of these cams, and the initial opening of the valves are accomplished by means of a hand lever, engaging with a movable sleeve extending across the cylinder. These gates each consist of two semi-disks, each half-disk being mounted on the end of a curved arm, which in turn is mounted on and turns with a vertical rock shaft. The two rock shafts which together actuate the two halves of the gate are geared by sectors to move together though in opposite directions. Projecting rearward from these rock shafts are arms which in turn are connected by a system of levers and rocking bars which are acted upon by a cam rotating with the central shaft. As the shaft and cam rotate under the action of the circularly moving pistons, the gates on one side or the other are opened to allow the passage of the piston, or closed after it to form the cylinder head.

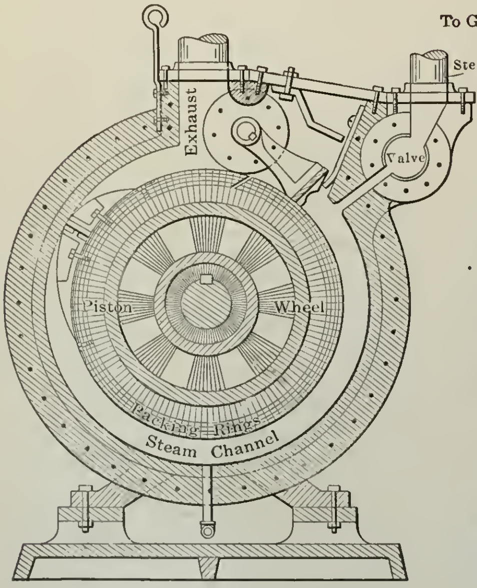


Fig. 8. Sectional View of Schmidt's Engine

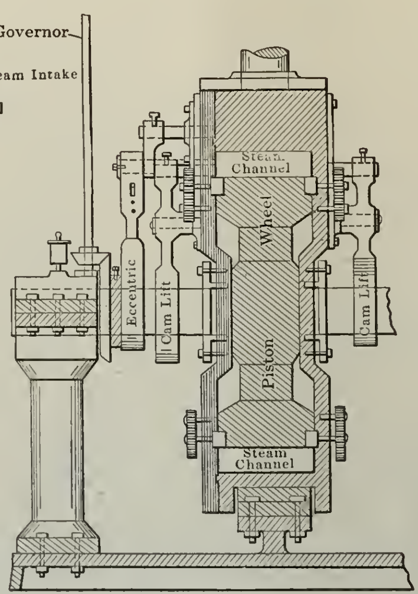


Fig. 9. Front Sectional View

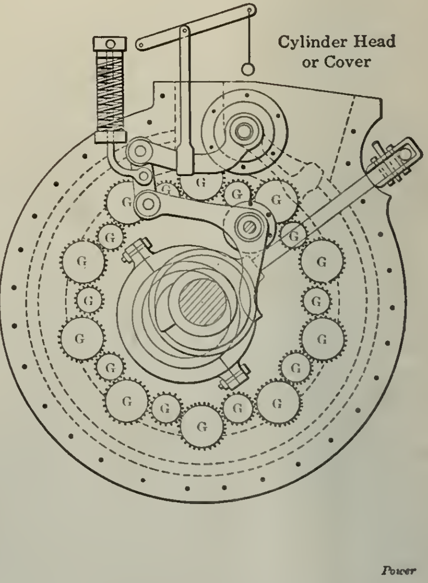


Fig. 10. Exterior View of Cylinder Head

Mounted on this central shaft is a pulley which, by means of a belt, drives a governor for controlling the inlet of steam to the cylinder.

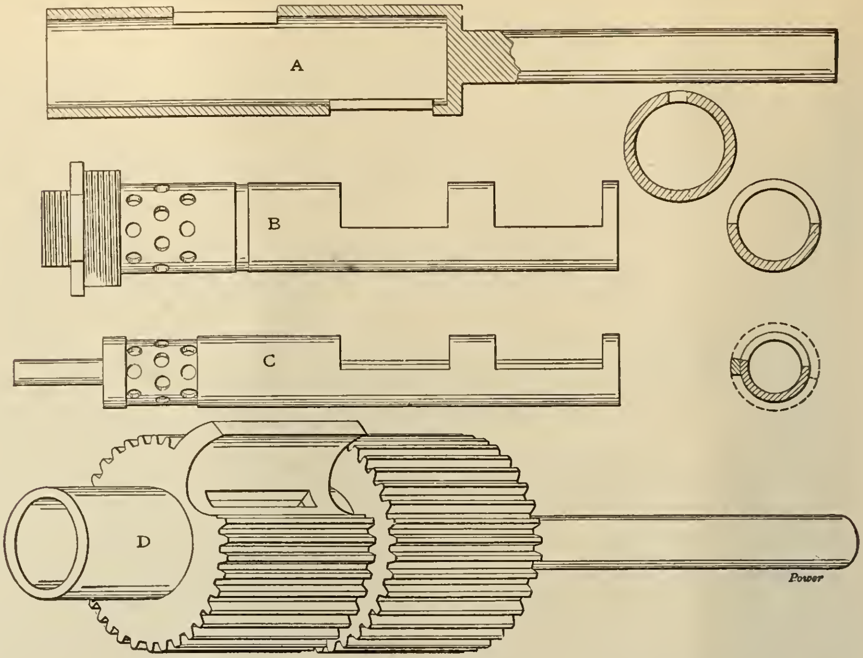


Fig. 7. Details of Valve and Rotor, Knowles Engine

When steam is admitted to the cylinder, it passes into one of the valve chambers and through one of the partly opened gates, the other gate being closed. The steam impinges on the rear face of the piston driving it forward. As the piston travels forward, the gate behind it closes, and an inlet valve opens just forward of the closed gate, while the exhaust valve of the gate ahead opens. This valve closes just before the piston reaches it, and the second or right-hand gate opens. After the piston passes the open right-hand gate and the inlet port of that gate is opened, the right-hand gate closes. After the piston has passed the right-hand gate and the inlet behind it has opened, the left-hand gate closes and the left-hand exhaust is opened before the moving piston.

In the first half of the revolution, the left-hand gate closes shortly before the right-hand gate opens and in the second half the right-hand gate closes shortly before the left-hand gate opens; thus each gate closes with pressure on both sides. While the gates are opening they are not under pressure.

If the load on the engine does not require the piston to operate under a direct head of steam during its whole travel, the engineer moves the cam-adjusting lever rearward, and thereby holds the inlet port valves closed for a longer period than when the steam is not being used expansively. All the valves are of rotary type and are actuated by cams. The two sides of the gate move in gate chambers or slideways which hold them rigidly in alinement with each other.

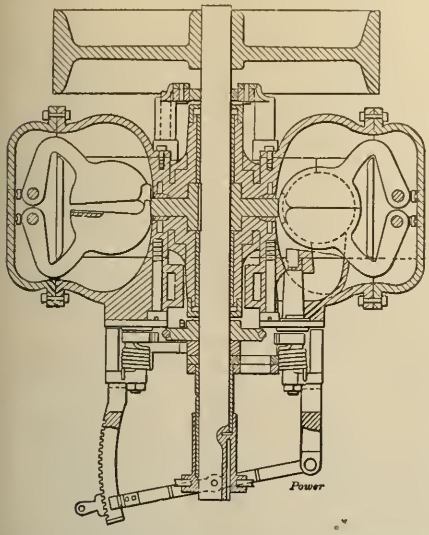


FIG. 11. SECTIONAL VIEW OF ENGLUND ROTARY ENGINE

MOTSINGER ROTARY ENGINE

This engine is shown in Fig. 13 direct connected to a small direct-current generator. The details of construction are shown in the accompanying illustrations.

The engine consists of three cast-iron rotors, as shown in Fig. 14. The center rotor is made with a round groove, and the two outer rotors each have a projecting tooth with a round face, which fits into the groove of the center rotor. These rotors are connected by means of gear wheels which are secured to one end of their shafts and cause the middle rotor to turn at such a speed as to permit the tooth of each outer rotor to fit in the groove. By setting the rotors 180 degrees apart, the vanes of both outside rotors may alternately use the groove in the middle rotor.

These rotors are incased in a cylinder having three bores whose centers are in the same horizontal plane. The radii of the outside bores are as much greater than the radius of the middle bore as the hight of the vane; the outside bores cut into the middle bore a depth equal to the difference of their radii. Placed in proper bearings of the heads covering the ends of these bores, the three rotors lie in contact with each other their entire length, forming two steam-tight chambers swept by the rotating vanes on each side of the center rotor. Both heads are cast from the same pattern and their steam chests each contain three chambers communicating with the ports that lead to and from the engines.

Fig. 14 shows the arrangement of the three rotors, also steam ports.

The tops of the rotor vanes are made steam tight by fitting T-shaped packing strips into T-shaped grooves, allowing a little space at the bottom for movement and wear. Small coil springs inserted beneath these strips and the centrifugal force of rotation hold them tight against the surface of the cylinder. The middle rotor is packed in the same way by the use of four packing strips set 90 degrees apart so that there are always two of them (one above and one below), doing service in preventing steam from passing from one engine to the other. The ends of the cylinders can be similarly packed, but as the ends of the rotors fit snugly against the heads, and there is so little wear, it is said that there is no need for packing. To take up the wear of the bearings and prevent any possible leakage between the rotors into the exhaust opening, a spring gib is used against a split bushing. It is adjusted and held tight by a set screw and jam nut against each bearing of the outside rotors.

Since both heads are alike and face each other on the engine, the positions of the ports *A* and *B*, Fig. 14, will be reversed, and when the head, shown removed, is put in place, if one could look through the engine toward the flywheel, the far port for the left-hand engine would be exactly opposite the upper port *X*. The left-hand rotor is shown in position to take steam at the far end of the cylinder, and when it has made almost a complete turn will exhaust through port *B* of the near head below the plane of the centers of all the rotors. Looking through the right-hand engine one would see the far port opposite lower *X*, and the right-hand rotor would be shown as having gone half a turn since taking steam and would exhaust through the near port *A*, above the plane of centers of all the rotors.

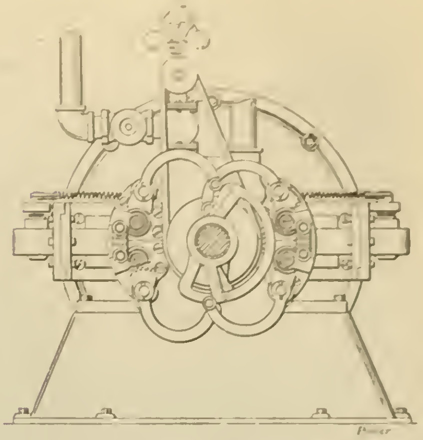


FIG. 12. END VIEW OF ENGLUND ROTARY ENGINE

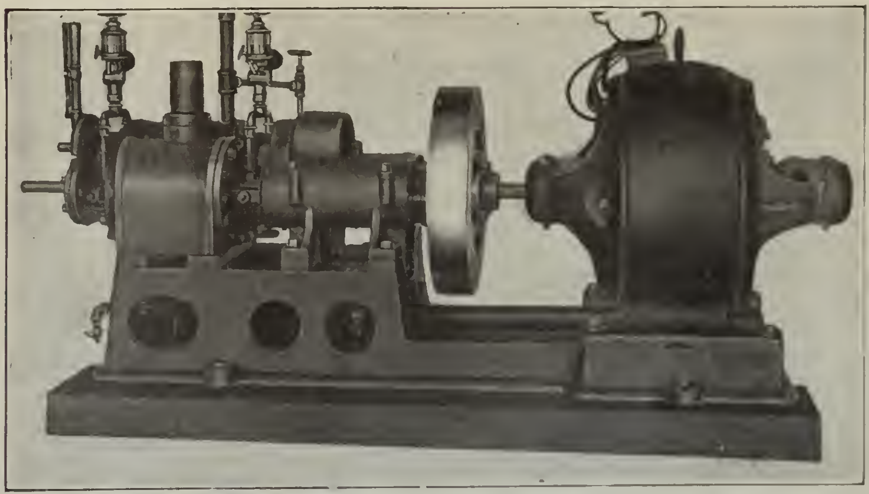


FIG. 13. MOTSINGER ROTARY ENGINE

Fig. 15 is a sectional view through the center line of the engine, showing the design of the valve and the passage for the steam to the exhaust.

The governing device, Fig. 16, consists of a central stud on which a gear revolves, driven from a second gear secured to the shaft of the center rotor. To the top gear are hung two weight arms, which are kept in position by means of a spring fitted to each. To the pivoted end of each weight arm is secured a segment and gear wheel which mesh with a gear fitted to the valve stem of the engine. As the speed of the engine increases or decreases the governor weights are moved in or out, and the gear segments, moving with the weight arms, turn the valve which cuts off the steam, earlier or later, as the speed of the engine dictates.

Steam enters the engine and passes to the port that registers with the two middle ports of the reverse valve shown at *A*, Fig. 15. Directly opposite and below in the live-steam port is the exhaust port. It also registers with the middle port of the reverse valve. The middle ports of the reverse valve are separated by a partition which makes the port on one side communicate with one head, and the port on the opposite side communicate with the other head, so that when the port is open for live steam to enter one end of the engine, the other port is open to receive the exhaust from the other end. The inlet and exhaust ports are on the same side of the valve but at opposite ends. The exhaust ports always open both behind and in front of the cutoff plate, making a free exhaust possible.

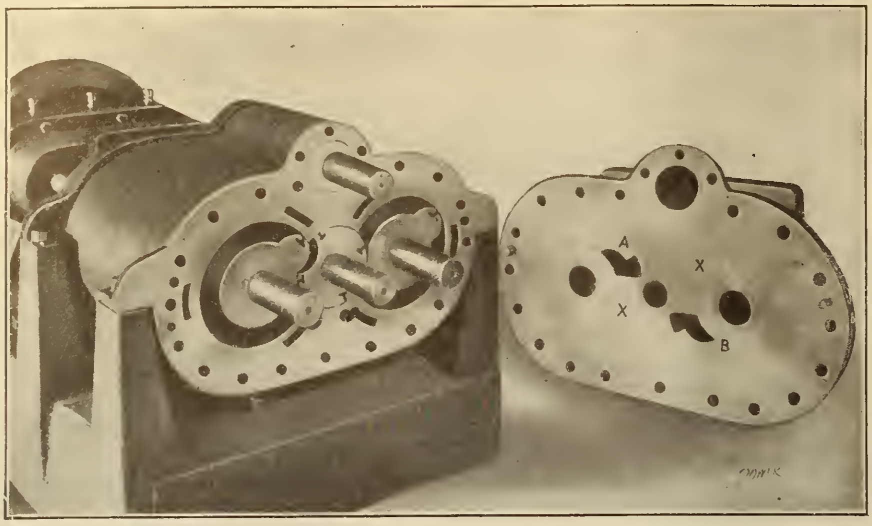


Fig. 14. View of Rotors and Ports in Casing, Also Ports in Head

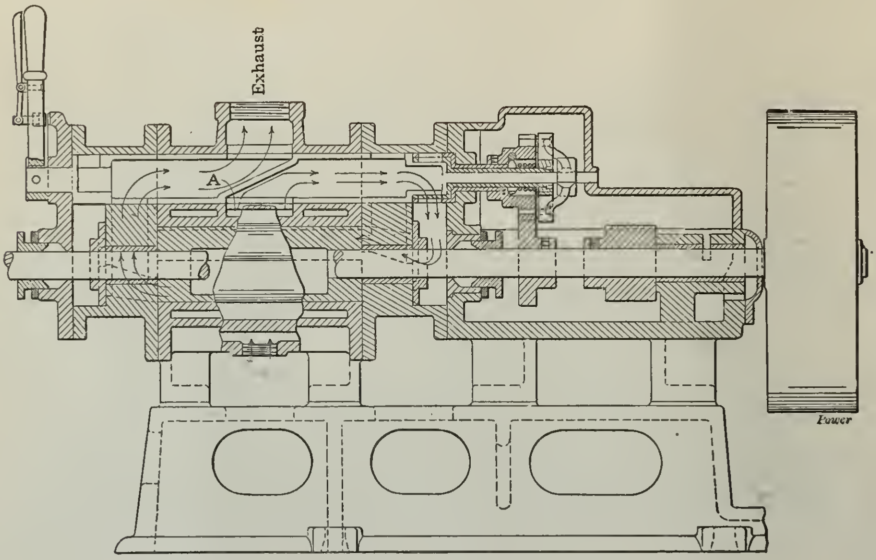


Fig. 15. Sectional View of Motsinger Rotary Engine

At the near end of the valve, Fig. 14, are shown three inlet ports. The largest port opens into the steam chest in front of a cutoff plate, and steam coming through it is fed to each engine alternately by the port in the revolving cutoff plate. The two slot-like ports open behind the cutoff plate to both engines where there are no dead points. The ports in the heads being on opposite sides, the engine is reversed by a half turn of the valve handle. When the reverse handle is perpendicular all steam is shut off both engines, but if moved either way to a point within 15 degrees of the horizontal, the two ports behind the plate will open to both engines where there are no dead points and the engine will start with any load it can pull. Pushing the handle completely home to its resting lug, the ports beneath the plates will be closed and the engine will run on the "economic" cutoff plates.

The exhaust is always open in front of the steam-driven vanes. The ports in the heads, Fig. 14, are cut the shape of a stylish shoe, *A* inverted and *B* upright. The toe part of the shoe is the main exhaust for the steam behind the vanes, but the heel part opens into the groove of the middle rotor, and as this groove naturally exhausts first, this heel part of the port serves to admit steam at the earliest point desired when running one way, and to let out the last bit of water or steam contained in the groove as the vane dashes quickly by in the opposite direction.

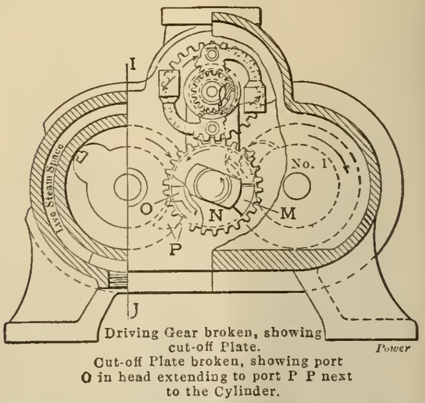

I
Live Steam Space
No. 1
O
N
M
P
J
Driving Gear broken, showing cut-off Plate.
Power
Cut-off Plate broken, showing port O in head extending to port P P next to the Cylinder.

Fig. 16. Governor and Valve Gear

This engine takes steam once every revolution, and exhausts but once. Steam pressure is maintained during almost a complete revolution.

The equal circular bodies of the rotors are cored to balance by taking out metal from the side the vanes are on and leaving in metal on the grooved side.

This engine is made by the Motsinger Rotary Engine Company, Greensburg, Penn.

The weight of the dry matter in a cubic foot of wet mud of the density to be expected at the bottom of a reservoir was determined by the engineers of the Tallahatchie drainage district, Miss., in connection with their investigations of the probable sedimentation in the reservoirs that may be constructed in the district. Two sections, 8 inches long, of 3½-inch wrought-iron pipe, were turned on a lathe to a cylinder bore inside, were milled at the ends, beveled outside at one end, and carefully measured with calipers and scale. According to the *Engineering Record,* each tube was pushed down into typical submerged mud, struck at the ends and the contents then washed out into an iron vessel, which had been first heated to drive off moisture, and then weighed. The vessel was then heated until all moisture had been driven off, and again weighed. By several such determinations it was ascertained that 1 cubic foot of wet mud, such as was tested, contains 60 pounds of dry matter, and "parts by weight" of sediment can be readily reduced to cubic yards of wet mud which would deposit in a reservoir, provided the column of water passing through the reservoir be known.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for June, 1910, *was* 139,000.

July 5................ 34,500

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

The Hydrostatic Test

There seems to be a general impression among laymen and those associated indirectly with power plants, that if a boiler shows no signs of distress under a hydrostatic pressure of a certain number of pounds, it is, therefore, capable for the next twelve months of carrying with safety a steam pressure equal to two-thirds of the hydrostatic pressure applied; or, to put it in another way, if the boiler does not rupture or show signs of distress under a water pressure of 150 pounds, it is safe to carry a steam pressure of 100 pounds. This is very illogical, and one might as well say that because a man can lift 150 pounds six inches from the floor—once, he ought to hit a blow of 100 pounds all day long for the next twelve months. We are aware that the comparison is a little overdrawn, but there are times when only a strong contrast will bring out vividly a commonplace truth.

Suppose a boiler with an ultimate bursting pressure of 400 pounds. Using a factor of safety of 5, the allowable working pressure would be 80 pounds, and 1.5 times 80 pounds would be 120 pounds, which would be the logical hydrostatic test to apply. As the elastic limit is generally a little above one-half the ultimate breaking strain, this boiler would stand a water pressure of, say, 200 pounds without stretching the metal beyond the elastic limit. Suppose such a pressure were applied, and then that pressure were divided by 1.5 to obtain the steam pressure. The result would be 133 pounds, giving a factor of safety of only 3. which, of course, is dangerously low. The boiler would appear to act about the same, whether it was subjected to a hydrostatic test of 120 or 200 pounds. In one case, the allowable pressure would be 80, and in the other, 133 pounds, such figures, of course, being meaningless until the accompanying factor of safety is determined.

Not only may the boiler be strained beyond its elastic limit by the application of the hydrostatic test, and a permanent set take place, but a boiler almost wholly void of ductility would show no signs of distress under the application of a steady hydrostatic load, whereas such a boiler might fly all to pieces when subjected to the alternating stresses produced in it when under steam.

That there is no relation between the hydrostatic test and the steam pressure to be allowed, is shown by the lack of uniformity on this question among different governing bodies. While the United States Board of Supervising Inspectors of Steam Boilers requires 1.5 times the steam pressure on ordinary boilers, it requires that the hydrostatic pressure be twice the steam pressure on all coil and pipe boilers. The American Boiler Manufacturers' Association recommends that the hydrostatic test shall not exceed the working pressure by more than one-third of itself, and this excess is limited to 100 pounds. The rules formulated by the Board of Boiler Rules of the Commonwealth of Massachusetts make the hydrostatic pressure equal to 1.5 times the maximum allowable working pressure, though twice the maximum working pressure may be applied to boilers to carry not over 25 pounds, or on pipe boilers. The British Board of Trade and Lloyds require twice the working pressure; the National Boiler Insurance Company of England 1¾ times the working pressure, and the Scottish Boiler Insurance Company, of Scotland, for pressures above 80 and under 150 pounds—the working pressure plus 80 pounds; above 150 pounds—the working pressure, plus one-half the working pressure, etc., all showing that there is little uniformity in regard to this ratio.

We do not wish to be understood as condemning the hydrostatic test; it is an excellent thing in its way, and at times very valuable. In the case of small boilers, which cannot be entered for inspection, or in specially fitted work, or in testing cast-iron boilers, or in special vessels such as jacketed kettles, whose interiors can be viewed only through a 2-inch plug hole, the hydrostatic test is practically the only available means of testing. In such cases, when that test has been applied, all that it is possible to do has been done. The hydrostatic test is also valuable in testing new boilers in the boiler shop, to determine whether or not they are tight.

In a large measure, it is unfortunate that the hydrostatic test does not give definite and absolute quantities, for if this were the case, boiler inspection, which is now a rather disagreeable job, would be made quite easy and pleasant, and instead of crawling through small holes into hot and dirty interiors, and later making many computations on the strength of each part of a boiler, the efficiency of the riveted seams, strength of the braces, resistance of the flat places, etc.; all an inspector would have to do would be to keep his eye on the gage.

Since there is no connection between the hydrostatic and the allowable steam pressure, the latter can be stipulated only after an inspector has carefully examined the boiler inside and outside, taken careful measurements, computed the strength of the weakest part, and applied to that part a factor of safety approved by the dictates of good engineering practice.

Depreciation

In estimating the cost of power, or in making an appraisal of a plant, no one item is as uncertain and as unsatisfactory as that of depreciation. Interest can easily be reckoned as that which the same capital would return if safely invested in a bank or in securities; taxes and insurance may readily be determined on the basis of local conditions; but when it comes to reckoning maintenance and depreciation, doctors disagree.

By maintenance, we mean the cost of keeping the machinery and equipment in fair working condition, irrespective of its "age, color or previous condition of servitude." This can only be determined in each type and size of plant by keeping careful records. In large central stations, the total cost of repairs on buildings and on steam and electric equipment is usually about one-eighth of the total expense of running the station.

If proper attention be not given to maintenance, depreciation increases alarmingly, but the most careful repairing will not prevent the gradual reduction of values.

Depreciation is of two kinds, direct and indirect, or perhaps it were better to say, normal and accidental. Normal depreciation represents the gradual decrease of value and of efficiency due to age and wear.

The life of an engine or boiler is generally assumed to be from twenty to twenty-five years and the same rule holds for the greater part of the auxiliary equipment. Coal and ash-handling machinery and mechanical stokers probably have a shorter term on account of their continual motion and the severe service to which they are subjected, while the natural life of a condenser or pump may be considerably greater than that of the engine. These facts would point to five per cent. a year on the initial cost as a safe amount to allow for depreciation. This may be considered the normal depreciation of a power plant. If, however, we examine the history of most installations of this character, we will find that the term of life has been much shorter than that above mentioned and that this reduction in term of service is not due to wear or corrosion but to the fact that the plant has become old fashioned or that it has insufficient capacity. The increasing demands for power in a prosperous business may justify a radical policy in discarding the old plant entirely and substituting for it new machinery in larger units and more economical in operation.

While a fifty- or seventy-five-horsepower slide-valve engine and a fire-tube boiler carrying one hundred pounds pressure may be the most economical arrangement for a small plant, an increase to two hundred of three hundred horsepower would make it desirable to put in a four-valve engine and sectional boilers capable of carrying higher pressures. Although the discarded machinery may have some value, it is not such as to discount on very much beforehand and it may mean a practical scrapping of the whole plant in ten years instead of twenty.

Aside from this question of growth, there is the other one of improved machinery. A man who installs an engine and generator today is not at all sure but that in ten years there will be such improvement in both the steam and electrical ends of the machine as to make it desirable to change. This is particularly true in central power stations where large engines, steam turbines and generators are employed. There have been a number of instances within the past ten years where units which, when installed, were the best of their kind, have been condemned and replaced by others within seven or eight years, the enormous expense being justified by the fact that the amount of steam and coal used per kilowatt-hour was reduced about one-half by the change.

In these days of close figuring and keen competition, it is not possible for a company to do business with old-fashioned, uneconomical machinery. This problem of indirect depreciation has confronted manufacturers of machinery in much the same way. The invention of the Taylor-White process of making tool steel some ten years ago and the subsequent developments in the manufacture of various alloys has revolutionized machine-shop practice. A lathe which formerly plodded along contentedly, driven by a two-inch belt and consuming perhaps one-half a horsepower, has been replaced by a motor-driven machine using ten times that amount of power. It is not at all uncommon today to see machines using from ten to twenty horsepower in cutting cast iron and steel with high-speed tools. The lathe, planer or milling machine which is in good repair and practically just as capable of doing its intended work as when new, has necessarily been discarded and replaced by a much higher-priced machine.

This same cause has produced another revolution in the boiler and engine room. The seventy-five-horsepower engine and boiler are no longer able to supply the new demand for power and have been replaced by a plant of much larger capacity. It is true that this has all been a profitable investment and is now paying a good interest, but, nevertheless, it has involved a practical abandonment of the old machinery and a sudden increase in the capital invested. Examples like this show why manufacturers and owners of power plants find it necessary to use a large depreciation factor in estimating the cost of power or the overhead charges in the machine shop. A depreciation factor of from eight to twelve per cent. is not at all unusual nor excessive. Of course, no hard and fast rule can be laid down and each owner will have to decide this question for himself; but it is apparent that in this age of close figuring, no owner of machinery can afford to neglect a careful study of this problem. Either by the maintenance of a sinking fund or close attention to immediate profits, he must make provision for these sudden changes which are liable to be necessary.

Some business men prefer to estimate depreciation as a gradually decreasing factor rather than as a definite per cent. each year. For example, if five per cent. of the original cost is assessed each year, this would mean an estimated life of twenty years for the plant. Now, it is a fact that the depreciation, as a mere question of market value, is more rapid at first; that is, it would hardly be possible to sell an engine or a boiler at the end of one year for ninety-five per cent. of its face value or at the end of two years for ninety per cent. If ten per cent. depreciation were allowed each year on the inventory value of the preceding year, instead of a straight-line depreciation of five per cent., we should have a curve which falls at first rapidly and then becomes nearly parallel to its axis. By such a method, the inventory value at the end of ten years instead of being fifty per cent. would be about thirty-five, while at the end of twenty years instead of none, we would have a value of about twelve per cent. Such a system gives probably a more correct definition of values from year to year and would be preferable in determining inventory values. As a matter of determining the cost of power or the overhead charges in the shop, the other method is probably just as reliable.

Coal which costs the least per ton often proves to be the most expensive when results are considered.

If everyone knew when to put up, pay up and shut up, how few arguments there would be.

When an engine joins the "Knockers" club, there is a reason; when an engineer does so, he seldom has any excuse.

Chimneys and Draft

By John S. Leese

Temperature at which maximum draft is secured. Formula for estimating the draft which a given chimney is capable of developing. Types of chimneys and materials of construction.

Natural Draft

The natural draft obtainable from a chimney depends upon the column of gas inside the chimney being lighter than the air outside. The fuel consumption of a furnace using natural draft is not generally more than about 20 pounds of coal per square foot of grate area per hour, but with forced draft this consumption may be increased fourfold with equally complete combustion. If a pound of coal requires 12 pounds of air for perfect combustion, the natural-draft furnace will take about 24 pounds in practice, the difference being dilution. With forced draft less diluting air is required, because of the intensification and localization of the process of combustion; then 18 pounds or less of air will suffice. Therefore, as less air is required for forced draft than for natural draft to burn the same amount of coal, the furnace temperature will be higher, the heating surfaces will be more effective and the heat lost by hot gases is reduced, providing that the heating surface is large enough to allow them to leave the boiler flues at the same temperature as they would with natural draft. When forced draft is employed there is no necessity to allow them to leave the flues at a high temperature; the draft is there already, so that the escaping gases need only be reasonably hotter than the feed-water temperature. With natural draft this heat is wanted to make the draft. In practice these theoretical advantages of forced draft are not always available as forced-draft systems are generally installed to "force" the boilers.

The most efficient boiler installation would be one with a heavy forced draft, a proportionately small grate area and a proportionately large heating surface, the latter being further developed by the use of an economizer. The waste gases would then leave the economizer as cool as possible and the important air and coal factor would be as small as practicable for good combustion. If

T = absolute temperature of the gases inside the chimney in degrees Fahrenheit, and

t = absolute temperature of the outside atmosphere in degrees Fahrenheit,

then, to get the best natural draft, the approximate relation is

$$T = 2.1\ t.$$

Taking an average of 520 degrees Fahrenheit for t, then

$$T = 2.1 \times 520 = 1092$$

degrees Fahrenheit absolute, or = 630 degrees above Fahrenheit zero.

Any temperature above or below this will be less efficient in inducing air. At this temperature a stage is reached where the gain in head and the strength of the current due to it are nullified by the diminution of density and the draft is actually reduced. It should, therefore, be remembered that there is no advantage whatever in carrying a higher waste-gas temperature than about 600 degrees Fahrenheit, and, that from the point of view of thermal efficiency, the lower the chimney temperature, the better because less heat will go to waste.

With natural draft and its most economical chimney temperature as obtained above, about one-fifth of the total energy contained in the coal will go up the chimney. If the temperature of the products of combustion leaving the boiler is more than 600 degrees Fahrenheit, it is, then, desirable to use an economizer.

The draft (d) in inches of water can be approximately obtained by the formula:

$$d = H\left(\frac{7.6}{t} - \frac{7.9}{T}\right)$$

where

H = hight of chimney above grates,

T and t are the absolute temperature as before.

If the above average values are given to T and t, we get

$$d = H\left(\frac{7.6}{520} - \frac{7.9}{1092}\right) = 0.0067$$

or

$d = 0.007\ H$ inches of water, approximately

and

$d_1 = 0.2\ H$ feet per second, approximately.

When

H = Hight of chimney above fire bars in feet,

C = Pounds of coal fired per square foot of grate area per hour,

a = Grate area in square feet,

E = Effective area of chimney in square feet,

then

$$H = \frac{a^2\,C^2}{200\,E^2} = \frac{(\textit{Total coal per hour})^2}{200\,E^2}$$

and hence:

$$E = 0.08\,\frac{a\,C}{\sqrt{H}}$$

If A equals actual area of chimney in square feet,

$$A = E + 0.7\,\sqrt{E}$$

and

$$E = A - 0.6\,\sqrt{A}$$

The capacity of a chimney varies as the square root of the hight and directly as the area.

If P equals the actual horsepower required in use, then

$$P = 3\tfrac{1}{3}\,E\,\sqrt{H}$$

$$H = \left(\frac{0.3\,P}{E}\right)^2$$

$$E = \frac{0.3\,P}{\sqrt{H}}$$

If a forced draft is obtained by means of a steam blast, as used on locomotives, the draft in inches of water is very approximately obtained with short chimneys by Longridge's formula:

$$d = \frac{37\,D^{1.662}\,p^{0.\text{[illegible]}}}{S^2}$$

where

d = draft in inches of water,

D = diameter of jet or blast pipe in inches,

p = pressure of steam in pounds per square inch,

S = diameter of chimney in inches.

With natural draft and low-grade fuel the hight of a chimney is seldom less than 70 feet, and the ordinary hight of a mill chimney is about 200 feet. If more boilers are being added to a battery already installed and connected to the same chimney the hight should be increased in a greater ratio than the area.

Types of Chimneys

The wind pressure in designing chimneys may be calculated at 56 pounds per square inch. If unity is taken for a square chimney the relative pressure on a hexagonal chimney will be 0.75, for an octagonal chimney 0.65 and for a round chimney 0.5. Hence, a round chimney is preferable to a square one in outside section, because it has less windage, and as regards inside section, because the friction and eddy-current effects are less.

The cheapest chimney is the one built up of steel plates, riveted together and unlined with brick. These should be suitably anchored by guy-stays. The self-supporting steel chimney is bolted to a foundation which is massive enough to withstand windage stresses. The durability of steel chimneys depends on the thickness of the plates, the kind of fuel used, the atmospheric conditions, the condition and state of maintenance. Steel chimneys should be painted frequently with the best quality of paint in order to stop and prevent corrosion. Next in order of cost comes the steel chimney partly lined with brick. Hard selected-brick chimneys, double-walled, are the next in cost. These should have an internal core such that it is free to expand without straining the outer supporting core. "Custodis" chimneys are becoming very popular. They are built without a core of specially molded perforated bricks and are more costly to erect than other types, but it is claimed that the maintenance is very low. Good bricks should give a metallic "clink" when knocked together. The absorption of water by bricks should not be more than one-sixth their weight, and no bricks with cracks, flaws or lumps, or which are not true in shape, should be used in chimney construction. Some facing bricks will harden on exposure. Brickwork weighs about 100 to 130 pounds per cubic foot. The total weight of a chimney may be approximately arrived at by the following:

For square chimneys, *Weight* = 56 × *breadth of base* × *area exposed.*

For octagonal chimneys, *Weight* = 35 × *breadth of base* × *area exposed.*

For round chimneys, *Weight* = 28 × *breadth of base* × *area exposed.*

The width of the base of a square chimney should be at least one-tenth the hight and for other shapes not less than one-twelfth. The batter should in no case be less than 0.25 inch per foot of hight. The thickness of the brickwork at the top of a chimney should be at least 8½ inches, or one brick. This thickness may extend 25 feet down. From 25 feet from the top to 50 feet from the top the thickness should be 1½ bricks, increasing ½ brick every 25 feet down. If the inside diameter of the top of the chimney is more than 4 feet 6 inches the top length should be at least 1½ bricks thick and so on.

The construction of chimney foundations is most important. After putting in the base, the concrete and brickwork must be allowed to settle before the stack is built. All brick chimneys should have cast-iron caps, well clamped to the brickwork. Copings should not be heavy or large to catch the wind and must be bound securely together. For this purpose copper cramps are preferable to slate dowels. The center of gravity of the coping in section must fall within the outer line of the shaft. In conclusion, all chimneys should be fitted with a substantial lightning conductor.

Questions for Hoisting Engineers

These questions were used in an examination of applicants for certificates of competency to serve as hoisting engineers in the State of Indiana and are published to give the reader an idea of the attainments required.

Following is a list of printed questions used at an examination for hoisting engineers held at Terre Haute, Ind., May 10 and 11:

Answers are not particularly desired to these questions, but a discussion of any or all of them as to their fitness for the purpose and their completeness will be welcome.

It should be understood that an examination is not held for the purpose of asking questions which the candidate cannot answer, but to find those which he can. He is expected to demonstrate his fitness to care for and operate the machinery to be placed in his hands, and this may be as clearly shown by intelligent answers to simple questions as by halting solutions which just "get by" the more complex ones.

1. Name the different capacities in which you have been employed in or about coal mines, giving the length of time you were employed in each capacity and the different localities where you have been employed.

2. (a) What are the duties of a hoisting engineer, as required by law? (b) What duties are incumbent on an engineer other than those required by law?

3. Steam boilers, water-fed. (a) Explain how it is possible to pump water into a boiler by means of its own steam pressure. (b) What prevents the water from running back after the stroke of the feed pump is finished?

4. (a) What is the best rule for finding how much pressure a boiler will actually stand? (b) What is the factor of safety usually allowed between working pressure and bursting strength?

5. (a) A return-tubular boiler is 72 inches by 18 feet long. What is the bursting pressure? (b) What would be a safe pressure? (c) What factor of safety is used when new?

6. (a) A pair of hoisting engines have cylinders thirty inches in diameter and sixty-inch stroke. (b) With a steam pressure of ninety pounds per square inch, how many revolutions per minute must the engine make to develop 1000 horsepower?

7. What is lap? What is lead? What is mean effective pressure?

8. (a) What is the cause of water foaming? (b) How can foaming in a boiler be detected?

9. How should the slide valve and pistons of an engine be tested to determine whether they leak?

10. What may be done to remedy the following faults: (a) A too early cutoff. (b) A too late cutoff.

11. State fully the various points to be considered in the arrangement of the tubes in a return-tubular boiler.

12. A weight of 5000 pounds is lifted by a hoisting engine to a hight of 400 feet. What is the theoretical quantity of heat consumed in the operation?

13. What three elements enter into each calculation relating to the safety valve?

14. A boiler 48 inches in diameter is subjected to a pressure of 100 pounds per square inch.

If the boiler plate is three-eighths of an inch thick, what is the stress per square inch of section along the sides?

15. What different kinds of stress or strain are placed upon the different materials used in a hoisting outfit, including the boilers?

16. (a) What is the best way to start a jet to draw water 185 feet vertically? (b) Explain how a duplex steam pump works.

17. If the forward eccentric rod were to break, could the backward eccentric rod be used to run the engine forward? If so, how?

18. (a) Does the acting eccentric lead or follow the crank when the link is in full gear? (b) In what position would you place the link motion when the slide valve is to be set? (c) What determines the length of the eccentric rod when setting the slide valve?

19. (a) Why should the temperature of the boiler furnace be kept as high as possible? (b) What causes the reduction of this temperature? (c) What care must be exercised in burning bituminous coals containing hydrocarbons?

20. (a) Name and define the principal parts of the machinery used in and around coal mines in this State.

21. How should a pump or an injector be worked when feeding more than one boiler?

22. (a) What effect does increasing the lead have on various operations of the valve? (b) When the diameter of a pinion or gear is known, also the pitch of the teeth, what is the rule for determining the number of teeth?

23. (a) State the three relations between force and motion. (b) What is the allowable working load for a steel wire rope 5¼ inches in circumference.

24. Why is a cylinder counterbored?

25. Show clearly the difference between a throttling governor and the automatic governor.

Inquiries of General Interest

The First Locomotive

My friend claims that Stephenson invented the locomotive. I claim that a man named Murdoch built a model before Stephenson did. Please decide the matter.

R. W.

The idea of applying steam to locomotion was probably first suggested by Robinson, in 1759. In 1784 Watt patented a locomotive; but he did not build it. About the same time Murdoch, who was Watt's assistant, made an efficient working model. Trevithick and Vivian built a locomotive about the year 1804, which traveled at five miles an hour and carried a net load of about ten tons. Blackett and Hedley discovered, in 1813, that the necessary tractive power could be obtained without the use of rack and pinion. Stephenson's "Rocket" was not built until about 1829; but it had a blast pipe and a tubular boiler, so that it may be fairly regarded as the parent of the lomocotive of today. See Rankine's "Steam Engine."

Gas Engine Temperatures

Is it true that the gases in a gas-engine cylinder are not at the same temperature throughout, and if so, why?

K. L. D.

It is, especially in large cylinders; because gases are poor heat conductors, and the heat in the gases which are in the center of the cylinder cannot get to the walls as fast as heat is taken from the gases there by the jacket water.

Flame Propagation

What is meant by "flame propagation?"

S. N. C.

The spread of flame through combustible gases. If a combustible mixture of gas and air were inclosed in a tank and ignited at one point, the flame would be communicated from the burning gases to those immediately next to them; when these kindled, the flame would spread to the next unlighted gases, and so on throughout the whole tankful of mixture. That is flame propagation. It occurs so rapidly that it seems instantaneous.

Sensible and Latent Heat

What is the difference between sensible and latent heat?

J. M. C.

Sensible heat is that which may be measured by the thermometer. Latent heat does not affect the thermometer. It is the heat that is absorbed by a body when passing from a solid to a liquid or from a liquid to a gaseous state without change of temperature.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Pump Discharge

A pump sends water through 1500 feet of 2-inch pipe, and then raises it into a tank, the water in which is 24 feet higher than the feed pipe. There are two inlets to the tank, one at the bottom and one rising over the side of the tank and passing in at the top. Which of the discharges will be easier on the pump?

N. W.

If the upper pipe dips into the water in the tank, there will be no difference. If it does not dip into the water in the tank, but empties a short distance above it, then the bottom discharge will be the easier.

Pitch of Propellers

What is the pitch of a screw propeller and how is it found?

J. R.

The pitch of a propeller is the degree of spirality or the distance it would advance in one revolution if the water in which it turned was solid and there was no slip. It is measured by drawing a line representing the shaft and at right angles to it another in its length representing the circumference of the circle made by the tips of the blades. Through the intersection of these lines a diagonal line is drawn at an angle equal to that which the outer edge of the blade makes with the shaft. The perpendicular distance from the end of the line representing the circumference to the diagonal line is the pitch in the scale to which the lines are drawn.

Bubbling Transformers

Two oil-cooled transformers hum excessively and throw oil out of the cases when the load is greatest. How can this be remedied?

B. R. P.

Try reducing the load on the transformers. Both the loud humming and oil bubbling are due to overheating. This may be caused by overloading, or the windings may be "leaky," permitting excessive local currents to flow, but the latter is not probable.

Chimney Dimensions

Will you tell me how to get the hight of a chimney without using a transit?

C. E.

It may be determined by comparing the length of its shadow with the length of the shadow of any object whose hight is known. Suppose the length of the shadow of the chimney to be 94 feet and the shadow cast by a board 12 feet long standing vertically is 8 feet long, the hight of the chimney will be to the hight of the board as the length of the chimney shadow is to the length of the shadow of the board, or 141 feet. Stated in the form of a proportion the solution reads

Lap and Lead

Is it necessary to have lap and lead on a slide-valve engine?

S. R. B.

That depends on the definition given to the word necessary. Lap is given to a valve to enable steam to be cut off before the end of the piston stroke and allow expansion to take place in the cylinder and thus reduce the quantity needed for the work. Lap reduces steam consumption within certain limits and is necessary if economy is desired.

Lead provides for full steam-chest pressure on the piston at the beginning of the stroke and is considered necessary by the majority of engineers.

Low Voltage and Hot Field Windings

A 2200-volt alternator will not keep its voltage quite up to normal at full load, although the field is so heavily excited that the magnet coils run hot. The motors on the circuit run too slow during the peak load. What is the trouble?

B. R. P.

The machine is not up to its rated speed; this causes the drop in motor speed, and helps to pull the voltage down; the low power factor of your load also pulls down the voltage, in spite of the excessive current delivered to the field winding by the exciter. Your engine regulation is also probably poor.

Strain on Staybolts

Please work an example for me by the rule for finding the strain on staybolts, given in POWER for February. Suppose that the bolts are four inches apart, that the gage pressure is 140 pounds.

C. R.

Following the rule, 4 × 4 = 16, and 16 × 140 = 2240 pounds. The stress is therefore 2240 pounds, or a little over a ton.

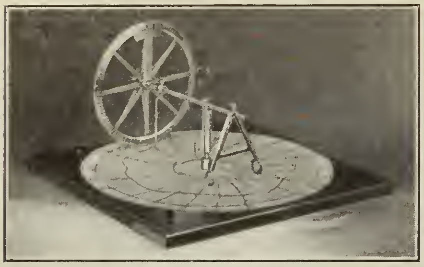
New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

An Averaging Instrument for Circular Charts

The Bristol-Durand radii averaging instrument for circular-chart records from recording meters is shown in the accompanying illustration. This instrument was devised by Prof. W. F. Durand, of the Stanford University, and is manufactured by the Bristol Company, Waterbury, Conn. The instrument is used for determining the average reading for a given circular-chart record, such as is drawn by recording pyrometers, thermometers, pressure gages, voltmeters and similar instruments.

The simple construction of the instrument is shown by the illustration. A wooden base with a metal socket is provided for supporting and centering the chart. The socket holds a rotatable pin which has a vertical slot at the top to receive the bar which carries the tracer point and triangular support. The vertical groove in the pin allows the integrating wheel to roll on the chart with uniform pressure due to its own weight.



Instrument for Averaging Circular Chart Records

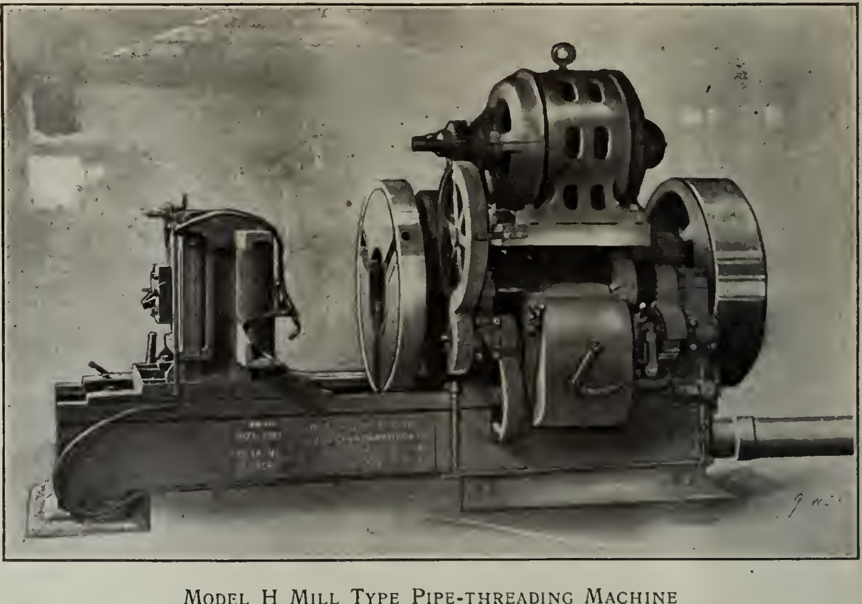
The integrating wheel is six inches in diameter; the rim is graduated into one hundred equal spaces, and is fitted with a vernier, which makes it possible to read easily with the naked eye to one-tenth of one division on the wheel. The wheel is of such large size it is not necessary to supply any counting device for the number of revolutions. The number of complete revolutions cannot be more than two, even for a record of maximum size on the large 12-inch charts.

To operate the instrument the thumb and forefinger of one hand are applied to the base of the triangular support, which is moved radially, so as to cause the tracer point continually to follow the record curve; the chart is turned with the other hand.

The principle upon which the device depends is that sliding of the horizontal shaft back and forth in the slot of the central post does not produce any rotation of the wheel. The amount of rotation depends upon the length of all the elementary circular arcs of which the irregular curve may be considered to be composed. As the length of a circular arc within a given angle is in direct proportion to its radius, the measurement of the sum of all these elementary arcs is in direct proportion to the average radius, which is the dimension desired. The instrument is equally applicable to charts having either straight radii, or circular-arc radii.

New Motor-Driven Pipe-Threading Machine

The motor-driven pipe-threading machine shown by the accompanying picture is one of several recently installed by the Stoever Foundry and Manufacturing Company, of Myerstown, Penn., at the pipe mill of the Spang-Chalfant Company, Sharpsburg, Penn. The capacity of the machine is from 4 to 12 inches inclusive. The installation is of especial interest in that this threader is the fastest machine for threading or cutting pipe on the market. It will thread pipe at the rate of 28 feet per minute, a very fast speed for a tool of this size.



Model H Mill Type Pipe-threading Machine

The machine is driven by a Westinghouse squirrel-cage induction mill motor, which has the rugged construction and the general reliability required for such service. The normal rating of the motor is 10 horsepower, and the full-load speed is 875 revolutions per minute on a three-phase 25-cycle 200-volt circuit.

The pipe-threading machine has ten speeds; the gear box gives five changes, and a double train of gears outside of the box give two speeds for each one in the box. All of the gears are made of carbon-steel castings and are fitted with bronze bushings. The gears that are subjected to the most wear have case-hardened teeth. All the gears run in oil. A number of changes in speed can be made without stopping the machine. The machine can be started or stopped by means of a clutch on the operating side of the tool. The gearing is on the outside of the machine and accessible to the operator. An internal driving gear placed in close to the body of the machine is used on the rear of the machine.

The bed of the machine is so designed and constructed as to give a very stiff and strong cross-section. The bed is supported on the outer ends by suitable legs, and under the headstock by a very heavy pedestal base that strongly supports the entire driving mechanism of the machine.

The headstock of the machine is made in one casting, similar to that of a lathe; this one casting contains all the principal bearings, which insures permanent alinement and great rigidity. The spindle or arbor of the machine is of such a diameter on the inside that the largest-sized fittings within the capacity of the machine can be passed through.

There are provided two gripping chucks of unusually heavy design, made in one piece without any separate parts or faceplates. Each chuck has three independent jaws operated by powerful screws. These jaws are made of steel and are equipped with hardened-steel plates for holding the pipe.

The die head is of the sliding or floating type that allows for any eccentricity in the pipe. The die head can be pushed to one side and the pipe cut off at a distance of 3 inches from the face of the front gripping chuck. This type of head permits inserting the pipe from the front or rear of the machine without injury to the dies. Rounding blocks are provided for rounding the pipe before it is threaded.

The adjusting mechanism is maintained in a straight line to prevent the digging of the chasers into the pipe. This mechanism consists of but two principal parts: a lever for opening and closing the dies, and a screw for making the adjustments to gage. The chasers are single pieces of steel without any pins or links. They are made by a special process so that any single chaser in a set can be replaced without reference to the rest of the set.

The oil pump on the machine is of the geared type and is driven direct from the constant-speed shaft. Valves are placed at both the die head and the cutting-off tool to regulate the supply of oil at these points. All surplus oil flows back into the reservoir in the bed of the machine through a relief valve.

All of the bearings of the pipe threader are provided with long and deep receptacles for holding lubricating wool and oil so that the machine is operated at all times with perfect lubrication.

Lisk's Universal Saddle

These saddles are made of cast iron, in various sizes, for supporting tanks, feed-water heaters, internally fired boilers, etc. They are light in weight, and can be used for different tank diameters by changing the length of the crossbars. The bearing pads are adjustable, so that they will conform to the curvature of the tank body.

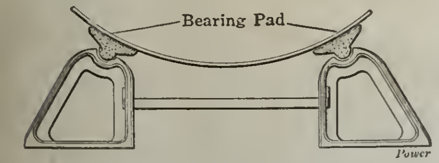


FIG. 1. TYPE A SADDLE AND FRAME

Fig. 1 shows the design of what is known as "Type A," which consists of a set of saddles made up of four pads and frames, four bases and two crossbars. It is used as shown in Fig. 3.

Fig. 2 shows the "Type B." This type of saddle is for use when it is desired to elevate the tank, heater, etc., some distance above the floor.

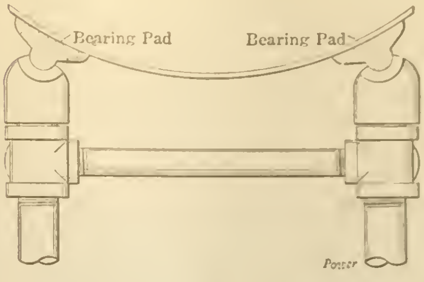


FIG. 2. TYPE B SADDLE AND FRAME

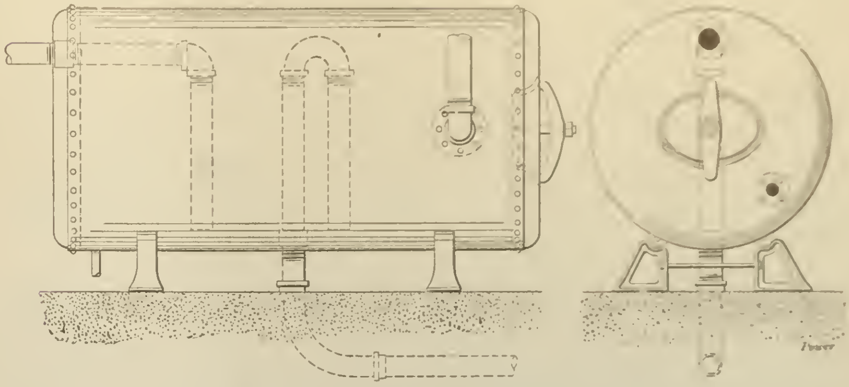
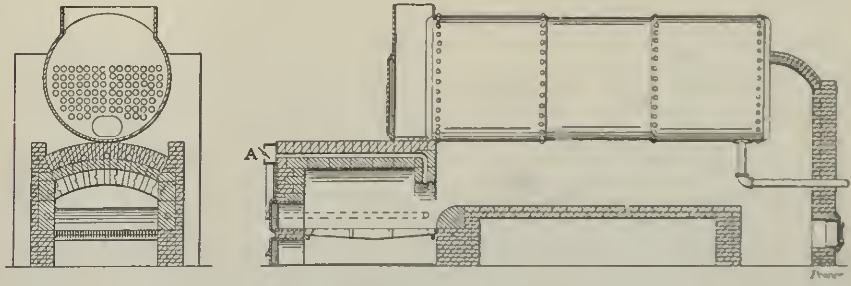


FIG. 3. TANK RESTING ON SADDLES AND FRAME

These saddles are made by J. P. Lisk, 151 Quincy street, Brooklyn, N. Y.

The Teller Air Burning Furnace

This furnace is designed to accomplish efficient and smokeless burning of fuels containing high percentages of volatile matter, such as are found in the Middle West. It is constructed on the Dutch-oven principle. The illustration shows it applied to a horizontal return-tubular boiler, but it is also being used in connection with Scotch and water-tube types of boiler.



TELLER'S AIR-BURNING FURNACE

The furnace proper consists of a double arch of specially designed blocks of steel mixture, reinforced on the outside with a heavy wall of ordinary brick and a supplementary arch above, through which air ducts are provided for admitting air to the fire, as shown. The air ducts are placed at a point that will permit the air to mingle with the products of distillation just as they are passing from the furnace, enabling them to thoroughly mix and burn before coming in contact with the heating surface.

The inverted firebrick bridgewall is a characteristic of the design and assists in mixing the products of combustion. An air damper is arranged, as shown at A, at a convenient point for the firemen to manipulate. Provision is made in the front wall of the furnace for observing the condition of the fire, so that the air may be regulated accordingly.

The bridgewall proper is 12 inches high and the grates are set 48 inches lower than the shell of the boiler. The bridgewall is continued back to a point just below the blowoff, the idea being to confine the gases within a reasonable distance of the heating surface. This furnace is regularly equipped with an efficient rocking grate. It is manufactured by the Teller Air-Burning Furnace Company, Rogers hotel, Minneapolis, Minn.

In the period 1846 to 1855 only 8,000,000 tons of coal were produced in this country. The tonnage increased thence up to the period of 1876 to 1885, when it was 84,000,000 tons. During the period 1896 to 1905 it increased to 283,000,000 tons, and in the last six years it increased to 436,000,000 tons.

Engine Wreck at Stafford Mill

On Tuesday morning, June 7, as the engine in No. 1 Stafford mill, Fall River, Mass., was being started, the vacuum was lost and as one of the attendants opened the forced-injection valve while the other closed the throttle through some misunderstanding, water found its way into the low-pressure cylinder, causing a serious wreck. As the water was caught in the head end of the cylinder the shaft was forced out of the frame, and the rim of the flywheel striking the side of the foundation broke one of the segments. A detailed description of the accident will appear in a later issue.

Locomotive Boiler Explosions

Blanks were sent by a committee appointed by the American Railway Master Mechanics' Association, upon Design, Construction and Inspection of Locomotive Boilers, to all of the principal railroads in the United States, asking for information in regard to boiler-inspection rules and regulations, and also as to casualties resulting from boiler explosions of all kinds. Replies were received from 157 railroads, giving the number of boiler explosions and failures, and casualties to employees and others, resulting therefrom during the period from January 1, 1905, to November 1, 1909. These 157 railways own and operate 43,787 locomotives and 157,169 miles of roadway, and during the period above mentioned made 6,012,057,467 locomotive miles. The committee estimates that the reports cover over 75 per cent. of the locomotives in use in the United States.

Explosions and failures of locomotive boilers are divided into five classes, as follows:

Explosions of boiler shells.
Explosions of fireboxes.
Damage by burning.
Rupture of flues.
Boiler-pitting failures.

Of the first three classes 98.3 per cent. were due to low water, and 1.7 per cent. to other causes. Of the failures due to low water 98.6 per cent. were due to the failure of men handling or in immediate charge of the locomotives to maintain a proper supply of water in the boilers. The remaining 1.4 per cent. were due to other causes.

Of the 407 killed and injured, 386, or 94.8 per cent., were due to accidents caused by low water, while the remaining 21, or 5.2 per cent., were from other causes, some of these being the result of or incident to wrecks, and a small number are thought to be due to accidents caused by defective design, material, workmanship, or to the poor condition of the boiler or fittings; but it is doubtful if any of them could have been prevented by any method of inspection, in addition to that which is now in force. The other classes of failures were as follows:

While the number of flue ruptures appears in itself to be large, it covers the record of an average number of 42,200 locomotives per annum, for a period of four years and ten months. Allowing 250 flues to each boiler, the result shows one flue failure per year to each 15,912 flues in service, or a percentage of only 0.00006 of 1 per cent. on an average 1,876,422 miles were made for every tube that was ruptured.

From Senate document No. 682 it appears that the average number of employees killed and injured per annum on account of boiler explosions and locomotives for the period from August 1, 1903, to November 1, 1908, was 47.7 employees and others killed, and 134.2 injured. The Senate documents include all the locomotives in use during the time mentioned. The committee's returns show an average of 38 killed and 63.1 injured for about 75 per cent. of the locomotives.

Suppressing Smoke in Milwaukee

According to the records of the smoke inspector's department in Milwaukee, there are 478 steam plants in operation in the city, containing a total of 1080 boilers. On the whole the plants are reported as not giving a great deal of trouble in the way of smoke. Nearly every make of stoker is represented in the equipment of these plants as well as various modifications of hand-fired furnaces. Tugs in the river are still a great nuisance to the smoke department and locomotive smoke is also a factor. The Northwestern is reported as having the best system of smoke suppression, the road employing a smoke inspector who devotes his entire time to the Milwaukee yards, while the St. Paul road is not so satisfactory, as the inspector for this road is required to divide his time between Chicago and Milwaukee.

An interesting series of tests was recently made to determine the accuracy with which a camera would record smoke coming from a locomotive. Railroad men claimed that the vapor caused from the steam blowers would show as heavy smoke in the camera. It was demonstrated, however, that the camera was not at fault in this regard.

Formerly it was difficult for the smoke department to obtain convictions on the report of its observers, the steam users insisting that they had not smoked at the time specified. There has now been worked out, however, a system of charts accompanied by photographs which has proved entirely satisfactory to the department. These charts divide the time up into minutes, the observer making note of the character of the discharge from the chimney for each minute during the hour. Accompanying this chart is a series of photographs of the stack with the time marked on the chart and also on the photograph. Using this system the department has never failed to obtain a conviction where a steam user has violated the smoke ordinance.

In this regard the annual report of the smoke inspector (Charles Poethke, third floor, City Hall) for the year 1909, issued in bulletin form, will undoubtedly be of interest. The bulletin tells of the progress made in Milwaukee in preventing smoke and gives the city ordinances relating to the suppression of smoke.

NEW PUBLICATIONS

Electricity. By H. M. Hobart. Published by D. Van Nostrand Company, New York. Cloth; 225 pages, 5½x 8½ inches; 115 illustrations; 43 tables. Price, $3.

Like all of Mr. Hobart's writings, this book has a strong practical-engineering flavor and his treatment is not restricted to the well beaten track commonly followed by textbook writers. The features which incite criticism are very few; the most objectionable one, so far as American readers are concerned, is the use throughout the work of metric units and the Continental practice of using commas for decimal points. To a student unfamiliar with this latter custom, the tables and numerical examples are likely to be confusing if not actually unintelligible.

A second fault is the use of arbitrary terms as though sanctioned by common usage or recognized authority; for example, one kilowatt-hour is called a "kelvin" and one watt-hour a "siemens." Both are universally honored names but their application to these units has never been approved by any authoritative body.

The author's use of the ton as the practical unit of weight is also unfortunate. Few students are accustomed to think in any such huge units; moreover, as the author is unwilling to talk in English units it would have been wiser to avoid the use of a term that might be mistaken for one.

In other respects, and especially as to technical accuracy and clarity of exposition, the book is excellent. It is not so much an elementary textbook as it is a presentation of fundamental principles. Chapter X, on Inductance, is particularly lucid and in Chapter XII, on Insulating Materials, the experience of the author in designing electrical machinery bears good fruit. We hope that an early future edition will be expanded to make a thoroughly comprehensive treatise; if the author will do that, correcting the few existing defects and maintaining the otherwise high standard of the present edition, the result will be what the small boy would call a "crack-a-jack."

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

GASOLENE ENGINE. Earl Vanderlip, Alto, Mich., assignor of one-half to Arthur O. Bickford, Alto, Mich. 957,278.

INTERNAL COMBUSTION ENGINE. Chas. W. Higgs, London, England. 957,336.

MULTISTAGE TURBINE. Francis Hodgkinson, Edgewood Park, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 957,338.

FLUID PRESSURE MOTOR. Harris B. Holt, Pittsburg, Penn., assignor of one-half to Charles A. Conn, Pittsburg, Penn. 957,339.

CURRENT MOTOR. Thomas J. Steele, Citra, Fla. 957,388.

ROTARY COMPOUND STEAM ENGINE. Charles R. Hilty, Chicago, Ill. 957,508.

ROTARY INTERNAL COMBUSTION ENGINE. James S. Stewart, Diamondville, Wyoming. 957,631.

INTERNAL COMBUSTION ENGINE. Paul Daniel, New York, N. Y., assignor to Benjamin G. Paskus, New York, N. Y. 957,657.

PNEUMATIC MOTOR. Thomas Danquard, New York, N. Y., assignor by mesne assignments to Auto Pneumatic Action Co., New York, N. Y., a Corporation of New York. 957,751.

INTERNAL COMBUSTION ENGINE. Nathaniel Metz, Lansdale, Penn., assignor of one-half to William D. Heebner, Lansdale, Penn. 957,788.

ROTARY ENGINE. Albert E. Rasmussen, North Yakima, Wash. 957,799.

BOILERS, FURNACES AND GAS PRODUCERS

FEEDER FOR FURNACES. Louis P. Bolander, San Francisco, Cal. 957,184.

MECHANICAL STOKER FOR FURNACES. Frank C. Armstead, Pittsburg, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 957,296.

FURNACE OF STEAM BOILERS AND THE LIKE. Walter G. Crosthwaite, Leeds, England, assignor by mesne assignments to the United Cigarette Machine Company, Ltd., London, England, a Corporation of Great Britain. 957,310.

SMOKE CONSUMER. Thomas Rees, Chicago, Ill. 957,377.

FURNACE ARCH CONSTRUCTION. Colin Campbell, Rutherford, N. J. 957,494.

OIL BURNER. Valdemar F. Lassoe, Brooklyn, N. Y. 957,614.

BURNER. Oriel M. Graves, Spokane, Wash. 957,857.

POWER PLANT AUXILIARIES AND APPLIANCES

SURFACE CONDENSER. Louis A. Alberger, Greenwich, Conn., assignor to Alberger Condenser Company, New York, N. Y., a Corporation of New York. 957,175.

VACUUM TRAP. Leslie P. Strong, Cleveland, Ohio, assignor to the Clarke Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 957,269.

VALVE FOR GAS ENGINES. Robert White, Beaver Falls, Penn., assignor of one-half to E. S. Burns, Beaver Falls, Penn. 957,285.

PISTON VALVE. John T. Wilson, Jersey Shore, Penn. 957,286.

VAPOR TORCH. Thomas Yates, Sunnyhurst, Darwen, England. 957,292.

STOP AND CHECK VALVE. George C. Davis, and Walter A. Daley, Chicago, Ill., assignors to George M. Davis Regulator Company, Chicago, Ill., a Corporation of Illinois. 957,311.

VALVE. Cifford A. Ellis, Bayonne, N. J., assignor to Standard Oil Company, of New York, a Corporation of New York. 957,318.

COMBINED STUFFING BOX, GASKET, VALVE AND SPRING. Guy L. Kennedy, San Francisco, Cal., assignor to National Carbonated Liquid Co., San Francisco, Cal., a Corporation of California. 957,347.

VALVE. Ernest E. Sweet, Detroit, Mich., assignor to Cadillac Motor Car Company, Detroit, Mich., a Corporation of Michigan. 957,391.

VALVE FOR STEAM TRAPS AND THE LIKE. Charles J. Mackerey, Newport News, Va. 957,469.

VALVE FOR STEAM TRAPS. Charles J. Mackerey, United States Navy. 957,470.

PRESSURE GAGE. Coleman J. Manning, Medford, Mass., assignor to American Steam Gauge and Manufacturing Company, Boston, Mass., a Corporation of New Jersey. 957,471.

DISTRIBUTING AND REVERSING EXPANSION VALVE. William M. Coward, Sydney, New South Wales, Australia. 957,499.

GAS ENGINE PRIMER. Lawrence R. Darymple, Abilene, Kan. 957,500.

PRESSURE CONTROLLING VALVE. Frank A. Lockwood, Denver, Colo., assignor to the Denver Engineering Works Co., a Corporation of Colorado. 957,511.

VALVE CHAMBER FOR WATER COLUMNS. Frank C. Anderson, Cincinnati, Ohio, assignor to American Valve and Meter Company, Cincinnati, Ohio, a Corporation of West Virginia. 957,533.

VALVE. Everett P. Allen, Chicago, Ill., assignor to Norwall Manufacturing Company, Chicago, Ill. 957,592.

OIL-PROOF SPARK PLUG. Eli J. Bushey, New York, N. Y. 957,653.

CENTRIFUGAL PUMP. Richard O. Jones, Dayton, Ohio, assignor by mesne assignments to the Computing Scale Company, Dayton, Ohio, a Corporation of Ohio. 957,682.

COUPLING. Ernest Marek, Chicago, Ill. 957,695.

VALVE FOR INTERNAL COMBUSTION ENGINES. Francis W. Brady, New York, N. Y. 957,730.

CARBURETER. Francis W. Brady, New York, N. Y. 957,731.

REGULATOR FOR PNEUMATIC MOTORS. Thomas Danquard, New York, N. Y., assignor by mesne assignments to Auto Pneumatic Action Co., New York, N. Y., a Corporation of New York. 957,752.

SLIDE-VALVE MOTION FOR STEAM ENGINES. Eugen Frikart, Mulhausen, Germany, assignor to Elsassische Maschinenbau-Gesellschaft, Mulhausen, Germany. 957,762.

PUMP. Atwell A. Parker, Waterford, N. Y. 957,794.

LUBRICATION OF THRUST BEARINGS. Charles A. Parsons, Newcastle-upon-Tyne, and John Turnbull, Wallsend, England; said Turnbull, assignor to said Parsons. 957,797.

VALVE. Charles E. Strong, Amityville, N. Y. 957,811.

ELECTRIC INVENTIONS AND APPLICATIONS

ELECTRIC BATTERY. Frank A. Decker, Philadelphia, Penn., assignor to Decker Electrical Manufacturing Company, Wilmington, Del., a Corporation of Delaware. 957,192.

ELECTRIC GENERATOR AND MOTOR. Charles T. Hibbard, Minneapolis, Minn., assignor to Electric Machinery Company, Minneapolis, Minn., a Corporation of Minnesota. 957,209.

COIL FOR ELECTRICAL PURPOSES. Elbert W. Jodrey, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 957,213.

ELECTRIC HEATER. Judson C. Logan, Pittsfield, Mass., assignor to General Electric Company, a Corporation of New York. 957,226.

DYNAMO ELECTRIC MACHINE. Jakob E. Noeggerath, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 957,242.

DYNAMO ELECTRIC MACHINE. William D. Pomeroy, Norwood, Ohio, assignor to Allis-Chalmers Company, a Corporation of New Jersey, and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 957,249.

ELECTRIC SOLDERING IRON. Edwin W. Rice, Jr., Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 957,256.

DIRECT CURRENT TURBO GENERATOR. Edwin C. Wright, Newport, Ky., assignor to Allis-Chalmers Company, a Corporation of New Jersey and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 957,290.

BATTERY-CHARGING AND DISCHARGING SYSTEM. Lothar Fiedler, Stoke Newington, London, England. 957,321.

ELECTRIC FURNACE. Hiram W. Hixon, Philadelphia, Penn. 957,337.

ALTERNATING-CURRENT MOTOR. Valere A. Fynn, London, England. 957,505.

ELECTRICAL SYSTEM OF DISTRIBUTION. Walter E. Winship, New York, N. Y., assignor to Gould Storage Battery Company, a Corporation of New York. 957,530.

ELECTRIC ROSETTE. James S. Crossley, Syracuse, N. Y., assignor to Pass & Seymour, Inc., Solvay, N. Y., a Corporation of New York. 957,548.

STARTING DEVICE FOR ELECTRICAL MOTORS. Hans Weichsel, St. Louis, Mo., assignor to Wagner Electric Manufacturing Company, St. Louis, Mo., a Corporation of Missouri. 957,587.

ENGINEERING SOCIETIES

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UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

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Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

One of the biggest businesses in the world is that of the General Electric Company.

It is one of those concerns that make our foreign friends talk about the hustling vigor of American enterprises.

And, of course, it is a very large advertiser in technical publications. It might be larger in some—but that's another story!

We recently received a letter from one of our good friends in the Advertising Department in which he says:

"No doubt you will be interested in a comment made by a friend of mine, the engineer in charge of a municipal water and light plant, who is an ardent admirer of POWER. He says: 'I do not like the way POWER advertises its ad. business. You would think it necessary to buy most of the stuff advertised in order to be right, when, as a matter of fact, the most of it can't be used in any one plant.'

"My correspondent bears out my belief in POWER advertising by admitting that he reads the advertising of prominent concerns regularly and finds it interesting and helpful in his business."

Which only proves again that there are two sides to every question.

We have received and heard lots of favorable comment on these advertising editorials from subscribers and advertisers.

And we are glad, too, to get the other viewpoint.

One man, at least, who has been reading these articles on advertising, has received the impression that we advocate it as "necessary to buy most of the stuff advertised" when, as a matter of fact, "most of it can't be used in any one plant."

A surmise very far from what we have tried to convey.

We have consistently advocated reading advertisements, because they are interesting and informative, and buying advertised goods because they are reliable and economical.

But to advocate that a man buy something that he cannot use and does not need—that is like advising a bachelor to lay in a stock of Mellin's Food.

Our contention is that it pays to go through the advertising pages with the same care that is given to the editorial pages.

New inventions, better devices for doing things, improved designs in machinery and equipment, are constantly being brought forward.

And the way to know about them is to read about them.

Certainly the single average plant can find use for but a few of the articles advertised.

But many plants are not using devices that might be used with advantage to the firm and credit to the engineer in charge.

And many of these things will never be heard of by the man who only *skims* the selling section of POWER.

We believe that it *pays* to advertise our advertising—pays readers, advertisers and ourselves.

Advertising is worth knowing about. It's one of the world's great forces. It works days, nights, Sundays and holidays. It makes two blades of wheat grow where but one grew before. It creates business, upbuilds enterprises, spreads information, knits the people of widely separated localities in closer bonds and carries its message in universal language to all peoples in all countries.

All of that is back of the advertisements in your paper.

Focused down to your own problem, every advertisement here has its bearing on power generation and transmission.

For you, it is "wasteless" reading.

Power
and The Engineer

Vol. 32 NEW YORK, JULY 12, 1910 No. 28

A DISCARDED boiler, bought by a dealer in second-hand machinery, was recently offered for sale. The boiler had been exposed to the elements so long that flakes of rust could be loosened from the outside surface with a pocket knife. Many of the rear tube ends had been so burned and corroded that they were flush with the tube sheet.

This boiler was not repaired and then offered for sale, but was to be sold just as it stood.

Doubtless some one who has little regard for human life will purchase that boiler and put it into use without spending a dollar for repairs.

The worst of it is that the men who will be employed to care for such a boiler will not know of their danger. This increases the responsibility of those employing them.

When a man is accidentally killed, little is thought about the matter unless the circumstances surrounding the case are of such a nature as to attract the attention of the public.

When a fatal accident occurs, the Press publishes the facts with a head line something like, "Only one man killed."

If a tube in a boiler fails and some of the boiler-room attendants are scalded, the officials will state that it was nothing, only a few men scalded—nothing to speak of. And yet some of these men die within a day or two.

As an illustration that human life is valued cheaply in the commercial world we offer this:

Recently a railroad company was sued for damages resultant of an accident which happened because the trains were not protected by a block-signal system. The lawyer for the company showed in cold figures that it was cheaper for the road to settle damage claims and defend itself in law suits than to fit the tracks with a few semaphores.

Isn't it about time for engineers and others employed about the steam plant to recognize that they must safeguard themselves?

The excuse is often given that an accident was unavoidable, being beyond human power to prevent. Yet men connected with the operation of steam plants know that there are many plants scattered throughout the country which are operated under conditions far from safe.

It may be a steam pipe line, designed for 100 pounds pressure but carrying 150; it may be a bag on the fire sheet of a boiler; it may be that the tubes of a water-tube boiler are badly scaled because there has been no time in which to clean them; or it may be one of a hundred other conditions that cannot be remedied without the sanction of the company. Dangerous conditions should be remedied, for the sake of life if for no other reason.

If human life were considered at its proper value, such conditions would not be allowed to exist. If a man lost his life in an accident the matter would not be reported as, "only one man killed."

Refuse to work where conditions are needlessly dangerous. Become better qualified to determine when a dangerous state of affairs exists.

Clean Water for Condensing Purposes

By Warren O. Rogers

> *Originally the water was taken from a pond supplied by a small creek. In times of freshet the waters of the creek soon filled the pond with mud and debris. To avoid the frequent removal of this matter and to insure a clean supply of water, concrete walls were built to form a raceway for the main current and gate connection made with the pond to supply water as needed.*

The available supply of condensing water is of considerable moment to engineers who run their plant condensing and the condition of the water has much to do with the successful operation of the plant.

In one instance recently noted, the water used for condensing purposes was obtained from a small creek running down from the hills of Tennessee. Along its banks were located slaughter houses, farm yards and manufacturing establishments. As a result, when the water reached a high stage various kinds of carcass, trees, shrubbery, leaves and other refuse almost too numerous to mention, floated down into the pond formed in the mill yard. The water used for condensing purposes was taken from this semi-circular shaped pond, through which the water of the creek rushed with considerable velocity, causing that portion of the pond not in the direct path of the current to fill up with mud and débris. The work of removing this accumulation after each heavy rain made the management of the mill look about for a means of relief, which was found by constructing concrete walls which divided the pond into three parts.

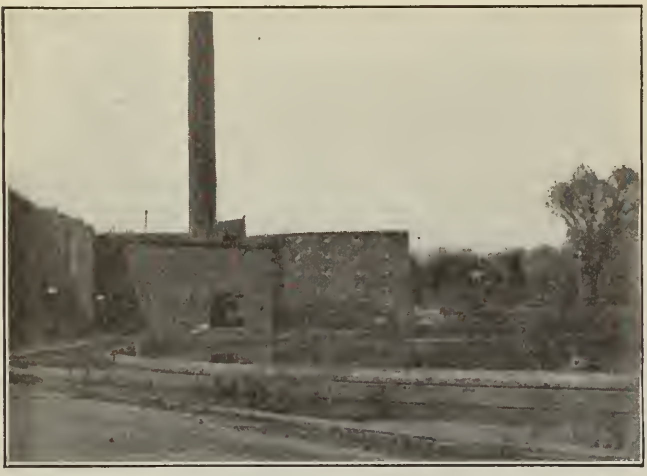


Fig. 1. Pond and Raceway

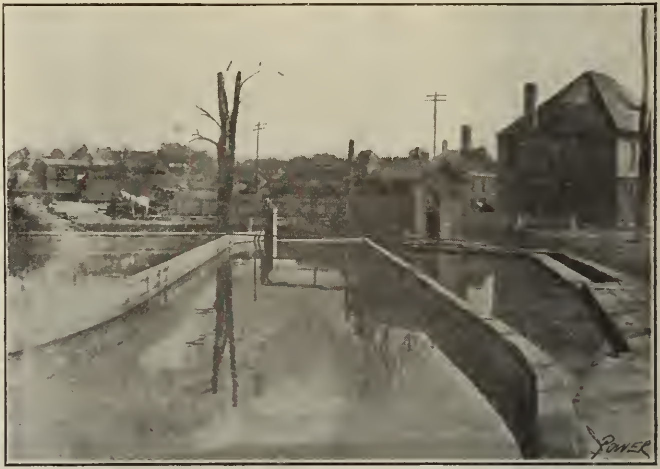


Fig. 2. Near View of Raceway

In Fig. 1 is shown a portion of the pond and the main raceway. Fig. 2 shows a portion of the other end of the pond. Fig. 3 shows a plan view of the entire pond and raceway.

The portion shown in the center of Fig. 2 and slightly in Fig. 1 is the raceway through which the water from the creek passes on its way to the mill yard. Through the winter months the water for condensing purposes can be taken from this raceway when necessary to clean out the reservoir without interfering with the condensing supply, but during the summer months when ten hours' supply from the creek is not sufficient for the condensers during the day, the water is taken from the large pond shown. By confining the water of the creek to the narrow channel of the raceway, the velocity of the current effectually cleans out any accumulation of débris that may come down from the hills, or dirt that may have accumulated during the lower stages of water, which in that vicinity is of a more or less muddy nature.

During the period of sufficient water the condensing water is allowed to run to waste through the narrow raceway *A* shown in Figs. 2 and 3, and over a small dam to the tailrace *B*. The pond *B*, located between the small and large raceways, is used as a reserve cooling pond. It is dammed at the lower end at the gate house, as are also the small and large raceways. The water in the large pond is controlled by gates. The small raceway and small pond are connected by the pipe *D*. At the upper and lower ends of the pond the large raceway and large pond are connected by wooden check valves, through which the flow of water is automatically controlled. This arrangement of valves will be explained later on.

In the summer months when the water is insufficient to provide for the day's run, the water flowing through the creek at night is held back by a dam constructed of boards, each section being movable so as to control the hight of the water at all times. The water held back flows through the gate *F* or *G*, Fig. 3, into the large pond, where it is stored for future use. In this manner enough water is obtained to supply the condensers during the next day's run.

The water from the condensers runs through the small raceway into the small pond and finally into the main raceway. By this arrangement the hot water from the condensers is obliged to heat the entire amount of water in the raceway *E* before it can even begin to heat the water in the large pond from which the condensing water is drawn.

When the water in the raceway *E* is heated, it must flow over into the large pond before it can transmit its heat to the water contained in it, and before the temperature of the water in the large pond is raised enough to interfere with its condensing properties the day's run is ended. By the next morning, the inflow from the creek and the effect of the night air has cooled the water in the pond sufficiently for another day's run.

Fig. 4 shows the device for controlling the gates regulating the flow of water between the raceway *E* and the large pond. It consists of a galvanized cylinder which extends down nearly to the bottom of the concrete wall. The water in the pond governs the operation of the float, which fits loosely in the galvanized

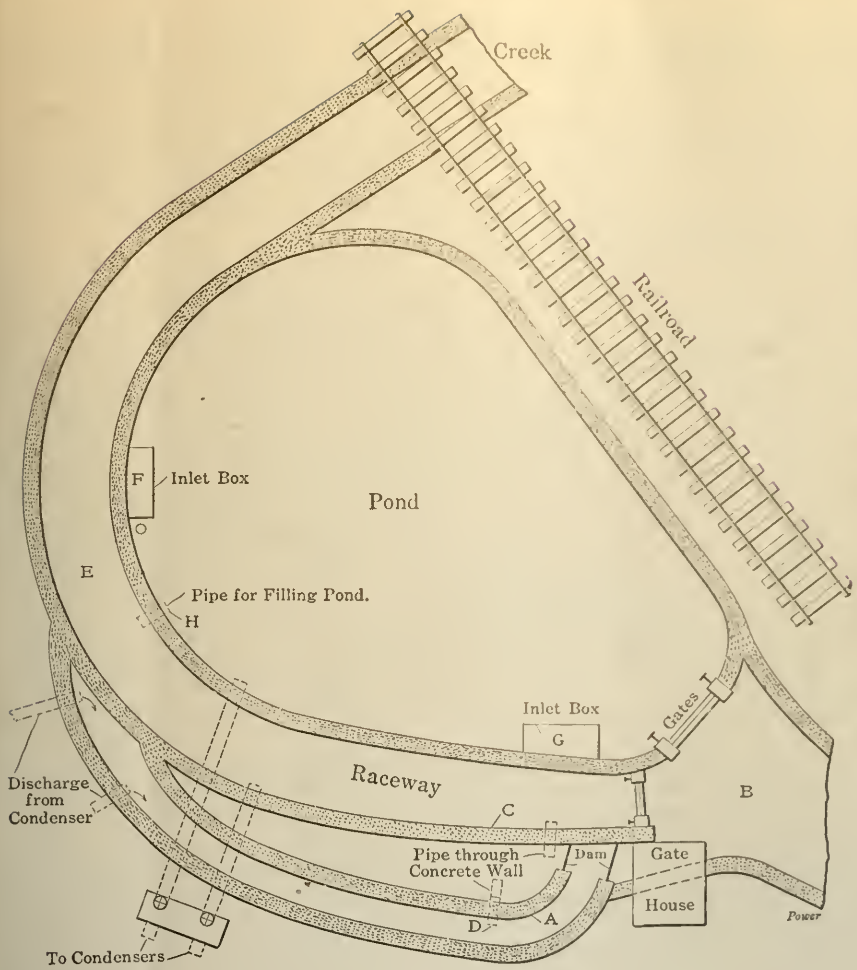


FIG. 3. PLAN OF POND, RACEWAY AND CONCRETE WORK

pipe, as shown. This float is lifted as the water rises, and in rising, lifts the lever *L*, Fig. 4, which is pivoted near the center at *A*. The attachment *B* is connected to an iron cross piece *C*, having a notch at each end. This cross piece engages with the two extension rods, which are secured to the gate rods by set screws. This device holds the gates open. The upper end of the rod *D*, connecting with the float ball, is arranged so that the maximum rise and fall of the ball can be controlled. The hight of the water which will cause the float to trip the gate is regulated by means of the adjusting holes in the float stem.

As the water in the pond rises to a certain predetermined hight, the float is forced up, lifting the cross piece *C* and releasing the extension levers *E* and *F* of the gate shaft. This allows the gates which shut off the water in the raceway from the pond to close by their own weight.

At first, leaves, grass and twigs getting into the gateway, caused considerable trouble, and it was found necessary to put in place the baffle plate *G* shown in Fig. 5. This baffle compels all water passing from the raceway to the pond to go down under it before passing through the passage in the concrete wall between the pond and raceway.

In case it is desired to empty the pond, the gates at the lower end are opened and the check gates *F* and *G* closed. If the water is at such a stage that it will not flow through the gate at the head of the pond, the funnel connection shown in Fig. 1 at the bend of the concrete wall and at *H* in Fig. 3 is lowered and the water allowed to flow from one side of the wall to the other. This pipe has a knuckle joint at the bottom and is fitted with an elbow at the top.

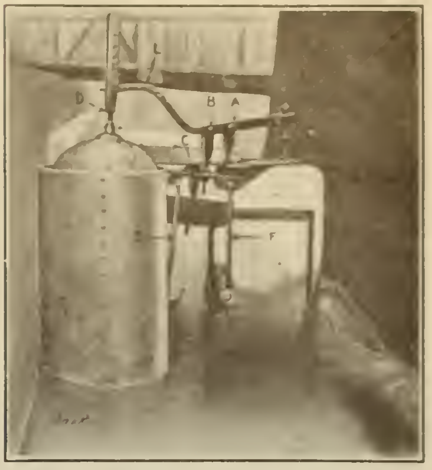


FIG. 4. FLOAT AND LEVER MECHANISM

The value of the entire arrangement is apparent. It furnishes a supply of clean condensing water in proper amount and at a temperature low enough to maintain a high vacuum during the running period.

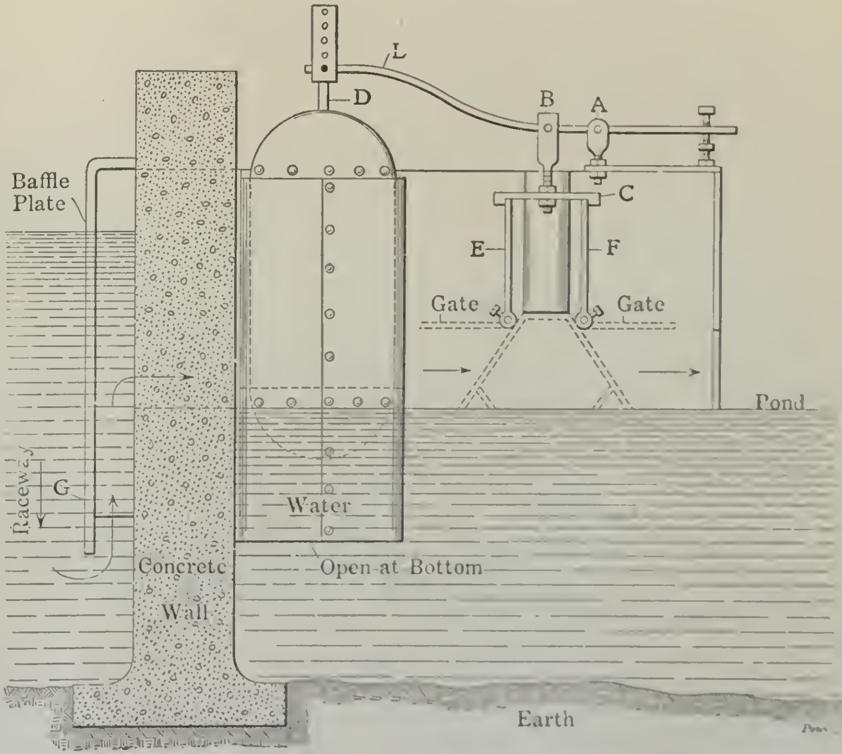


FIG. 5. DETAILS OF AUTOMATIC GATE

Front Feed Automatic Stokers

By J. F. Springer

Front feed stokers have grates inclined from front to back. Gravity and a slight movement given by mechanical means to the individual elements of which the grate is composed cause the fire and fuel to move downward at a uniform rate. These stokers are economical to operate and require relatively small floor area.

By borrowing a slang expression we may speak of gravitation as being "always on the job." Nothing can be done to material things which will affect gravitation. It exerts its influence upon all substances whether they be hot or cold. So it comes about that stoking machines which take advantage of gravitation secure thereby a valuable ally. One great difficulty in the automatic management of a furnace fire is experienced in keeping the grate well covered with fuel. In certain classes of stokers, the grate may become bare in places. This often results in damage to the exposed metal. In the front-overfeed stoker, this difficulty is overcome by tilting and agitating the grate surface. Any tendency to form a bare spot is promptly checked by the gravitation of coal from above. The side-overfeed stokers likewise possess this desirable characteristic.

It will be remembered, perhaps, that William Brunton obtained a British patent in 1822 which disclosed a form of stoker which consisted of tilted grate bars, whose motion, or the motion of one set of alternate rows of bars, combined with gravitation, had the effect of moving the fire *en masse*. At the rear, these bars slid back and forth on a fixed grate. All of these features are retained to the present day, but with important additions.

The present-day grate is made up of individual, tilted grate bars, but these are no longer merely strips of solid metal somewhat thinner between the ends. In Fig. 1 is shown in perspective one type of the modern grate section. It will be noticed that the section is of uniform width. A series of sections arranged side by side, allows but little space between the sections for a supply of air to enter. In fact, no reliance is put upon any supply obtained in this way. If the figure be examined closely, the face of each of the steps will be found to have a transverse slot. It is through these that air is supplied. But these slots do not communicate with the external air. The grate bars are hollow. This will be understood upon referring to Fig. 2. The upper end of each section is open. At this point is set a steam jet discharging into the bar. There is thus supplied a current of mingled air and steam which passes through the slots in the faces of the steps. The object of the steam jet is, primarily, to preserve the bars. This is a big problem for stoker manufacturers; how to give a reasonably long life to the bars, exposed as they are to the highly destructive furnace heat.

A view of the bars assembled is given in Fig. 3. It will be noticed that the sections are not flush with one another. The reason for this will be understood when attention is called to the fact that, as with Brunton's device perfected nearly 90 years ago, provision is made for their movement in sets composed of alternate bars. Further reference will be made to this later. At the present, attention is called to the ash table arranged horizontally at the feet of the bars. The coal which is, of course, fed in at the upper end of the grate surface, moves backward and downward under the combined influence of gravitation and the motion of the bars, burning as it moves. By the time it reaches the bottom of the grate surface it has become mostly ashes and clinkers. These, thrust back into the furnace more and more, fall over the firebrick curb onto the ash slide. This slide is arranged to permit the dumping of the refuse by being drawn forward, the rod for this purpose being plainly in view.

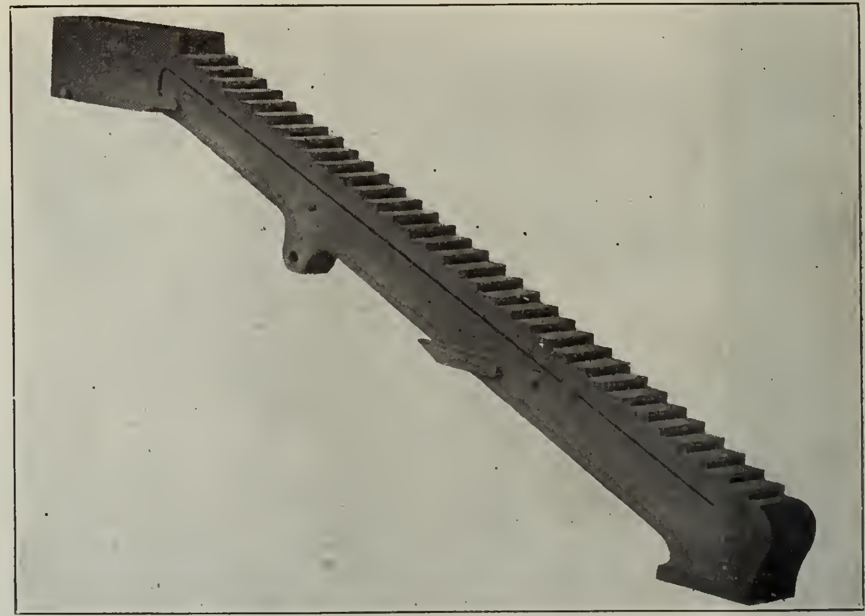


Fig. 1. A Grate Section of the Wilkinson Stoker

Referring to Fig. 2 it will be seen that after the coal is fed in at the hopper, its movement onto the grate surface is assisted by the pusher which slides back and forth on the upper horizontal surface formed by the tops of the grate bars. These bars are supported at their upper ends by a box called the wind saddle which extends across the front of the furnace. The bars communicate with this box through openings *A*. At their lower ends, the grate bars rest on a hollow bearer bar running across the furnace. The grate bars are moved back and forth as they rest on their transverse supports. To accomplish this movement, two hydraulic motors are installed with each stoker. In the upper right-hand corner of Fig. 2 will be seen a detail view of one of the motors together with the operating mechanism. The operation of the motors is effected by means of water supplied by a small pump and tank placed in the boiler room and constituting a part of the stoker equipment. There is a return line of piping, so that the same water is used over and over again. It is said that the steam required for operating the stoker amounts to no more than ½ to 1½ per cent. of the boiler output. The bars are heavy, weighing from 160 to 210 pounds each, varying with the installation. The steam used in the jets is thought to pay for itself in augmenting the heat of the furnace.

In fact, the manufacturing company has made comparative tests, using an engine-driven blower in one case and a steam jet in another, and it was found that the evaporation per pound of coal was 9.22 pounds of water when the blower was used and 10.15 pounds when the steam jet was employed. The advantage secured by the use of the jet was, thus, nearly 10 per cent.

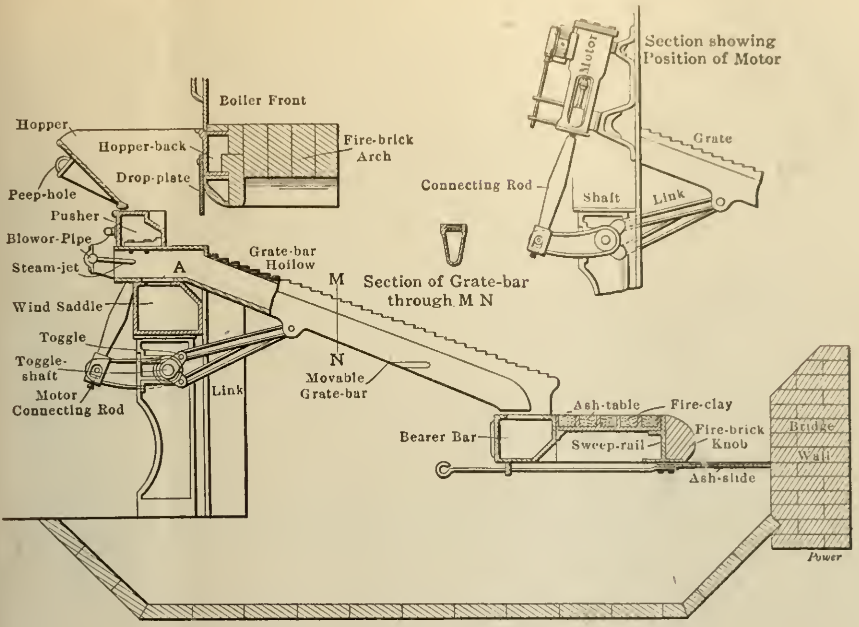


Fig. 2. Details of the Wilkinson Stoker

One feature which is possessed by the front-feed stokers in common with the side-feed type, would seem to promise well for economic operation. In these styles of stoker the power required for the progressive feeding of the furnace is obtained largely from gravity. The coal, by its own weight, drops from point to point on the grate. Gravitation is not only "on the job" but involves no expense in its use. It is to be expected, then, that any stoker which, in whole or in part, is fed by gravity should show a marked economy in cost of operation.

Another type of front-feed stoker will now be considered. In general appearance the grate surface of this apparatus resembles a flight of steps with rounded upper edges. This will be understood by referring to Fig. 4. Here, as will be noticed, a portion has been cut away to show the construction more clearly. Each "step" may be considered as a grate bar. Each of these bars consists of a number of separate tops assembled upon a transverse supporting bar. It is only the upper edges of the tops which come into actual contact with the bed of incandescent coal. The tops used for the four upper bars of the grate have horizontal ledges on their sides. When the grate bar is completely assembled, there is presented an ordinary grate surface with longitudinal air slots. The depth of these slots affords a considerable cooling surface which tends to reduce the temperature of the grate. The horizontal ledges of the four upper bars are for the purpose of preventing the loss of fine coal while still permitting the ingress of air from below. When the coal has reached a lower portion of the grate, it is in a more or less consumed state and it is desirable that any ash coming in contact with the grate surface shall have opportunity to fall through. Hence, the style of the detachable tops is changed for the lower bars, as may be seen by referring to Fig. 4. Just above the uppermost grate bar there is a dead plate and upon this the coal is delivered from the hopper. Below the lowest grate bar are two perforated plates. The upper one of these is the guard grate. It may be raised by means of a rod to the position shown by dotted lines in Fig. 5. When thus raised, it prevents the bed of coals from slipping down. The guard grate is raised preliminary to operating the lower perforated plate. This latter is the dumping grate. It is hinged at a point back of its center and permits ashes and clinkers to be dumped in front or in back of the hinge. In Fig. 4 may be seen the rods which control the guard and the dumping grate.

The grate tops are interchangeable and can be readily removed or inserted. Inasmuch as the grate surface is not everywhere equally exposed to the heat, the wear of the upper surface of the tops varies with the position in the grate. However, the interchangeability of these parts makes it possible to redistribute them from time to time and thus equalize the wear.

A grate bar is provided at each end and below with a cylindrical bearing surface. The bars rest, by means of these surfaces, in suitable notches arranged in two inclined supports. The upper portion of one of these supports may be seen in Fig. 4. Each grate bar may be lifted into and out of its position as a unit. When in place, the character of its bearings permits a rocking motion of the bar. This rocking movement of the grate bars is desired largely for the purpose of assisting gravitation in progressively moving the fuel on and down. To provide this rocking movement is the object of

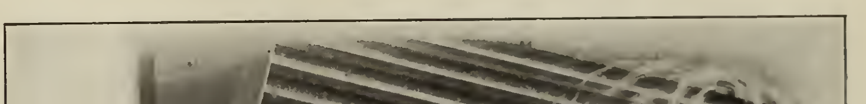


Fig. 3. Sectional View, Showing the Operation of the Wilkinson Stoker

the application of power to the stoker. When a grate bar is lifted into its bearings, it is so balanced as to assume the normal position by its own weight. The rocking motion is accomplished by means of two arms which engage the lugs depending from the supporting arm of the grate bar. By means of suitable mechanism, these arms are moved back and forth and so cause the tops to rock. A portion of one of the rocking arms may be seen in Fig. 4. The cylindrical form of the surface which engages the lugs of the grate bar may, perhaps, be better seen in Fig. 5. The wear on the bearings of the grate bars is automatically taken up as it occurs by their weight.

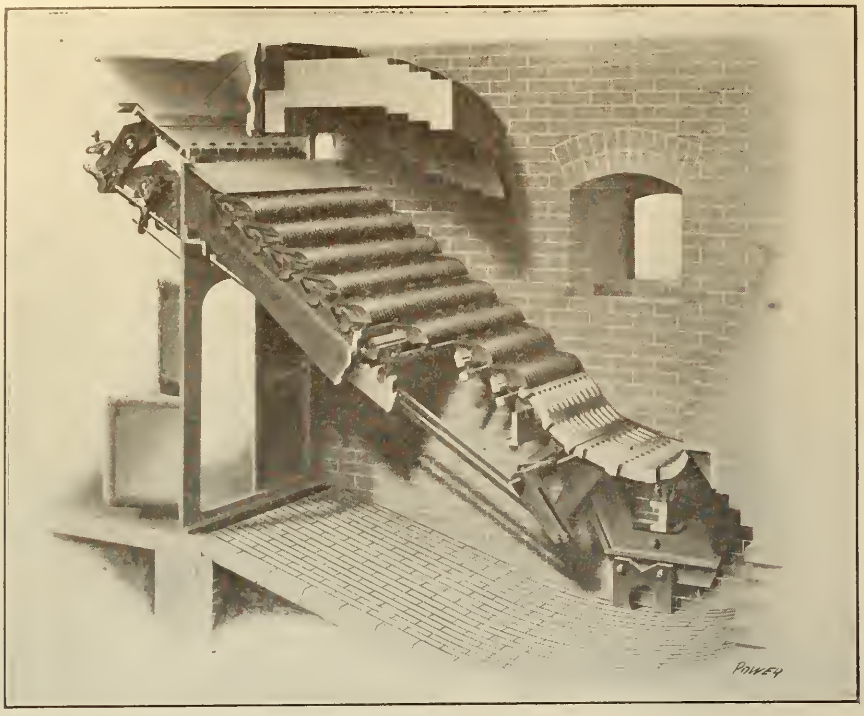


Fig. 4. Sectional View of the Roney Stoker

For establishing the amount of wear undergone by the tops, a test was made of eight samples taken from different parts of a grate surface below the line of the arch. These tops had been in service for seven months. The weight of a top, when new, is two pounds and fourteen ounces. It was found that the greatest loss of weight suffered by any top was eight ounces. Two lost five ounces each; two tops, four ounces each; one lost three ounces; one lost two ounces; and one top lost one ounce. The average loss was four ounces per top for a period of seven months. For a year, at the same rate, the loss would be nearly seven ounces per top. The life of the actual burning surface determines the life of the whole top. It has been calculated that the average loss in weight of the burning surfaces is about 17 per cent. per year. The operation of the grates, in the test just mentioned, was exceptionally severe. The fires were forced continually. At times the boiler developed a capacity 100 per cent. beyond its rating. It is reasonable to expect that the detachable tops of this stoker need replacement but once in six years. To secure this result, however, it is necessary to redistribute the tops from time to time.

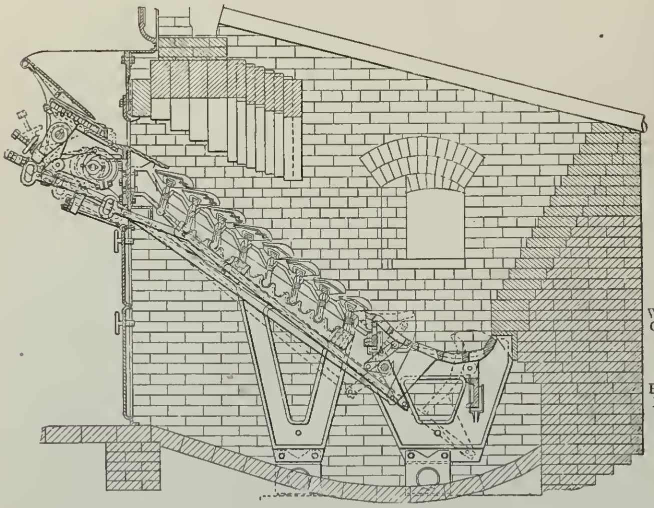


Fig. 5. Details of Roney Stoker

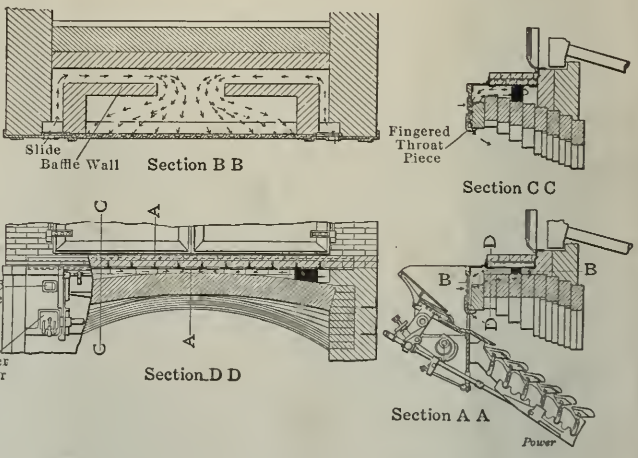


Fig. 6. Details of Arch and Baffle Wall

The mechanism controlling the rocking movement may be seen in Figs. 4 and 5. It will readily be granted if the mechanical construction is considered, that the power required for operating the mechanism is small. The grate bars themselves are well balanced so that the power is mainly expended on the actual movement of the fire and the friction of the bearings. Gravitation assists materially in the onward movement of the fire.

The furnace in which a stoker of this type is installed should fulfil certain conditions. A low arch should extend across the mouth of the furnace. There should be a baffle wall above this arch for the purpose of causing a current of air, admitted from the front of the furnace, to circulate over the top of the arch, and then to find an exit into the furnace at the front. This will be better understood by referring to Fig 6. The poker door may be seen at the lower left-hand corner. In Fig. 4 underneath the arch, may be seen the opening in the front wall covered by this door. Above this poker door, Fig. 6, is shown the wind gate. This is for the purpose of admitting air above the arch. The air passes in and back to about one-half the depth of the arch. The arrows show the path which the air follows, first back and then to the front, where it is directed downward and finds its way into the combustion chamber just under the front of the arch. It will be noticed in Fig. 6 that there are two wind gates, supplying the air from both sides, and that there is a symmetrical arrangement of the baffle wall. The fuel is fed in from the hopper onto the dead plate. The coal is supplied with no circulation of air through its mass. However, it is subjected to the heat of the contiguous fire, reinforced by that of the combustion arch. A process of distillation takes place and the volatile hydrocarbons rise from the surface. The air coming down through the nozzles from the hot chamber above is mingled with these gases and the whole is directed downward and to the rear by the contour of the arch. This arrangement would seem to favor complete combustion. The oxygen necessary for the production of CO_2 and H_2O from the hydrocarbons is not only supplied but it is supplied in a heated form. Moreover, the many nozzles distribute the supply well. The arch tends to promote combustion in that it maintains a high temperature itself and directs the mixed hydrocarbons and air into the hot part of the combustion chamber. It will be remembered that the first four grate bars are provided with non-sifting tops; in other words, provision is made to meet the possibility that there may be some coal on the first four bars that has scarcely begun to be consumed. However, there is a supply of air from below, the arrangement of the horizontal sifting ledges permitting this. That is to say, the design of the stoker contemplates that more or less coking may be going on until the fuel has dropped onto the fifth grate bar. The rocking motion of the bars, which is always going on slowly, results in the progress of the fuel down the stepped surface of the grate. As the bars rock back and forth there is a change in the amount of air supplied from below. The amount of this variation is about one-third of the air supply at its maximum.

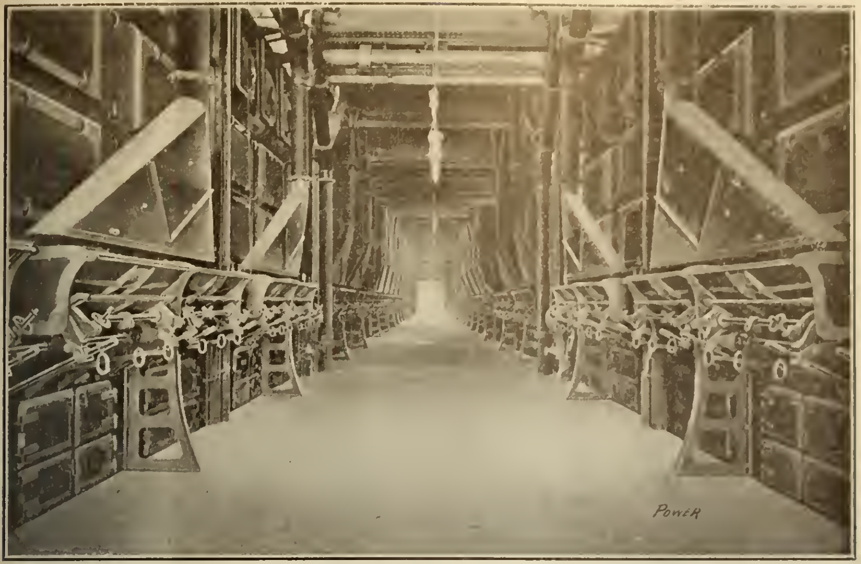


Fig. 7. Installation of Roney Stokers

By referring to Fig. 6 it will be noticed that the air coming forward from the rear of the baffle walls does so over the center of the arch. Other things being equal, one would expect that it would rush straight forward to the central nozzles. This would have a tendency to oversupply the center of the mass of hydrocarbons rising from the coking fuel and to undersupply the sides. However, by studying the figures, it will be seen that the space between the roof and the arch is increased toward the sides. This has the effect of equalizing the distribution of the hot air to the nozzles. The tendency of the air to rush straight forward tends to produce a congestion which reacts in forcing air into the freer side spaces.

That this stoker and furnace are competent to produce a smokeless combustion of the hydrocarbons is well shown by the observation made by the Cleveland, Ohio, authorities of the combustion obtained by the Cleveland Illuminating Company. The combustion beneath the boilers supplied with these stokers, although they were being operated above their rated capacity continuously, was determined to be 99 per cent. perfect.

In Fig. 7 is shown a large installation of these stokers at the plant of the Interborough Rapid Transit Company, New York City. The total boiler capacity of the entire plant is about 37,500 horsepower.

Stokers of this general type have been in use for the past 25 years. It is said that in this time machines having an aggregate boiler-horsepower capacity of 1,250,000 have been installed. From this it will be gathered that it is by no means an untried apparatus.

Another type of front-feed stoker is shown in the sectional view in Fig. 8. The position of the arch and the wall at the inner end of the fire chamber is shown. The grate is inclined at an angle of about 40 degrees. It is made up of detachable bars. These are practically smooth on one side and have ribs on the other. In the figure the ribbed side is shown. The upper ribs are but slightly inclined to the horizontal. In fact, what dip there is, is directed inward. The ribs below incline steeply to the front and provide between them ready exits for ash. The horizontal character of the upper ribs is for the purpose of preventing unconsolidated fine coal from sifting through but at the same time permitting an adequate supply of air to pass into

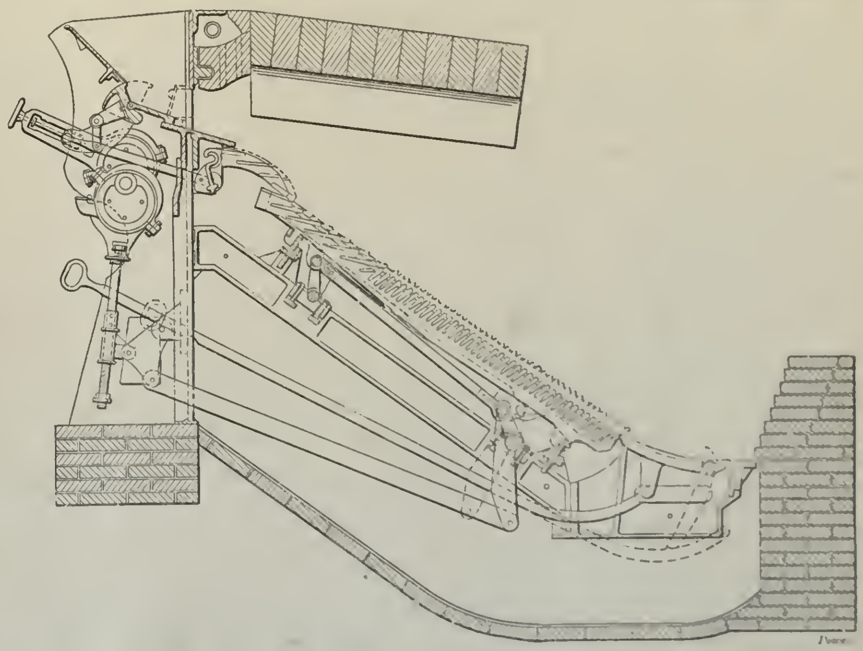


Fig. 8. Details of the Wetzel Stoker

the furnace. Above the grate proper is the coking shelf where the air supply from below is either almost entirely or wholly cut off. This consists, in the main, of a movable and sharply inclined surface. Above this is a stationary shelf having a moderate dip. This may be regarded as really a part of the feeding hopper. Above, and to the left, the bottom of the hopper consists of two surfaces linked together and capable of a back and forth movement. When moved toward the rear of the furnace, the one straightens up and executes a considerable thrust, forcing the coal onward. The other surface moves on somewhat, sliding on the fixed shelf. A study of the link arrangement connected with the movable bottom will show that a rod goes off to the rear to operate the movable part of the coking shelf.

The arrangement is such that both movements are simultaneously accomplished. The amount of this movement is controlled by a screw seen on the left. The movement itself is secured by means of an eccentric arrangement of which the screw forms a part. This device is partly hidden in the view.

Another eccentric is to be seen in front. These eccentrics are oppositely placed but operated by the same shaft. The length of the rod of the latter eccentric is regulated by a lock nut as shown. By means of a link arrangement, pivoted to a fixed pin secured to a lug on the face of the furnace, the entire device is operated by the eccentric to produce a slow shaking movement of the grate. The dumping of the ash grate is effected by the hand rod running the length of the stoker.

A notable installation of this type of stoker is that in the power plant and water works at Tacony, Philadelphia, Penn. There are six boilers of 500 horsepower each. These are arranged in pairs, the members of the pairs being connected by underground flues. The hot gases are guided alternately upward and downward by three transverse plates with the result that they leave the boiler flowing downward by an underground flue.

In Fig. 9 is a perspective view of this installation. A small engine supplies the power. The enormous speed reduction necessary is accomplished by a worm-and-gear arrangement inclosed in a cast-iron housing. The view of the first boiler shows the handwheel which controls the link movement connected with the hopper bottom and part of the coking shelf. The eccentric which operates the grate movements may be plainly seen at the right-hand side of each hopper in the photograph.

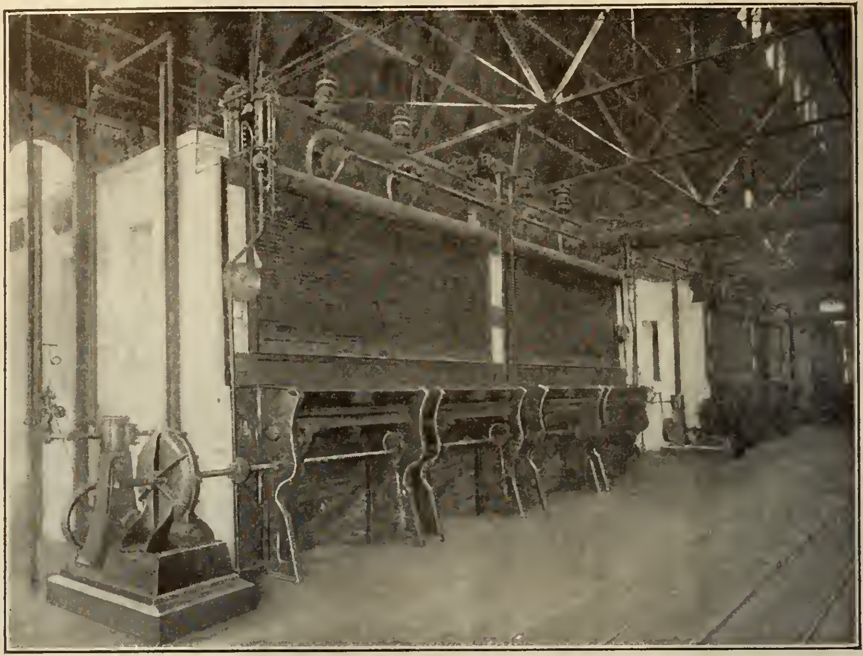


Fig. 9. Installation of Wetzel Stokers

Purchasing Oil By Specification

By Alfred Williamson

> *Lubricating oil is difficult to purchase on specification because its lubricating qualities cannot easily be measured. The Department of Water Supply of New York City purposes to contract for its lubrication on a basis of foot-gallons of water pumped.*

Few contributions on the subject of lubrication come from the purchaser; the majority deal with the matter from the viewpoint of the ultimate consumer, or of the laboratory investigator. There are, doubtless, many cases of central-office supervision of a number of isolated plants, in which all supplies are purchased and repairs and renewals are directed by an engineer who never actually sees the materials or work ordered by him.

The City of New York is bound by its charter publicly to open bids on all expenditures in excess of $1000. Thus, in the matter of lubricants it is necessary that an annual estimate of the consumption be prepared, and if the total exceeds the limit imposed, specifications must be prepared and a contract awarded to the lowest bidder.

A further provision in the charter prohibits the specific mention of trade or brand names unless the destructive clause "or equal thereto" be added. If the engineer wants Jones' oil, for instance, which he knows to be a good brand, he must call for "Jones' A No. 1 brand, or equal thereto" and it will be his duty to prove that a cheaper grade submitted by Smith is *not* equal thereto.

Packing, oil and coal are the most troublesome commodities which any engineer is called upon to consider. Private consumers, however, who can purchase the two first by brand names and get what they wish or expect, are absolutely care-free in comparison to the municipal engineer who is hampered by the limitations mentioned. The coal problem is common to all, but the B.t.u. basis of purchase is expected to solve it.

In purchasing oil, the procedure followed by the Department of Water Supply, Gas and Electricity is typical of the method employed in all of the city departments. A standard form of supply contract is employed, the embodied specifications meeting with annual alterations as experience accumulates. Hitherto it has been customary to state that the oil must be one of a number of stated brands, *or equal thereto,* and to require that a sample be submitted with the bid. The list of brands was followed by a carefully considered schedule of physical properties, approved by the department chemist, and based upon what really good oils it had been the good fortune of the engineer to procure. A further clause, weakly clamoring for a good lubricant, was added as a last wish.

The results have been unsatisfactory, due to the difficulty of writing an ironclad specification; for it is well known that the measurable qualities of a lubricant can be closely approximated in any viscous compound, even molasses, the one quality, namely, *lubrication*, being practically indeterminate. Low bids have always been accepted and although, in certain cases, fair oil (at the price) was furnished, it has never been possible to arouse the interest of the producers of the higher grades whose brands were specified.

The last specification prepared set forth the following physical requirements:

"Cylinder oil must be a high-grade mineral oil, steam refined, containing not less than 6 per cent. of acidless tallow oil. The mixture must be absolutely free from resinous or insoluble matter, and it must conform with the following requirements:

"Baumé gravity at 60 degrees Fahrenheit, not more than 26 degrees. Flash point, not lower than 580 degrees Fahrenheit. Burning point, not lower than 650 degrees Fahrenheit. Viscosity at 210 degrees Fahrenheit, not less than 175 Tagliabue. Acid, not more than 0.25 of 1 per cent.

"Engine oil must be a high-grade mineral oil containing not more than 6 per cent. of animal or vegetable oil. The mixture must be absolutely free from resinous or insoluble matter and it must conform with the following requirements:

"Baumé gravity at 60 degrees Fahrenheit, not more than 29 degrees. Flash point, not lower than 420 degrees Fahrenheit. Burning point, not lower than 480 degrees Fahrenheit. Viscosity at 70 degrees Fahrenheit, not less than 450 Tagliabue. Acid, not more than 0.25 of 1 per cent."

In place of the list of standard brands the following was inserted:

"In addition to the requirements above specified, all oil must possess satisfactory lubricating quality and the quality must remain uniform throughout the term of this contract. The lubricating quality will be manifested in the operation of the pumping engines and smooth, silent and economical operation must be maintained with minimum quantities of oil. The minimum quantities have been set by previous consumption and any oil furnished under this contract which for ten consecutive days necessitates increased consumption over the daily average for the previous six months, at any pumping station, will be rejected."

Prior to the preparation of this contract a satisfactory brand of oil had been purchased in the open market pending the public opening of bids, and it was on the consumption of this oil that the above standard was established. The present contract has not been in operation long enough to permit of a thorough judgment of the consumption clause, but this clause is believed to be of greater value as a specification than any enumeration of physical limits, brand names, or other requirements.

A form of contract is now considered whereby lubricants will be paid for on the basis of the work done in the plants; the unit on which payment is to be based will be the gallon-foot of water pumped instead of the gallon of oil as heretofore. On this basis it is believed that the contractor will find it more economical to furnish higher grades of oil because the total cost to him, per unit of work done, will be less than it would be if a larger amount of low-grade lubricant were consumed.

Economical lubrication, without corrosive or abrasive detriments, is sought, and lubrication cannot be better determined than in the engine room. As a standard of economical lubrication, satisfactory consumption may reasonably be set. Lubrication without good economy and parsimony without good lubrication are the limits of consideration; the mean is satisfaction.

An Unexpected Power Factor

By Charles C. Klein

Looms on two floors were driven by shafting located on the ceiling of the lower floor. The tension in the upper belts had offset that in the lower ones. The removal of upper looms resulted in a disproportionate increase in the friction load. The cause of this, until conditions were closely studied, baffled detection.

In the adjustment of differences between those who buy and those who sell mechanical power the consulting engineer often meets perplexing difficulties. Not only do the textbooks, upon which he bases his calculations, seem to ignore and omit important data, but his own practical experience appears, at times, valueless in aiding him to solve the problems which confront him. He is engaged primarily to guard the interests of his client, but in doing this he must always refrain from doing any injustice to opposing interests. Quite unexpectedly he may be called upon to settle very unpleasant and perplexing difficulties, as the following incidents will show.

Two large rooms, each 53x202 feet in size, in a mill in which the writer had charge of the power plant, were rented to a party having 480 power looms, of which 240 were on the second and 240 on the third floor.

Numerous engine tests have established the fact that one horsepower is ample to drive eight of these looms. The system of power transmission is shown in Figs. 1 and 2, in which the dotted lines on the half plan represent belts. The shafting was hung underneath the third floor.

A main line shaft, capable of carrying fully 80 horsepower, was located close to and parallel with one of the longitudinal walls. Three cross-counter line shafts, receiving their power from the main line through 30-inch mortise-miter gears, served to distribute the power through the rooms.

The looms were placed in three double rows, running from one end of the mill to the other. The arrangement is indicated at the left of Fig. 1, the small circle representing the position of the operator of four looms. Each bank of eight looms was provided with a countershaft 1 15/16 inches by 6 feet, 4 inches, which drove four looms on the third floor and four on the second, the belts for the upper four looms passing through holes in the third floor, as indicated in Fig. 2.

Monthly tests showed that 75 horsepower was the average total consumption of power of the two rooms, the working load keeping remarkably close to 60 horsepower, which was charged to the tenant at $50 per horsepower per year. The friction load of 15 horsepower was charged to the account of the landlord.

The average power consumed was slightly less in the summer months than during the cold-weather months. The tests always showed a slightly greater consumption of power during Monday forenoon.

In the course of time conditions arose which made the removal of all of the looms from the third floor necessary. The first intimation, after the removal, that something was wrong, came to the engineer in the form of a much smaller reduction in the consumption of coal than was commensurate with the amount of machinery that had been removed. If the former power amounted to 60 plus 15 horsepower, or 75 horsepower, the amount after the removal should have been 30 plus 15 horsepower, or 45 horsepower, under the most unfavorable conditions. For a decrease of 30 horsepower there should have been a proportionate decrease in the consumption of coal to the amount of about half a ton per day. Instead of this the amount saved was less than a ton a week. It was quite natural to attribute this meager saving in coal to increased friction caused by the shafting being thrown out of level by the removal of the 240 looms from the third floor to which the shafting was hung, the looms aggregating a total weight of over 100 tons.

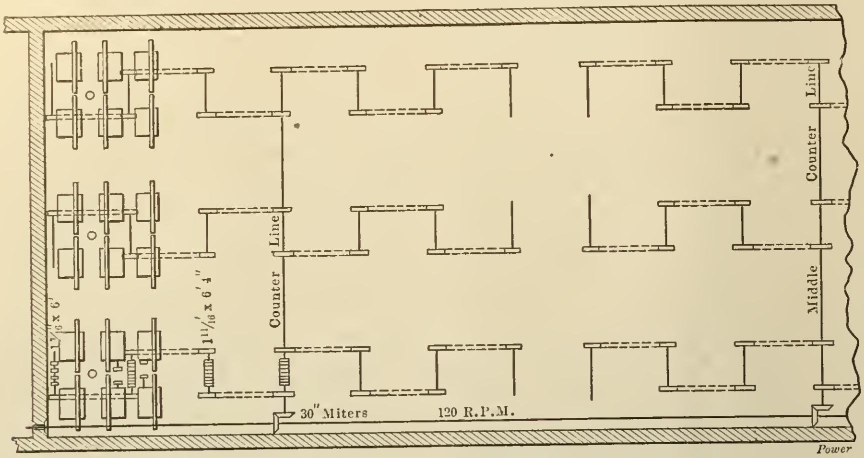


Fig. 1. Plan Showing Arrangement of Shafting

As a first step in the settling of the question the engine was indicated for the total load and then for the friction load. The first diagrams taken were immediately computed and showed a total load of 65 horsepower and a friction load of 30 horsepower, leaving, thus, 35 horsepower consumed by the 240 looms, as the working load. Here was an increase in the consumption of power of five horsepower for the looms and 15 horsepower, or 100 per cent., in the transmission.

A force of men were set to work leveling up the shafting and examining all the hangers to make sure that the bearings were free and that the lubrication was good. As no glaring defects were discovered, the puzzle became greater as the work proceeded. The loom fixers were told that they must look carefully over every loom, that none of them must be "overpicked" and that at the next test every loom must be in first-class working condition. Another test was made with the following average results: Working load, 31 horsepower; friction load, 27 horsepower; total, 58 horsepower.

A slight gain over previous conditions had been made, but as everything had been done that possibly could be done, the consulting engineer went home very much disappointed, if not discouraged, determined, however, that he would solve the problem. He was familiar with every detail of the power plant and the transmission of the mill, having full detail drawings of every belt, gear, shaft, coupling and hanger and their relative positions. He lost no time in looking over all of the drawings appertaining to the transmissions for the two rooms. As all of the trouble had been caused by the removal of the looms from the upper floor the solution of the problem had evidently to be sought for in this change.

The mere removal of the weight had had no effect or only a very slight one in disturbing the level of the shafting, for the mill was very substantially built with 12x18-inch yellow-pine girders, 10 feet from center to center. Hence, the cause of the trouble had to be sought in a primary effect which had been produced by the removal of the looms from the upper floor.

After a prolonged search among the blueprints and a great amount of thinking, he picked up a drawing which is represented by Fig. 2. No arrows were shown on it. He sketched them in with his pencil and a flood of mental light seemed to emanate at once from that drawing. The pulling sides of the belts for the upper looms were on the opposite sides of the pulleys from those for the lower looms and they pulled in opposite directions, thus balancing the strain. What would have been the effect if the upper-loom belts had been put on before those of the lower ones? He made a rough calculation thus: There were 240 two-inch belts, each one having a tension of 60 pounds, which would have a tendency to lift the weight of the shafting and transfer the weight to the small shafts of the looms and as

$$60 \times 240 = 14{,}400$$

pounds, there would be that much less weight to produce friction in the transmissions. Some of this counterbalancing effect would, no doubt, remain after the belts for the lower looms were put on. Not only this tension, but also the driving strain of 60 pounds for every loom on the lower floor balanced by the same amount of driving strain for every upper loom, was what had been relied upon to produce a minimum of friction.

Under these circumstances, strenuous efforts were made to find a new tenant. In the course of a few months one was found who was willing to use the third floor as a weave room and to have his beaming, bobbin-winding and other auxiliary work done outside.

The looms were delivered in instalments of about 40 each and as it was dangerous to introduce the new belts through the belt holes in the upper floor, dangling among the running belts and pulleys underneath the floor, it was decided to put the new belts on at night when the engine was stopped. The engineer was very particular to have all of the belts for the lower looms thrown off before those for the upper looms were put on, and all of the new belting had to be guaranteed to be fully stretched.

After the first instalment of 40 looms had been running about a week a test was made and, as was anticipated, it showed an average reduction of 2.5 horsepower in the friction load. The engine tests were performed regularly as each succeeding instalment of looms was put into service and in each test a regular proportionate decrease in the friction load kept pace with the uniform increase of the working load.

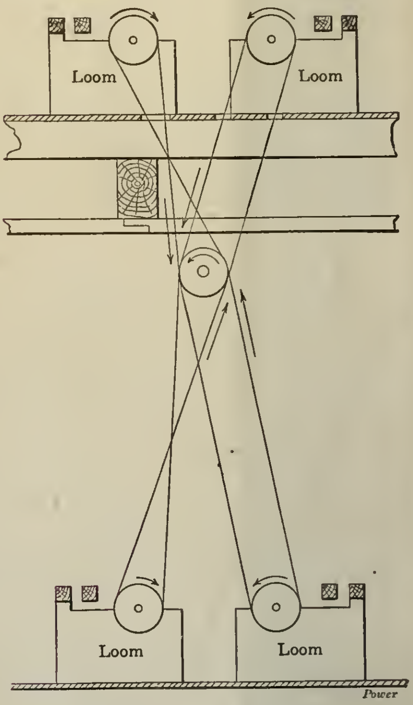


Fig. 2. Vertical Section

About a month after the last instalment was in full working order the indicator diagrams averaged precisely what they had before the removal of the original looms from the third floor.

The electric-lighting system at Fort Rosencranz, which is located near San Diego, Cal., has been connected with the transmission lines of the San Diego Consolidated Gas and Electric Company for service.

A Peculiar Case of Wear

Peculiar cases of wear come to notice occasionally, which are more or less interesting. Fig. 1 shows a case which to all appearances is an ordinary ring made from round brass stock and an oblong piece of metal. Originally, however, the ring was a locknut and the oblong piece an adjusting bolt used in the type of piston shown in Fig. 2.

In this design of piston, the ring was originally set out and the piston centered by means of the bolts *A A*, the studs bearing against the springs *B B*. The studs and nuts, however, had long been out of commission, as the condition of the one shown in Fig. 1 is evidence that it had long passed the point of usefulness. The stud was originally ⅝ inch in diameter, the illustration showing the worn portions enlarged to four times their original size. The piston was apparently leaking steam, and upon being overhauled it was found that some of the locknuts and studs had disappeared entirely, and one was found worn to the shape shown, by the constant working inside of the piston.

The engineer, who had run the engine for years, believed in letting things alone as long as the engine would turn, so the piston had never been examined until the excessive leakage of steam passing by the piston ring made it absolutely necessary.

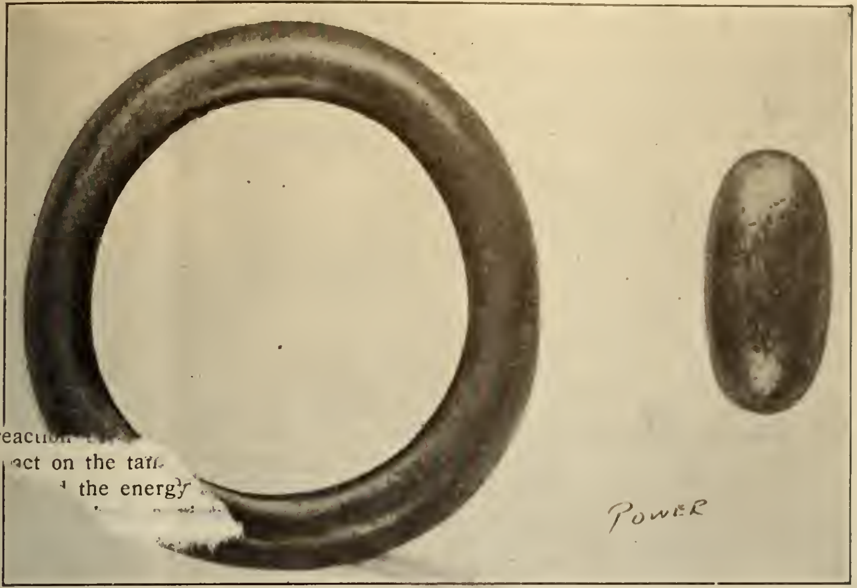


Fig. 1. Worn Nut and Stud

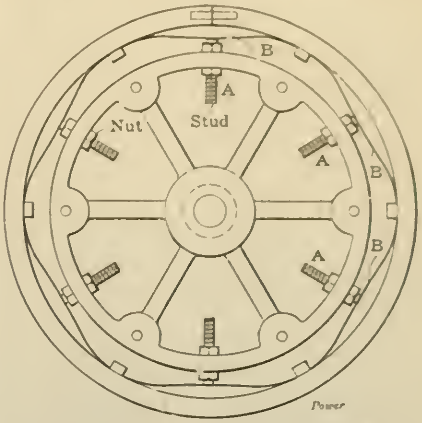


Fig. 2. Showing Type of Piston

Uncle Pegleg's Philosophy

> *Uncle Pegleg explains why a jet has eight times the energy when it runs twice as fast, also why an impact turbine must run at one-half the velocity of the jet to get the energy all out of it; but I get him stuck when he tries to explain that a reaction turbine must run as fast as the jet.*

When the edge had been taken off of our appetites the next day, Uncle Pegleg pulled out a sketch like Fig. 1.

"Suppose," he said, "that the pump was running just fast enough to keep the water level steady while the hose is pulling it out. Then the pump is pumping as much water as the hose is delivering, and against the head *h*, which we will say is 100 feet. Now what would happen if you doubled the speed of the pump?"

"The tank would run over," I said.

"Sure, because the hose is taking out all it can *under that 100-foot head.* But say, that's what you wanted to do yesterday without increasing the head at all."

"We wasn't talking about heads yesterday," I retorted.

"I told you a stream would have eight times the energy in it if you doubled its velocity, and you said it would have only twice as much because the same pump would pump it running twice as fast. The fact is that to get twice as much water a second out of that hose you have to have a whole lot more pressure behind it. Your pump has not only to run twice as fast but to pump against four times the pressure, and this is what makes

$$2 \times 4 = 8$$

times the power.

"Now I want," continued the old man, "to prove to you, or to make you prove to yourself, that that four times the pressure is so. You understand the formula we worked out yesterday

$$E = \frac{W\ v^2}{2g}$$

and allow that it is all right, don't you?"

I agreed to it.

"And you agree that energy is weight × hight, don't you? If you lift 10 pounds, 4 feet you do

$$10 \times 4 = 40$$

foot-pounds of work."

I admitted that too, and he went on:

"Two things that are equal to a third must be equal to each other. If

$$E = Wh \text{ and } E = \frac{Wv^2}{2g}$$

then

$$Wh = \frac{Wv^2}{2g}$$

"If two things are equal," he went on, "and you take the same quantity away from each of them, the remainders will still be equal. The weight is in the quantities on both sides of the equal mark as a multiplier. Take it out of both and you have

$$h = \frac{v^2}{2g}$$

meaning that the hight of column or head *h* necessary to get up a velocity *v* is equal to the square of that velocity divided by 2 × *g*, and *g*, you remember, was about 32 feet per second. Now multiply both sides of the equation by 2g."

I worked it out like this:

$$h \times 2g = \frac{v^2 \times \cancel{2g}}{\cancel{2g}} = 2gh = v^2$$

"Good!" said the old man. "After you have done it times enough you will find that all you have to do to change quantities from one side of an equation to the other is to make them multipliers if they have been dividers, or dividers if they have been multipliers."

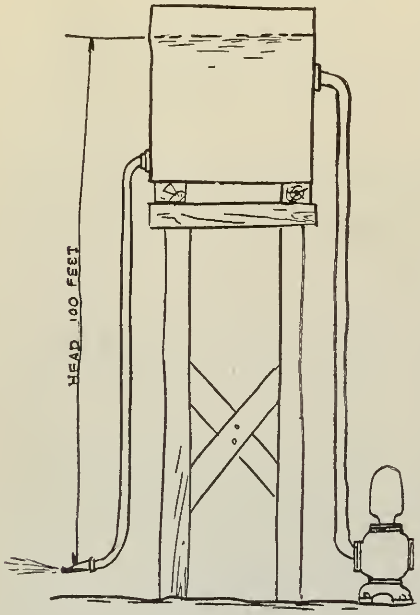


FIG. 1

That one trick has been of more use to me than almost anything else that the old man told me, and I want to devote a chapter later to showing how helpful it is.

"Suppose you had a head of 100 feet, what velocity would you get?" he asked. "Do it with that formula you just got and call *g* 32 feet per second."

I put the figures which I knew, 100 for *h* and 32 for *g*, in the place of the letters, and got

$$V^2 = 2 \times 32 \times 100$$
$$v^2 = 6400$$

"Then," said the old man, "the velocity must be 80 feet per second, because

$$80^2 \text{ or } 80 \times 80 = 6400$$

and

$$80^2 = 2 \times 32 \times 100$$
$$v^2 = 2 \times g \times h$$

Now suppose you wanted to double that velocity; what head would you have to have? Call 2 *g* = 64. Your doubled velocity will be

$$2 \times 80 = 160$$

feet per second. Then going back to the formula before you transposed it you have

$$h = \frac{v^2}{2g} = h = \frac{160^2}{64} = \frac{\overset{20}{\cancel{160}} \times \overset{20}{\cancel{160}}}{\underset{8}{\cancel{64}}} = 400 \text{ feet}$$

You see that to double the velocity you have to have four times the head. You can see that at a glance by your formula which says that the head equals the *square* of the velocity divided by a constant number, so that if the velocity has been 1 and you want to make it 2 the head becomes $2^2 = 4$. Am I right?"

I looked it all over and concluded that I didn't have Uncle Pegleg in so tight a corner as I had thought.

"How hard would it hit a man?" I asked, thinking back to the fire stream and its velocity which had started the discussion.

"Well, that's harder to figure," replied Uncle Pegleg. "All the energy that's stored up in the stream will be given up when it's brought to rest. I've showed you how to figure the energy stored up in it, the energy it takes to get it to going. If you can tell how completely the man takes the motion out of it you can get the energy it spends on him, and if you know how far he yields under the impact you can get the force end of the energy. You see the jet must be brought back to rest before it will give up all the energy it took to get it into motion, just as a raised weight must be let back to the starting level before it will give up all the energy it took to raise it. Take one of those impulse waterwheels with a jet playing on a bucket like this, Fig. 2.

"Suppose the jet was moving 1000 feet per second and that the wheel was standing still. Then if there was no loss by impact and friction, the water would be simply turned around and would travel backward as fast as it came. It would have lost no velocity, have given up none of its energy and done no work. Suppose the bucket to be going as fast as the jet; then the jet certainly wouldn't lose any velocity on it. The head of the jet would be touching it all the time, going with its original speed, and the rest of the water would never get to it. But suppose the bucket to be going at 500 feet per second, half the speed of the jet. Then the water would hit the bucket with a velocity of 1000 — 500 feet per second, 1000 because the jet is traveling at that speed and less 500 because the bucket is running away from it at that speed. The water hitting the bucket at a speed of 500 feet per second would be turned around and shot backward from it at 500 feet per second relatively to [illegible] bucket; but since the [illegible] forward at a speed of [illegible] shoots the water backward at a speed of 500 feet per second measured from itself the water will be motionless with reference to the ground and will fall dead off of the wheel. The forward motion of 500 feet per second which it shares with the bucket will balance the backward motion of 500 feet with which it is thrown off of the bucket.

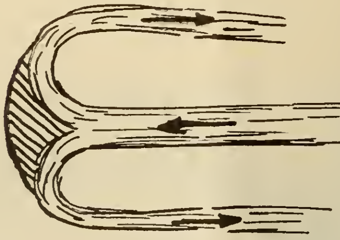


FIG. 2

"So you see in a wheel of this kind or in a turbine the wheel must run at one-half the velocity of the jet or stream which plays on it to get all the motion and energy out of it; a little less actually on account of friction and direction if the water doesn't come and leave straight on.

"It is different if the wheel is pushed around by a jet coming out of it, by 'reaction,' they call it. Ever see one of those lawn sprinklers that is spun by the water that it sprinkles? Well, take a turbine that's run on that principle, the wheel has got to run as fast as the water to get the energy all out."

"How so?" I asked.

"Well, supposing we had a tank like this on a car, Fig. 3, and air pressure on the water so as to make it spout out with a certain velocity. Then in order that the water may come out of the nozzle and fall dead onto the ground the car must run as fast as the water does, so that it can just pay the jet of water out like a rope and let it lay dead between the tracks. If the car is carrying the nozzle ahead as fast as the water comes out of it then the forward motion of the water with the nozzle will just balance the backward velocity with which it squirts out of it and, so far as the ground is concerned, the water will have no motion whatever."

"And if you held the car still the jet

would push like blazes," I ventured, "and when it got up to the speed of the jet it wouldn't be pushing at all. In one case your power is all push and no speed, and in the other all speed and no push."

"What makes you think so?" Pegleg asked. "The tank don't know whether it's on a railroad car or in a brewery, or whether it's playing backward or forward or sidewise. A certain amount of pressure will make that stream come out of there with a certain velocity wherever the tank is and whatever it is doing, and a certain amount of water leaving with a certain velocity will need a certain push to give it that velocity, and as action and reaction are equal and opposite it will react on the tank with the same force."

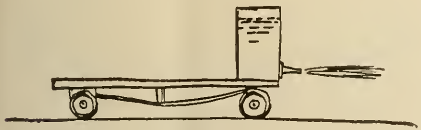


Fig. 3

"And the energy that the car gets out of it is this force times the distance that it moves?" I argued.

"Sure thing!"

"Well, now, if you double up that velocity of the jet you put eight times the energy into it, don't you?"

"Yes."

"Well, how do you get eight times the energy out of the jet by running the car twice as fast; only twice as far in the same time?"

"Because, my boy, you had to have four times the pressure to get that doubled velocity and that makes four times the pressure or reaction on the car and four times the pressure times twice the distance makes eight times the energy."

I was ashamed of having been caught again on this pressure thing, for it was the same point upon which I had gone wrong regarding the pump and the hose.

"How can you tell how hard a jet will push; what the shove on the car would be, for instance?" I asked.

"Do you remember the weight I told you about sliding on the ice? A pound of force got up 32 feet of velocity per second in a pound of matter. Just divide the velocity by 32 (or g) and you will have the pounds of force necessary to get up the given velocity in *one* pound of matter in a second. Multiply that by the number of pounds put into motion in a second and you will have the force required to squirt the jet. The force pushes backward as hard as it does forward, action and reaction being equal and opposite, so the jet reacts with a force equal to

$$\text{Force} = \frac{\text{weight} \times \text{velocity}}{g} \text{ or } F = \frac{wv}{g}$$

Take your weight in pounds and the velocity in feet and both per second, because g is in feet per second, if you call it 32 or thereabout."

"And the product of this force and the distance that the car moves is the energy taken up by the car?" I reasoned.

"Yes, if the velocity of the car is the same at the end as at the beginning."

"If the car moves with half the velocity of the jet, the space that it will go through in a second is the velocity divided by 2, isn't it?"

"Sure as preaching!" assented the old man.

"Then the energy taken up by the car per second, if it moved at one-half the velocity of the jet, would be

$$\text{Energy} = \text{force} \times \text{space}$$
$$E = \frac{wv}{g} \times \frac{v}{2} = \frac{wv^2}{2g}$$

I didn't go to do it but I had brought out the formula for the total energy of the jet which he had explained to me the day before.

"Why!" I exclaimed, "That's all there is in it."

"What's that? Let's see," said the old man, reaching for the pad.

I passed it to him, explaining that he had said that the car must run as fast as the jet came out so that it could pay it out like a rope, and let it fall dead on the ground in order to get out all the energy, but that here it figured out to have taken up all the energy that the jet had in it with the car going at only half the speed of the jet.

"There's something funny about that," said Uncle Pegleg, settling down further into his chair and screwing up his lips as he did when he was puzzled, and he was still screwing at it when I went to start her up.

Examination in Thermodynamics

Complete answers to the list of examination questions given to the junior class in Polytechnic Institute, Brooklyn. The questions were published in Power *for April 5 on page 643.*

The Polytechnic Institute of Brooklyn is a pioneer in evening-class work. It was among the first colleges of engineering to give evening courses which are equal in scope and quality to those offered at the day sessions. A man may secure the degree of Mechanical Engineer, Civil Engineer or Electrical Engineer by attending a sufficient number of evening courses to pass the required examinations and secure the necessary number of counts.

In the April 5 issue, we published a sample of the kind of work which Polytechnic is doing in the form of a set of examination questions in thermodynamics which was put to the junior class at the end of the first semester of the present year. Sufficient interest in these problems was evinced to warrant the publishing of the answers thereto.

1. No engine can have an efficiency exceeding

$$\frac{T - t}{T}$$

in wihch T and t are the highest and lowest temperatures attained, expressed in absolute degrees. (To express temperatures in absolute degrees, add 460 to the Fahrenheit temperature). For the engine in question, the maximum possible efficiency is then

$$\frac{(300 + 460) - (212 + 460)}{(300 + 460)} = 0.1158$$

One horsepower-hour is equivalent to

$$\frac{33{,}000 \times 60}{778} = 2545 \text{ B.t.u.}$$

the figure 778 denoting the number of foot-pounds in one B.t.u. At 0.1158 efficiency, the engine must then receive, per horsepower-hour,

$$2545 \div 0.1158 = 22{,}000 \text{ B.t.u.}$$

and since each pound of steam "contains" 1100 B.t.u., the minimum possible steam consumption, corresponding to maximum possible efficiency, is

$$22{,}000 \div 1100 = 20 \textit{ pounds per horsepower-hour}$$

2. Air conforms closely to the law

$$PV = RT$$

in which,

P = Absolute pressure in pounds per square foot,

V = Volume of one pound in cubic feet,

R = 53.36, an experimentally determined coefficient.

T = Absolute temperature.

Then by substitution,

$$2000 \times 144 \times V = 53.36 \times (460 + 65)$$
$$V = 0.097 \textit{ cubic feet}$$

The volume of the tank is then,

$$10 \times 0.097 = 0.97 \textit{ cubic feet.}$$

3. Since $P_1 = P_2$, the path 1 — 2 is of constant pressure, $PV^0 = c$. Since $V_2 = V_3$ the path 2 — 3 is of constant volume, $PV^{\infty} = c$. A P — V diagram of the cycle would then appear as in the illustra-

tion, the path 1 — 3 being a hyperbolic curve. The area of this cycle represents the work done and also the heat expended. Drawing the dotted lines, this work area is

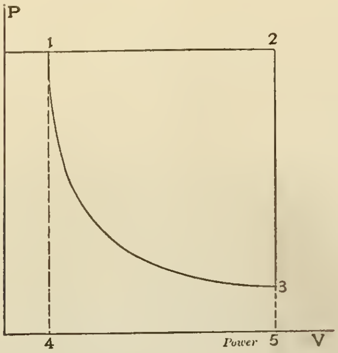
$$W = P_1 (V_2 - V_1) - P_1 V_1 \ loge \frac{V_3}{V_1}$$

the second term being the formula for the work area under a hyperbolic curve. To find V_1:

$$P_1 V_1 = R T_1$$

$$V_1 = \frac{R T_1}{P_1} =$$

$$\frac{53.36 \times (3000 + 460)}{100,000} = 1.85 \textit{ cubic feet}$$



We then have,

$$W = 100,000 (40 - 1.85) -$$

$$\left(100,000 \times 1.85 \times loge \frac{40}{1.85}\right) =$$

$$3,246,000 \textit{ foot-pounds}$$

and the amount of heat converted into work is

$$3,246,000 \div 778 = 4170 \textit{ B.t.u.}$$

4. We first find $P b$:

$$P a \, V a^{1.7} = P b \, V b^{1.7}$$

$$P b = P a \left(\frac{V a}{V b}\right)^{1.7} =$$

$$1000 \left(\tfrac{1}{2}\right)^{1.7} = 307.7$$

The increase of internal energy is, the value of y for air being 1.402,

$$\frac{P b \, V b - P a \, V a}{y - 1} =$$

$$\frac{(307.7 \times 2) - (1000 \times 1)}{1.402 - 1.0} = -956$$

foot-pounds

the negative result denoting that the internal energy has *decreased* in passing from a to b. The formula for external work done along such a path as that given is, n being the exponent of the path,

$$\frac{P a \, V a - P b \, V b}{n - 1} =$$

$$\frac{(1000 \times 1) - (307.7 \times 2)}{1.7 - 1.0} = 549 \textit{ foot-pounds}$$

The heat absorbed is the sum of the external work done and the internal energy inparted, or

$$549 - 956 = -407 \textit{ foot-pounds} =$$

$$-\frac{407}{778} = -0.522 \textit{ B.t.u.}$$

the negative result denoting that heat has been *emitted,* which might have been anticipated from the fact that the path slopes downward more rapidly than the adiabatic,

$$p v^y = c$$

Second Method—The specific heat along any polytropic path is given by the formula

$$s = l \frac{n - y}{n - 1}$$

in which l is the specific heat at constant volume, which for air equals 0.1689. This leads to

$$s = 0.1689 \frac{1.7 - 1.402}{1.7 - 1.0} = 0.0723$$

The temperatures at a and b are

$$T a = \frac{P a \, V a}{R} = \frac{1000 \times 1}{53.36} = 18.77$$

$$T b = \frac{P b \, V b}{R} = \frac{307.7 \times 2}{53.36} = 11.54$$

The heat absorbed is equal to the gain of temperature, multiplied by the specific heat, or to

$$s \, (T b - T a) = 0.0723 \, (11.54 - 18.77)$$
$$= -0.522 \textit{ B.t.u.},$$

as before.

5. In the illustration, we have

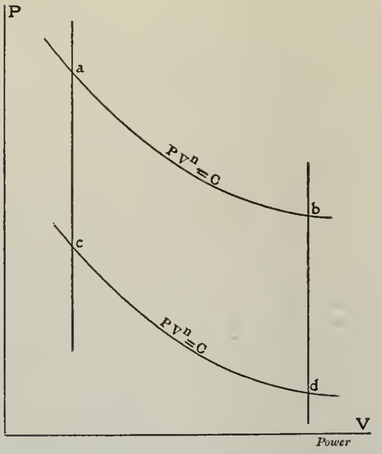
$$P a \, V a^n = P b \, V b^n \qquad P c \, V c^n = P d \, V d^n$$

and since generally $P V = R T$,

$$\frac{P a \, V a}{T a} = \frac{P b \, V b}{T b} = \frac{P c \, V c}{T c} = \frac{P d \, V d}{T d} = R$$

We may then write

$$\frac{\dfrac{P a \, V a^n}{\dfrac{P a \, V a}{T a}}}{} = \frac{\dfrac{P b \, V b^n}{\dfrac{P b \, V b}{T b}}}{} \text{ or } \left(\frac{V a}{V b}\right)^{n-1} = \frac{T b}{T a}$$



Similarly,

$$\left(\frac{V c}{V d}\right)^{n-1} = \frac{T d}{T c}$$

But,

$$V c = V a, \ V d = V b;$$

hence,

$$\left(\frac{V a}{V b}\right)^{n-1} = \left(\frac{V c}{V d}\right)^{n-1} \text{ and } \frac{T b}{T a} = \frac{T d}{T c}$$

or, by inversion,

$$\frac{T a}{T c} = \frac{T b}{T d}$$

which was to be proved.

6. The heat of the liquid of wet or dry steam at 327.8 degrees Fahrenheit is approximately

$$327.8 - 32 = 295.8 = h.$$

The latent heat of vaporization of the dry steam is

$$L = H - h = 1186.3 - 295.8 = 890.5.$$

The latent heat of vaporization for steam 0.95 dry is

$$0.95 \times 890.5 = 854 = L.$$

The total heat of the wet steam is

$$845 + 295.8 = 1140.8 = H.$$

The volume of the wet steam is

$$0.017 + 0.95 \, (4.429 - 0.017) = 4.212,$$

the quantity 0.017 representing the specific volume of water. The entropy of the liquid, if the specific heat be taken as constant, is

$$log_e \frac{T}{t} = loge \frac{327.8 + 460}{32 + 460} = 0.469$$

The entropy of vaporization is

$$\frac{L}{T} = \frac{845}{327.8 + 460} = 1.075$$

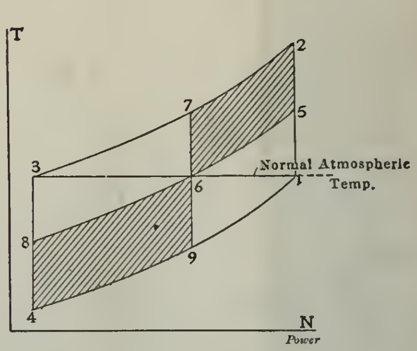
The total entropy of the wet steam is

$$0.469 + 1.075 = 1.544.$$

7. In the single-stage apparatus, we have adiabatic compression 12, cooling at constant pressure 23, to the initial temperature $T_1 = T_3$, expansion (adiabatic) 34, warming of the exhausted air from the engine at constant pressure along 41, so that it (or its equivalent) enters the compressor at normal atmospheric temperature T_1.

The inclosed area 1234 represents the net work expenditure, measured in heat units.

In two-stage compression, adiabatic compression ceases sooner, at 5, and the air is passed through an intercooler where its temperature is reduced to $T_6 = T_1$, at constant pressure along 56. It is then recompressed to the desired pressure along 67, the temperature at 7 being less than that at 2. It cools as before, after compression, along 73, before reaching the engine at 3. The area 7256 represents a saving of work at the compressor. Expansion in the two-stage engine along 38 is similarly stopped early, and the air allowed to regain atmospheric temperature at constant pressure along 86. A second expansion 69 then follows. The pressures at 4 and 9 are equal, but the temperatures are different. So also at



the points 7 and 2. The saving of work in the engine is represented by the area 8694.

8. We have given,

$V_2 = 1 \quad P_2 = P_1 = 14.7 \quad P_3 = P_4 = 100.$

Then since,

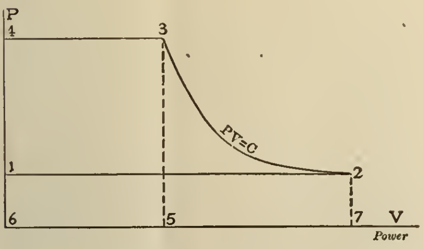
$$P_3 V_3 = P_2 V_2$$

$$V_3 = \frac{P_2 V_2}{P_3} = \frac{14.7 \times 1}{100} = 0.147$$

The area 4321 is the work per cycle; it is

$$6435 + 5327 - 6127 = P_3 V_3 + P_3 V_3 \log e \frac{V_2}{V_3} - P_1 V_2 =$$

$$144[(100 \times 0.147) + \left(100 \times 0.147 \log e \frac{1}{0.147}\right) - (14.7 \times 1)] = 144\,[14.7 + 28.1 - 14.7] = 4050 \textit{ foot-pounds}$$



This work might be about 60 per cent. of that necessary per cubic foot of free air in an actual compressor. A 2000-foot compressor, for example, should indicate about

$$\frac{2000 \times 4050}{0.60 \times 33000} = 409 \textit{ horsepower}$$

9 (b). $PV^{\frac{17}{16}} = c$, Rankine's equation for relation between pressure and volume of dry steam;

$H = H_{212} + a\,(t - 212) - b\,(t - 212)^2$, Davis' equation for total heat in dry steam;

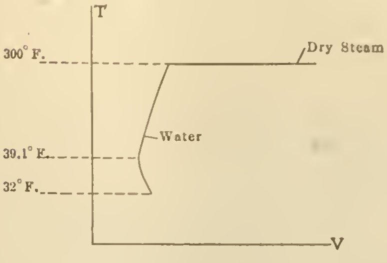
$e = \dfrac{P(W - V)}{778}$ external work of vaporization, in B.t.u., when

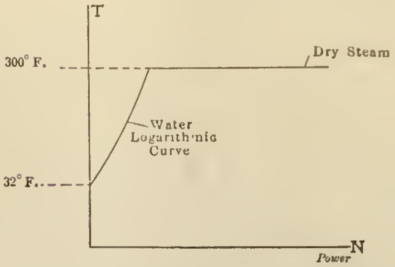
P = Pressure in pounds per square foot,

W = Volume of one pound of steam at pressure P,

V = Volume of one pound of water.

9 (a).





10. The factor of evaporation is the ratio of the member of B.t.u. imparted to the feed water in generating steam, to 970.4. From problem 6, the total heat of the wet steam, measured above 32 degrees, is 1140.8. Feed water at 212 degrees Fahrenheit already contains

$$212 - 32 = 180 \textit{ B.t.u.}$$

The amount of heat imparted to each pound of water is then

$$1140.8 - 180 = 960.8 \textit{ B.t.u.},$$

and the factor of evaporation is

$$\frac{960.8}{970.4} = 0.99$$

11. In the illustration, the expansion is from 1 to 2, and we have, from problem 6, entropy at

$$1 = 32 = 1.544.$$

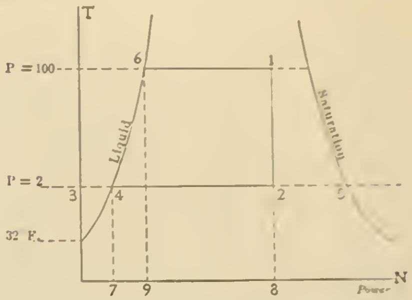
We have also given, entropy of liquid at

$$T_2 = 34 = 0.1749,$$

and entropy of vaporization at

$$T = 45 = 1.7431.$$

The dryness at 2 is



$$\frac{42}{45} = \frac{32 - 34}{45} = \frac{1.544 - 0.1749}{1.7431} = 0.783$$

12. In the previous illustration, the cycle is 4612. The efficiency is

$$\frac{\textit{Heat converted into work}}{\textit{Gross heat absorbed}} =$$

$$\frac{4612}{74618} = \frac{7469 + 9618 - 4287}{7469 + 9618} =$$

$$\frac{\left(\begin{array}{c}(\textit{Heat of liquid at } T_6 - \textit{heat of liquid at } T_4) \\ + (\textit{heat of vaporization of the wet steam} - \\ \textit{heat of vaporization of the steam at the} \\ \textit{end of expansion}\end{array}\right)}{\left(\begin{array}{c}(\textit{Heat of liquid at } T_6 - \textit{heat of liquid at } T_4) \\ + (\textit{heat of vaporization of the wet steam})\end{array}\right)}$$

The heat of vaporization of the steam at the end of expansion is not given; but we may compute it as follows: The area

$$4287 = 42 \times 82 = (32 - 34) \times 82 = (1.544 - 0.1749)\ (126 + 460) = 801 \textit{ B.t.u.}$$

The efficiency of the cycle is then, taking values from problem 6:

$$\frac{[295.8 - (126 - 32)] + 845 - 801}{295.8 - (126 - 32) + 845} =$$

$$\frac{201.8 + 845 - 801}{201.8 + 845} = \frac{245.8}{1046.8} = 0.235$$

Auxiliary Apparatus in Close Quarters

> *By using an engine of the Lentz type, which has a short base, and by employing a short-center rope drive, the Hammermill Paper Company was able to install auxiliary power equipment in very limited quarters.*

Seldom is it possible to do as one would like in arrangement and equipment when alterations are necessary. Every man who tackles a job of planning for enlargement of productive capacity and its consequent demand for increase of power knows at the outset that the problem involves comparison of various plans and a discussion of relative costs and economies before a decision can be reached.

Sometimes things work out all right and sometimes a great deal of squirming is needed before all the conflicting conditions are satisfied or compromised.

Recent changes that became necessary at the Hammermill Paper Company's mills at Erie, Penn., were made very satisfactorily and economically, yet at the expense of importing an engine of a type not yet being built in this country. The fact that a special engine was needed so badly as to justify the payment of a high import duty for its admission to the country is an indication of the difficulty of the problem.

A large cross-compound Corliss engine was the sole source of power for all of the mills. Its broad wheel carried the ropes of two English or multiple-loop rope drives, about two-thirds of the rope leading forward to the main line shaft and the other third running backward between the engine cylinders to drive a secondary shaft.

As additional machinery was gradually installed in various parts of the mill, the engine became overloaded and relief was necessary. The natural thing to do was to relieve the engine of the load imposed by the secondary shaft and leave it only the main line shaft to drive. This decision being made, the problem was to install within the available space and under existing conditions an engine and drive for the secondary line.

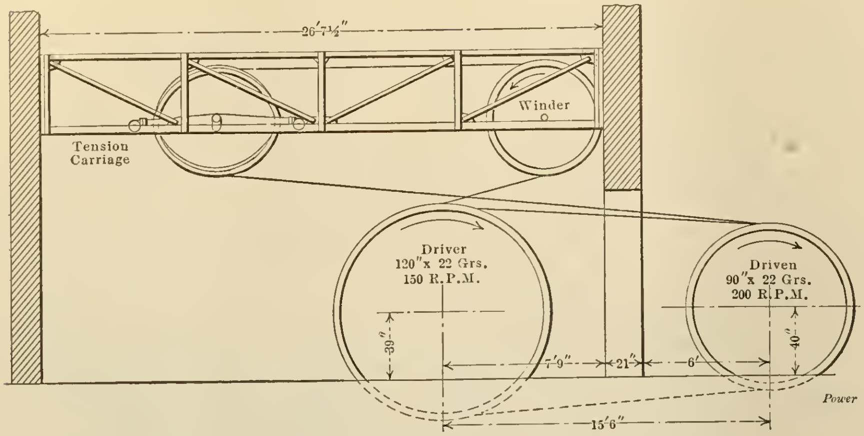


Fig. 1. Arrangement of Drive

No prime mover of inferior economy would be acceptable, and within the space available no economical engine of standard American type could be installed. Being familiar, through its German connections, with the Lentz type of engine, the Hammermill company decided to purchase and import an engine of this type and to make the connection with the line shaft through a short-center rope drive of the Dodge American system.

The facts of the case are summed up in the statement that under the conditions imposed, and to accomplish the results desired, the best and most economical solution of the problem was the combination of this extremely short type of engine and the Dodge rope drive, which operates with success on short centers. Fig. 1 is a diagrammatic section showing the arrangement of the rope-drive system, while Fig. 2 is a general view of the engine and the overhead tension carriage of the drive.

The distance on centers from the engine shaft to the line shaft is only 15 feet 6 inches, the sheave diameters, giving speed increase from 150 revolutions at the engine to 200 at the line shaft, are 120 and 90 inches respectively, leaving about 6½ feet of space in the clear between. The drive consists of 22 wraps of rope served by a tension carriage traveling on a horizontal track of steel construction.

The engine is a tandem compound with cylinder diameters, reducing metric measures to nearest inches, of 24 and 38 inches. The stroke is 28 inches. The steam pressure carried is 150 pounds; the steam is superheated 100 degrees. The engine runs condensing, and, although not operating under the most economical load, shows a steam consumption of only 11½ pounds per horsepower-hour.

The normal rating is 750 horsepower and the maximum 930 with ten atmospheres (147 pounds) initial cylinder pressure. The guaranteed economy is 13.34 pounds per horsepower-hour at ten atmospheres with dry and saturated steam or 11.68 pounds with steam superheated to 480 degrees Fahrenheit, or 116 degrees above saturation at ten atmospheres. The engine was imported complete with the exception of its flywheel, which was furnished by the Dodge Manufacturing Company. The wheel is of the plain balanced type cast in halves, joined by bolts at hub and rim, and by I keepers shrunk into the sides of the rim. The diameter is 12½ feet, the face width 12 inches and the weight 14 tons.

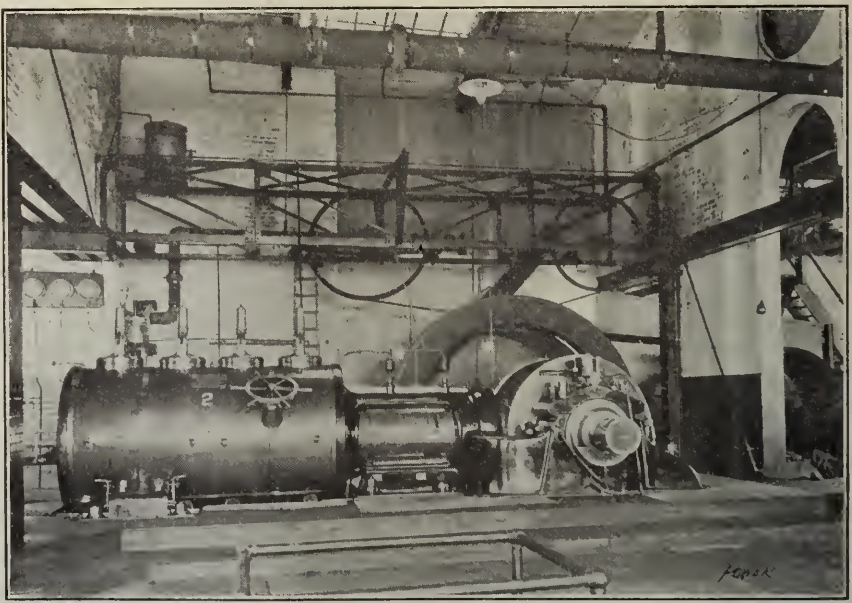


Fig. 2. The Lentz Engine and Dodge Rope Drive in Close Quarters

A continuous oiling system is installed for economical and automatic lubrication of the engine. Overhead in Fig. 2 may be seen a supply tank from which the oil feeds by gravity through suitable piping directly to the bearings.

Drippings drain to a filter in the basement, whence the purified oil is pumped to the overhead reservoir to be used again. The White Star continuous oiling system, which is used, effects a saving of from 50 to 80 per cent., as compared to ordinary hand oiling, while at the same time it improves the effectiveness of the lubrication.

The features of the engine are: extreme simplicity and compactness; greatly reduced number of parts in valves and valve gears; a very simple shaft-driven governor, which, if stopped for any reason, will shut the engine down instead of letting it run away; the possibility of running at a very high speed, up to 300 revolutions per minute or more without impairing the economy; and poppet valves which are seated quickly, yet without shock, and are particularly adapted to high pressures and high superheat, and which—most important of all—have no frictional surfaces, either sliding or rotary, and therefore need no lubrication and remain always tight. In Germany, a steam consumption of less than 10 pounds per horsepower is reported to have been attained with highly superheated steam.

The Lentz system has been adopted widely abroad and is used on locomotives and marine engines as well as in stationary practice. A large number of engine builders in Germany, Holland, Belgium, France, Italy and England are using the Lentz patents. The American rights have been acquired by the Erie City Iron Works, Erie, Penn.

Expansion in Steam Pipes

By W. L. Durand

The power-plant engineer is frequently called upon to compute the expansion in steam-pipe lines. For convenience in such cases a set of curves has been drawn, showing the expansion for different lengths of cast-iron pipe at varying temperatures. By multiplying these values by certain coefficients, the expansion of brass and wrought-iron pipes can be found.

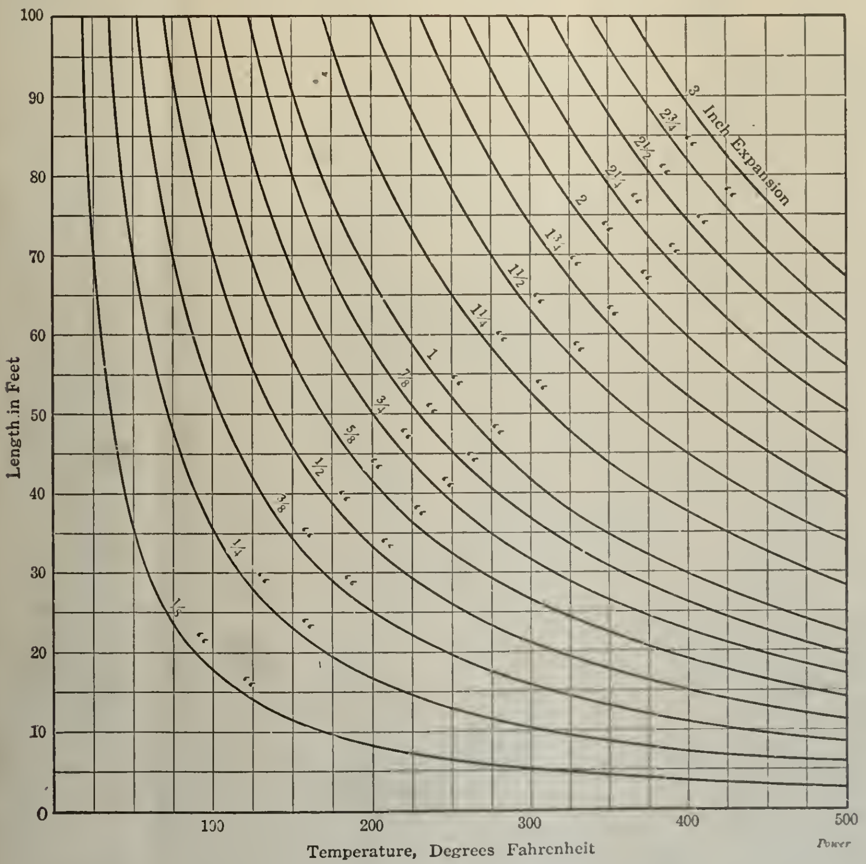
Among the many problems that confront the power-plant engineer is that of satisfactorily taking care of the expansion in steam pipes. In many of the newer plants using pressures of 175 pounds and 200 to 300 degrees superheat, this becomes a problem in itself.

From many experiments it has been found that wrought iron expands 0.0000066 per cent. for each degree Fahrenheit rise in temperature between 32 and 100 degrees and this amount increases until at 500 degrees it is 0.00000814 per cent. This increase in length is called the coefficient of expansion. The rule for finding the lineal expansion of any metal is as follows:

Expansion = length × increase in temperature × coefficient of expansion.

While this formula is very easy and simple to use, still one may not have at hand the various coefficients of expansion for varying temperatures, and if many problems are to be solved much time will be saved by the use of a set of curves similar to the accompanying chart. This is plotted for cast iron and the expansion for other metals may be found by multiplying the result taken from the chart by the ratio of their coefficients of expansion to that of cast iron. For wrought iron this is 1.1 and for brass 1.6. The curves are plotted from 0 to 100 feet and from 0 to 500 degrees Fahrenheit, for every ⅛-inch expansion up to 1 inch, and for every ¼ inch up to 3 inches. The following coefficients for the different temperatures were used in laying out the chart:

100 degrees	0.0000600
150 degrees	0.0000612
200 degrees	0.0000626
250 degrees	0.0000642
300 degrees	0.0000660
400 degrees	0.0000700
500 degrees	0.0000740



Curves Showing Expansion in Pipes

Suppose it is desired to find the expansion of a wrought-iron steam pipe 55 feet long when carrying steam at a pressure of 150 pounds gage, with the surrounding air at 70 degrees Fahrenheit. The temperature corresponding to 150 pounds is 366 degrees so that the rise in temperature will be 296 degrees. Referring to the chart we find that the expansion corresponding to 296 degrees and 55 feet is 1¼ inches. Multiplying this by 1.1 for wrought iron we find the expansion of the steam pipe to be 1⅜ inches.

The use of the chart may be greatly extended by finding the expansion for such part as will come within its limit and then multiplying this expansion by the ratio of that part to the whole. For example, if we want to find the expansion of a cast-iron pipe 800 feet long with a rise of 200 degrees in temperature, we find the expansion for 100 feet with a rise of 200 degrees to be 1½ inches, so that for 800 feet the expansion will be eight times as much or 12 inches.

Alaska Has Vast Deposits of Coal

According to Richard Hamilton Byrd, in the *World To-day*, the geological work of the Government in Alaska shows an aggregate of 8,106,880 acres of coalfields, and of these 769,280 acres have been surveyed in such detail as to warrant the statement of the Geological Survey, always highly conservative, that in these areas there is a reasonable degree of certainty that commercial coal beds can be opened up. In considering the coal underlying these 769,000 acres, the Survey, in a report to the National Conservation Commission, estimated a tonnage of 15,104,500,000 tons of minable coal.

"In view of the conditions," the statement concludes, "it is perhaps conservative to multiply the figures of 15,104,500,000 tons by ten or even one hundred to arrive at an approximation of the fuel resources of this vast, little explored region." And this does not take into consideration, apparently, the ninety million acres of totally unexplored territory, geologically, but in which coal is known to exist, and which, it is fair to assume, contains its share.

Great as is Alaskan gold reserve—the small Seward peninsula alone producing annually, and promising for many years to come, a sum equal to the purchase price of Alaska—the Territory's coal resource is of far greater value, albeit the latter will contribute immensely to the production of the former.

Furthermore, Alaska's stores of high-grade fuel cannot be equaled in quality west of the Rockies.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

The Electric Elevator

By William Baxter, Jr.

The Elektron System

(Continued)

The starting switch of the Elektron controller, partly described last week, is located within the controller box. It surrounds the operating shaft *S*, and consists of a number of copper disks mounted upon the shaft, but insulated from it and also from each other, and corresponding stationary contacts with which the disks come into contact whenever the shaft is turned to start the elevator. The construction of these disks and stationary contacts, and their location, can be understood from the drawings, Figs. 118 and 119. The first drawing is a view endwise of the shaft, and shows the outline of a disk with about one-quarter of it cut away, and also a shorter segment. An edge view of these is given in Fig. 119, which is an elevation of the upper part of the controller box, taken parallel with the shaft *S*. Two of these disks slide into stationary contact jaws that are connected to the main line, and are marked "Line disks" in Fig. 119; the central disk slides at the same time into the contact jaws marked *J J'* and thereby closes the circuit of the brake magnet.

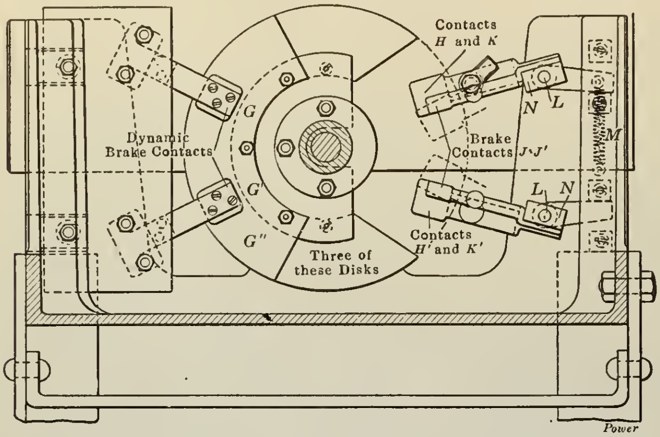


Fig. 118. End View of Starting Switch

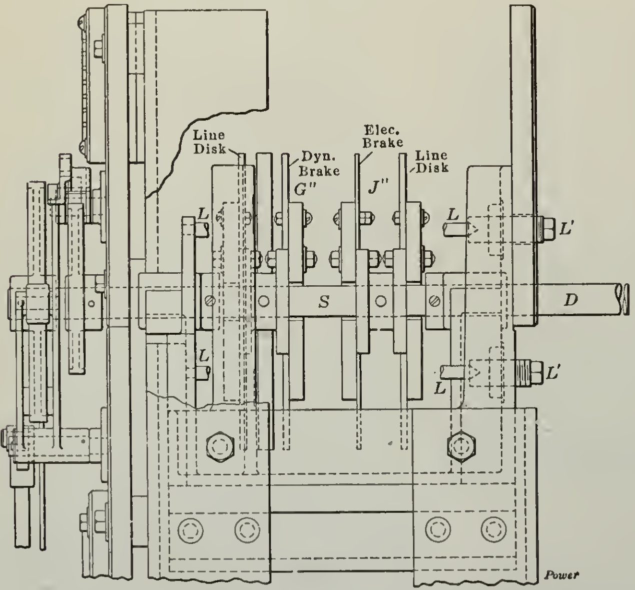


Fig. 119. Side View of Starting Switch

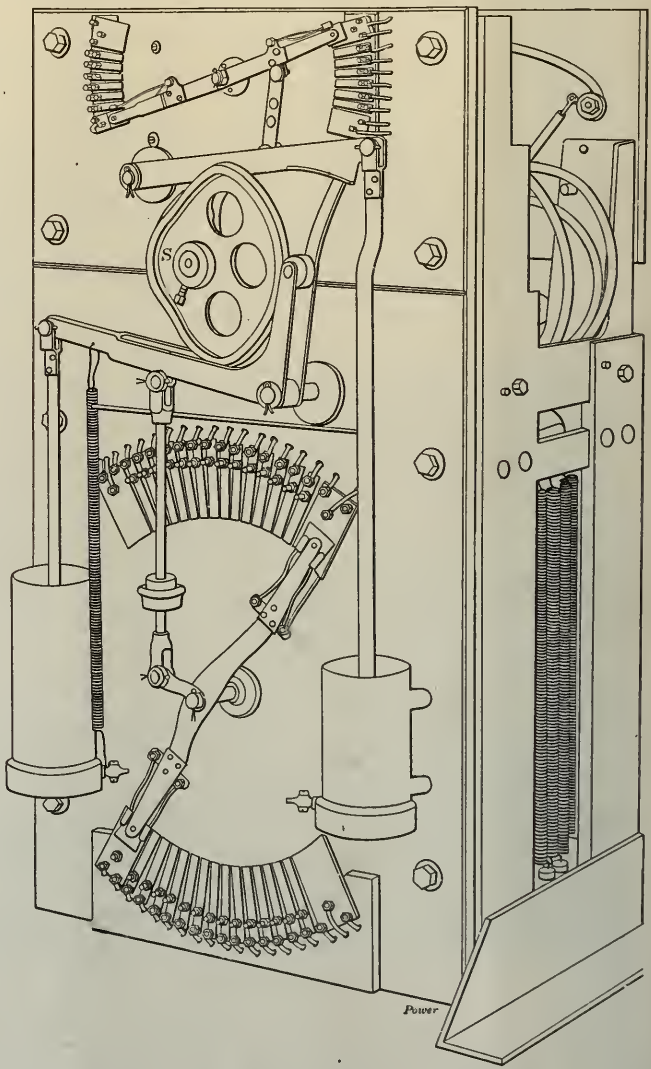


Fig. 120. Rear View of Controller

When the shaft is rotated in either direction, the first movement causes the three disks to slide into their respective jaws, thereby connecting the motor to the line and at the same time energizing the brake magnet. To close the line connections, however, it is necessary for the disks to rotate far enough to slide through the two sets of contact jaws marked *H, K* and *H' K'*, in Fig. 118, and this movement is more than is necessary to enable one or the other of the cams *D, D'*, Fig. 117, to throw the reversing-switch blades to the running position. Therefore the reversing-switch contacts are closed before the line switches are closed. The contacts *H* and *K* are arranged to swing around the spindles *L* upon which they are mounted, so as to compress a spring *M*; the result is that when the switch is opened, as the disks pull out of the jaws the latter are jerked away quickly, to break the flash abruptly.

The segment G'' in Fig. 118, upon which the contacts G and G' rest, closes a dynamic braking circuit around the motor armature when the motor current is cut off. The construction of the jaws H, K and J is shown in Fig. 120, and also the spindle L upon which they are mounted. The jaws are insulated from each other, the blocks N being made of insulating material.

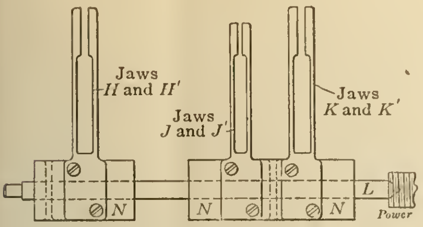


FIG. 121. CONTACT JAWS

The back of the controller illustrated in Fig. 114 is shown in Fig. 121. The parts at the bottom of the panel act to cut out the starting resistor, and those at the top act similarly to cut resistors into and out of the shunt field circuit to increase and decrease the speed of the motor. The operating shaft S, which runs through to the back of the controller box, carries on its end two cams, one of which actuates the lower switch lever, that cuts out the starting resistor, and the other the upper lever, that controls in the field resistor. These cams are so set that all of the starting resistor is cut out before any of the field resistor is cut in, and means are provided to prevent the apparatus from acting in any other way. When the operating shaft S is rotated to the stop position, the cam is in such a position that it holds the switch levers in the stop position; that is, the lower switch lever is rotated clockwise as far as it will move, cutting into the armature circuit all of the starting resistor coils, and the upper lever is moved counter-clockwise as far as it will go, in which position it cuts out all the resistor in the field circuit. The two levers are shown in the stop position. When the operating shaft is rotated in either direction, the cams act first to release the lower switch lever, and shortly thereafter to release the upper lever. When released by the cams the levers are drawn over their switch contacts by the weight of cam levers and their parts, and two dashpots are provided to regulate the rapidity with which the levers move. The action of these switches may be more clearly understood by reference to Fig. 122, which is a face view of the mechanism. It will be noticed that cam A can rotate through a considerable angle; that is, until its center line reaches the position F or F', before the roller A' can move, because this part of the cam is a circular curve concentric with the shaft. The cam is made this shape so that the starting and reversing switches may be closed before A' moves. When the cam moves through the angle from F'' to G, the roller A' can move to the left far enough to permit the lever B to drop and carry the switch lever L over to the extreme left position.

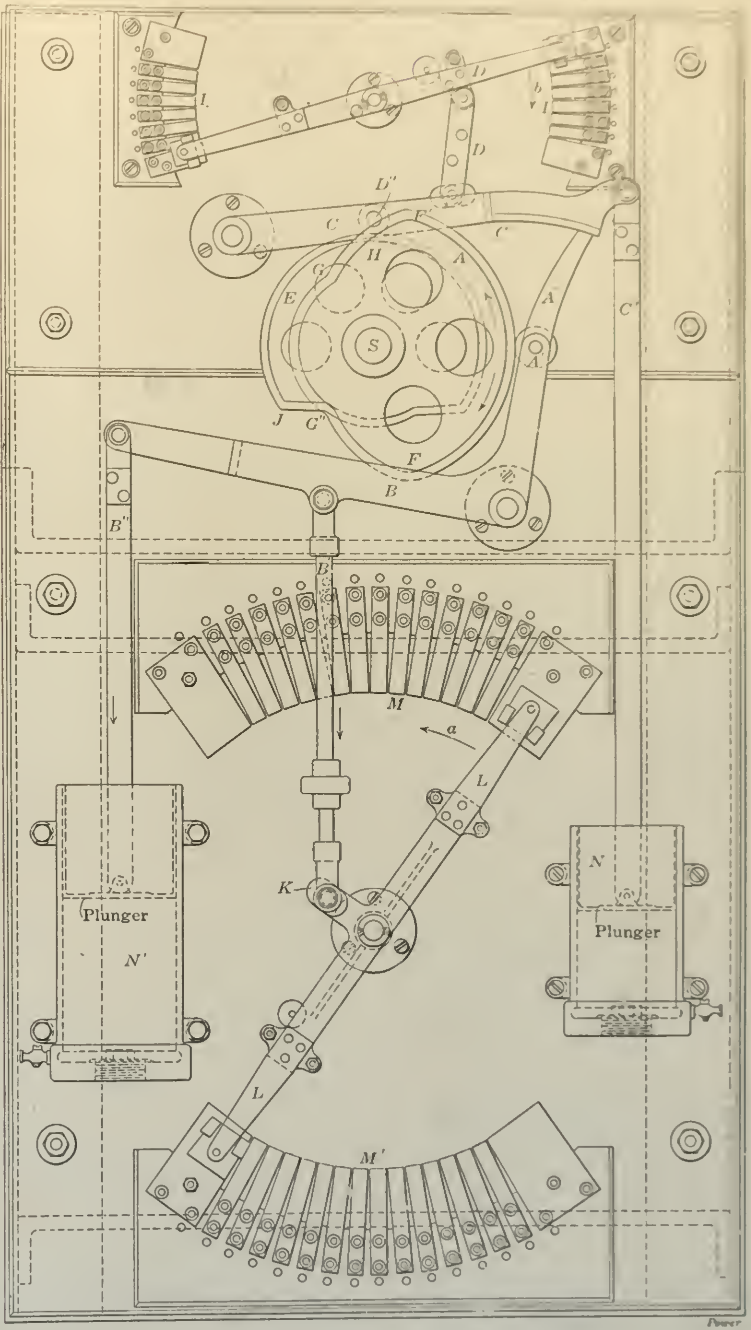


FIG. 122. ACCELERATING SWITCHES

The lever C carries a roller D'' that rides on the cam E, and the circular part of this cam covers such an angle that when the roller A' reaches the point G on the cam A, the roller D'' reaches the point J of the cam E; consequently, when the cam A has rotated far enough to permit the lever B to swing L around to its extreme position, the cam E will be far enough around to permit the roller D'' to run down the incline from J to G, and permit the switch lever D' to drop in the direction of arrow b to the lowest position. The lever C, however, cannot drop when the cam E rotates far enough for the roller D'' to run down the incline from J to G, unless the upper end A'' of the lever B has moved far enough to the left to carry it beyond the curved surface C' on the end of the lever C. From this it will be seen that the extension A'', in connection with the curved plate C' on the lever C, constitutes a locking device to prevent the field resistor from being cut into the circuit until after all the starting resistor has been cut out of the armature circuit. This provision is necessary because the velocity with which the lever *B* descends and moves the lever *L* over the switch contacts is controlled by the dashpot *N'*, and it is possible for the operator to rotate the operating shaft *S* so rapidly that the cam *E* will free the lever *C* before the lever *L* has moved around to the extreme left-hand position. The curved plate *C'* on the end of the lever *C* is not directly under the lever but on one side, so that when the arm *A"* passes to the left of the face *C'* the lever is free to drop as fast as the dashpot *N* will permit it.

In starting the elevator, if the operator desires to get the car under full headway as rapidly as possible, he rotates the operating shaft *S* to the full running position at once; the starting resistor is then cut out of the armature circuit as rapidly as the dashpot *N'* will permit the switch lever *L* to swing around to the left, but during this time the switch lever *D'* is held stationary because the end of the arm *A"* is under the curved plate *C'* on the lever *C*. As soon, however, as *A"* passes to the left of *C'*, the switch lever *D'* begins to descend and cuts in the field resistor as fast as the dashpot *N* will permit. When the lever *L* first reaches its extreme position, the motor runs at its slowest speed, but immediately thereafter the speed begins to increase because the lever *D'* descends and the switch gradually cuts the resistor coils into the circuit of the shunt field winding. If the controller were moved only far enough to rotate the cams *A* and *E* far enough to carry the lever *L* over to the extreme left position, but not far enough to allow *A"* to free the lever *C*, then, with the parts held in this position, the elevator would speed up to its slowest velocity only; upon moving the controller shaft *S* the balance of the distance, the lever *C* would be released and the switch lever *D'* would cut the field resistor into the circuit, increasing the motor speed to its maximum. By operating the controller as described, the motor would operate as a two-speed machine.

With this controller also the operator can start up with different rates of acceleration, within reasonable limits. If he moves the operating wheel or lever in the car more slowly than the dashpot *N'* will permit the lever *L* to move, he will start the car more slowly than if he moves the wheel around to the running position in one rapid sweep. Another advantage is that in stopping, the speed of the motor is first reduced by the cutting out of the field resistor by the upward movement of the lever *D'*, the starting resistor being cut in after the speed has been reduced; then the motor is stopped by opening the main line and applying the brake. The advantage of reducing the speed before stopping is that more accurate stops can be made automatically at the top and bottom landings.

Electrical Equipment of Rail Roads*

By George Westinghouse

It is my intention in this paper to direct attention to the necessity for the very early selection of a comprehensive electrical system embracing fundamental standards of construction which must be accepted by all railway companies in order to insure a continuance of that interchange of traffic which, through force of circumstances, has become practically universal, to the great advantage of transportation companies and of the public. I feel that the time is ripe for such a selection unless we are willing to regard with complacency the extension of the existing diversified systems and the creation of conditions which will prevent the general use of the most practical methods of operation.

Indeed, the tendency seems to be toward diversity rather than unity, since different types of third-rail construction have been adopted, even for the several direct-current systems in and about New York City, which renders interchange of cars or locomotives difficult or impossible.

In the first days of railway operation, there was probably no idea of an interchange of traffic involving the use of the engines and cars of one railway upon the lines of another railway. It then made no difference whether the gage of track was 4 feet 8½ inches, the one ultimately selected, or one of a greater or lesser width by a few inches. However, the absolute necessity for uniformity of gage of tracks, both in the United States and Canada, became so apparent that in due course all of the roads which had gages wider than 4 feet 8½ inches changed to that standard. Among the remarkable achievements of engineering was the change of the tracks of an entire system of railway of some hundreds of miles within twenty-four hours, this change having, however, required months of preparation. The losses entailed in the change of gage and of equipment have ever since been serious burdens to most of those railways, in that the costs were in most cases covered by capital charges.

So far as steam railroads are concerned, there are now no obstacles to the interchange of traffic in the broadest sense, except in the size of vehicles in certain countries where the cost of changing tunnels and bridges would be prohibitive.

Requirements for Interchange of Traffic

To insure interchange of traffic, the fundamental requirements, so far as operation by steam is concerned, with full regard for safety, speed and comfort, are the following: A standard gage of track; a standard or interchangeable type of coupling for vehicles; a uniform interchangeable type of brake apparatus; interchangeable heating apparatus, and a uniform system of train signals.

The additional fundamental requirements for electrically operated railways are a supply of electricity of uniform quality as to voltage and periodicity; conductors to convey this electricity so uniformly located with reference to the rails that, without change of any kind, an electrically fitted locomotive or car of any company can collect its supply of current when upon the line of any other company; uniform apparatus for control of electric supply whereby two or more electrically fitted locomotives or cars from different lines can be operated together from one locomotive or car.

Electrical Systems for Railways

In the past twenty years three important electrical systems for the operation of railways have been put into practical operation, all using alternating current in whole or in part. These systems are the direct-current system, usually employing the "third-rail" conductor and alternating current for transmitting power when the distance is considerable; the three-phase alternating-current system with two overhead trolley wires, and the single-phase alternating-current, high-tension system with a single overhead trolley wire.

The Motors

In a notable case of the latter system, namely, that of the New York, New Haven & Hartford Railroad, the motors and controlling apparatus are arranged to utilize single-phase current from an overhead trolley wire at 11,000 volts, and also to be operated by current from the 650-volt third-rail system of the New York Central & Hudson River Railroad.

The equipment of the power houses which generate the current is essentially similar in the three systems which I have enumerated; but the systems differ in the kind of motors and the auxiliary apparatus for controlling them, and in the methods and apparatus for transmitting the current from the power house to the locomotive or car.

The three types of electric motors have certain fundamental differences in speed performance which are important factors in determining the advantages, disadvantages and limitations of the several systems.

The direct-current series railway motor automatically adjusts its speed in accordance with the load, running more slowly if the weight of the train be greater, or the grade steeper. The speed with a given load, however, is definite; it is dependent upon the voltage applied to the motor and cannot readily be varied. It is true that the speed can be decreased by inserting a resistance in the motor circuit, but this is wasteful and is inadmissible except as a temporary expedient. It is true also that the motors may be connected in series, thereby dividing the pressure between two motors and reducing the speed one-half; or, if four motors, to one-quarter speed. As the system of current supply involves a fixed voltage, it is obvious that for emergencies no speeds much above the maximum speed determined in the construction of the motor can be obtained. Furthermore, on account of the high cost involved in maintaining a practically constant voltage throughout the system, the voltage supplied to the motors often decreases considerably at the end of long lines, at the time of heavy load, thereby further reducing the speed attainable. It often happens in railway service that a locomotive should be operated somewhat above the normal speed, and sometimes a locomotive designed for freight service has to be pressed into passenger service. In such cases the speed with the direct-current locomotive would be considerably less than that necessary to maintain the schedule speed. A special form of field control can be used, in certain cases, for varying the speed, but this has not been utilized very much.

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*Abstract of a paper presented before the American Society of Mechanical Engineers.

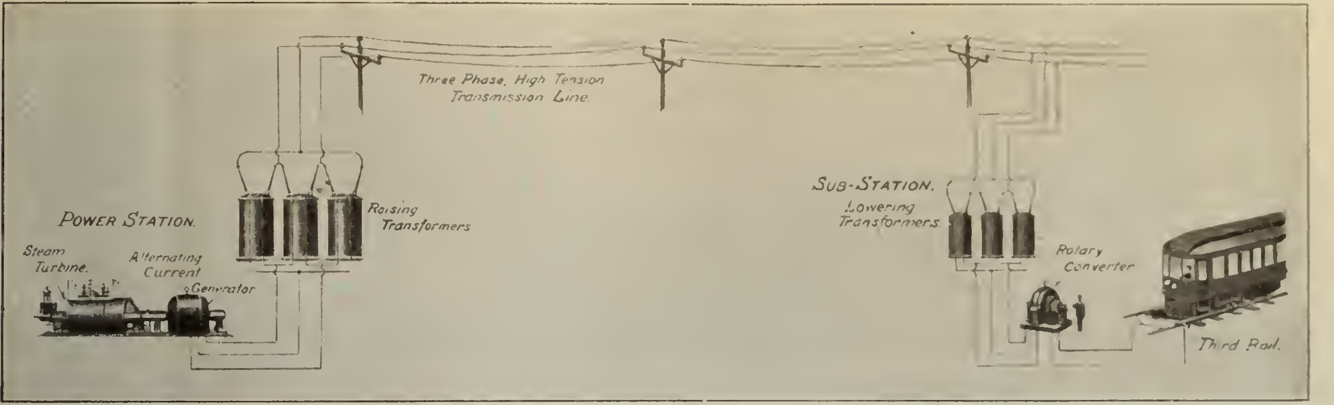

Fig. 1. Direct-current Railway System

The three-phase motor is inherently a constant-speed machine; it runs at approximately the same speed at light load and at full load; it runs at nearly the same speed up a grade as on level track, although the horsepower required on the grade may be several times that on the level. Conversely, it can run no faster on a level than it can climb a grade. In order to give a lower speed, however, the motors may be arranged upon the locomotive in pairs in a manner equivalent to the arrangement of two direct-current motors in series, just described. Motors may also be built for two or more speeds, but this involves some complication in windings and connections. In all cases lower speeds can be secured by the introduction of resistances which increase the losses and lower the efficiency. In no case can the speed in any of the arrangements of motors be appreciably higher at light load than at full load.

The motors are of the induction type without commutators and their inherent limitations, and are of relatively simple construction. The current is usually supplied at 3000 volts from two overhead lines through two sets of "trolleys." With three-phase motors as now constructed and arranged upon locomotives, it is possible with no additional complication so to utilize the motors when running down grade that they become generators and return power to the line, a feature of value in certain mountainous districts but not of controlling importance in the selection of a universal system.

The single-phase railway motor is a series-wound machine with speed characteristics very similar to those of the direct-current motor, as the speed at a given voltage is greater or less, depending upon the load. The speed with a given load is also greater or less, depending upon the pressure applied to the motor; and this is not limited, as with direct-current motors, to that supplied by the circuit, and to one-half and one-fourth of that pressure, but is capable of adjustment to any desired degree of refinement by means of auxiliary connections from the secondary winding of the transformer (on the locomotive) which is necessary for reducing the line voltage to the lower voltage required by the motors. Not only may numerous voltages less than the normal be obtained for lower speeds, but higher voltages can be provided to give speeds considerably above the normal. In this manner a wide range of efficient speed adjustment is secured which is impossible with other systems.

Like the throttle lever of the steam locomotive, the control lever of the single-phase locomotive may be placed in any one of its numerous notches to maintain the required speed. This facility of efficient operation over a wide range of speed and power requirements is one of the especially valuable features of the single-phase system.

The question of determination of the frequency for use on single-phase railways is one of very great importance. Twenty-five cycles are in general use for power-transmission purposes and have been adopted by nearly all the single-phase railroads now operating. The Midi Railway of France has adopted 15 cycles. The lower frequency permits of a marked reduction in the size of a motor for a given output, or conversely of a considerable increase in output from a motor of given dimensions and weight. Three-phase installations in nearly all cases employ approximately 15 cycles.

Locomotives equipped with each of the three types of motors mentioned are in successful operation and have demonstrated their usefulness and reliability in practical railway service. The three-phase motor, having a definite constant-speed characteristic, is particularly adapted to certain conditions; but on the other hand, it has a less general adaptability to the ordinary varying conditions of railway operation. The single-phase motor has a facility of voltage control which gives an efficient means of speed adjustment, and is in this particular superior to other systems. The relative weights and costs of the several types of motors, and of the locomotives designed to accommodate them, depend upon so many conditions that comparisons must necessarily be general. It will be found, however, that these differences in locomotive cost are in many cases more than offset by the cost of the other elements in the electrical system.

Transmission of Power from Power House to Locomotive

The controlling factor in the cost of electrification in nearly all cases is the system for transmitting power from the power house to the locomotive, and not the locomotive itself. The choice between the several systems must, therefore, be based upon a comparison of the complete systems. The differences between the methods of transmitting power are of far greater importance than the differences between power houses or between locomotives. The current for all systems is generated in usual practice as high-tension alternating current, for the reason that electric energy can be most economically transmitted by high-tension alternating current.

The transmission systems for the three types of locomotives are illustrated in Figs. 1, 2 and 3. Even a superficial glance at these diagrams gives the following comparison:

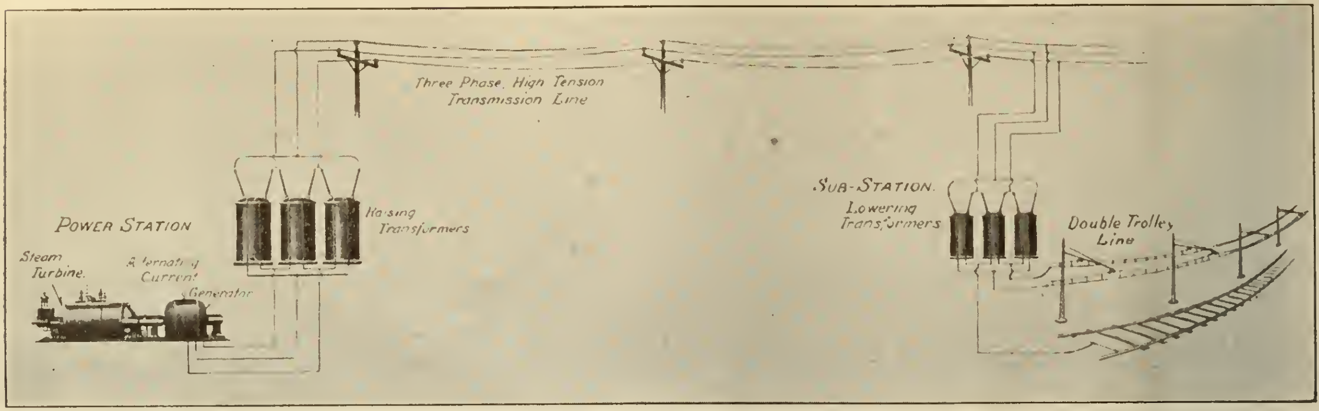

Fig. 2. Three-phase Railway System

In the direct-current system the apparatus which intervenes between the alternating-current generator and the locomotive consists of step-up transformers in groups of three; a transmission line of three wires; transformers in groups of three, and rotary converters for changing the alternating current to direct current; a third-rail contact conductor, which for heavy work must often be supplemented by copper feeders; the track-return circuit, which must be provided with heavy bonds, and in certain cases supplemented by feeders and so called negative boosters.

For the three-phase system the links between the generator and the locomotives are step-up transformers in groups of three; a transmission line of three wires; substation transformers in groups of three; two overhead wires as the contact system; a track return which usually requires only inexpensive bonding.

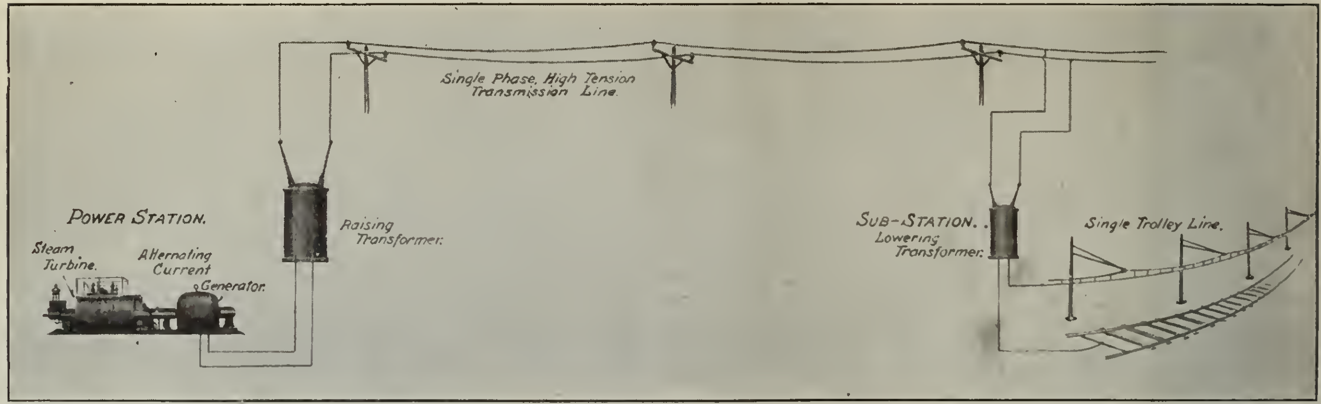

Fig. 3. Single-phase Railway System

For the single-phase system there are required a raising transformer; a transmission line of two wires; a lowering transformer in each substation; a single trolley wire; a track return, usually requiring only inexpensive bonding.

In certain cases where the distance from the power station is not more than 15 or 20 miles, the single-phase trolley wire can be supplied directly from the power house, with no transformers between the generators and the locomotives.

Requisites for a Universal Electric System

In selecting a proper electrical system for railway operation, it will probably be generally conceded that the following elements are of prime importance:

The electric locomotive should be capable of performing the same kinds of service which the steam locomotive now performs. This will be most readily secured by electric locomotives which can practically duplicate the steam locomotives in speed and power characteristics. This naturally calls for wide variation in tractive effort and in speed, both for the operation of different kinds of trains, and also for the operation of the same train under the varying conditions usually incident to railway service.

The electric locomotive should be able to handle heavier trains and loads, to operate at higher speeds, and in general to exceed the ordinary limits of the steam locomotive in these regards. The readiness with which several electric locomotives can be operated as a single unit enables any amount of power to be applied to a train.

The electric system should adapt itself to requirements beyond the ordinary limitations of the steam locomotive in small as well as large things. It should be adapted for use on branch lines, and for light passenger and freight service similar to that so profitably conducted by interurban electric roads, which in many cases run parallel to steam roads, not only taking away the traffic of the steam roads, but building up a new and highly profitable traffic, both in passenger and in express service.

A universal electrical system requires that power should be transmitted economically over long distances, and that the contact conductor should be economical to construct, both for the heaviest locomotives where the traffic is dense and for light service on branch lines. It should impose minimum inconvenience to track maintenance; should give minimum probability of disarrangement in case of derailment, or in case of snow and sleet, and should in general be so placed and constructed as to give a maximum assurance of continuity of service.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Elementary Lectures on the Gas Producer*

By Cecil P. Poole

The Wet Scrubber

After passing through the economizer, or upon leaving the generator, if the economizer and vaporizer are of the "built-in" sort represented in Fig. 3, the gases are carried through an apparatus known as the "wet scrubber," in order to cool them further and also to cleanse them of minute particles of dust, ash, grit, etc., that they have picked up in the fuel bed of the generator. The wet scrubber is a tower-like structure almost completely filled with either crushed coke or wooden grids and containing a water spray at the top which keeps the coke or the grids covered with a film of cool water. The gases in passing through the small spaces between the lumps of coke, or the openings in the wooden grids, are broken up and forced to come into contact with the water-covered surfaces. When they have passed through the "filling" of the scrubber they pass finally through the water spray, which tends to wash out the particles of objectionable material. Fig. 5 shows a vertical section of a coke-filled scrubber and Fig. 6 is a corresponding section of a lattice-work scrubber. The gases from the economizer enter the bottom of the scrubber, where a pool of water is maintained by the descent of the water from the spray at the top. The mouth of the gas pipe is submerged in the water, so that the gases must pass through the water before ascending to the coke or grids; this removes the larger particles carried over by the gases and reduces the tendency to clog up the "filling" with accumulations of these larger particles. The gases then pass up through the filling and the spray, as just explained, and go to what is known as the "dry purifier," indicated at *P*, Fig. 5.

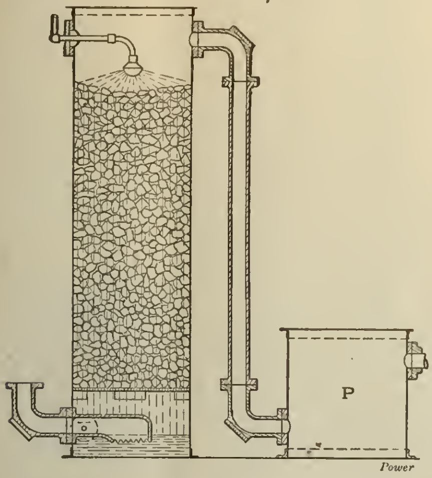

Fig. 5. Coke-filled Wet Scrubber

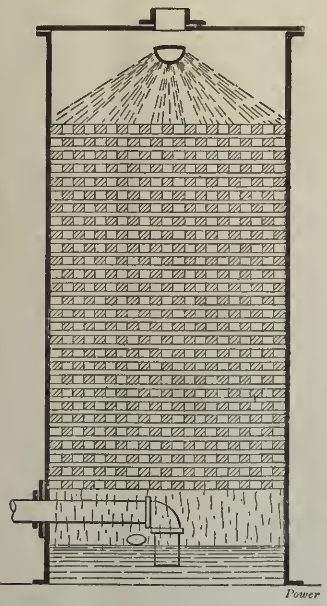

Fig. 6. Lattice-filled Wet Scrubber

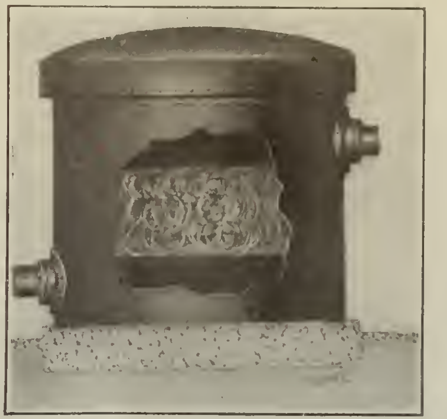

Fig. 7. Dry Purifier

The Dry Purifier

The dry purifier is merely a mechanical filter consisting of a box or tank filled with excelsior, sawdust or some similar material which removes the moisture that the gas has picked up in the scrubber and also tends to take out any solid particles of matter that have escaped the filtering material and water in the scrubber. Fig. 7 illustrates the usual construction of the dry purifier. The shell is provided with a sort of grating or perforated shelf near the bottom, which supports the excelsior, and the gases are admitted to the compartment beneath this; they are compelled to pass through the mass of excelsior to reach the outlet pipe, which is located above the excelsior.

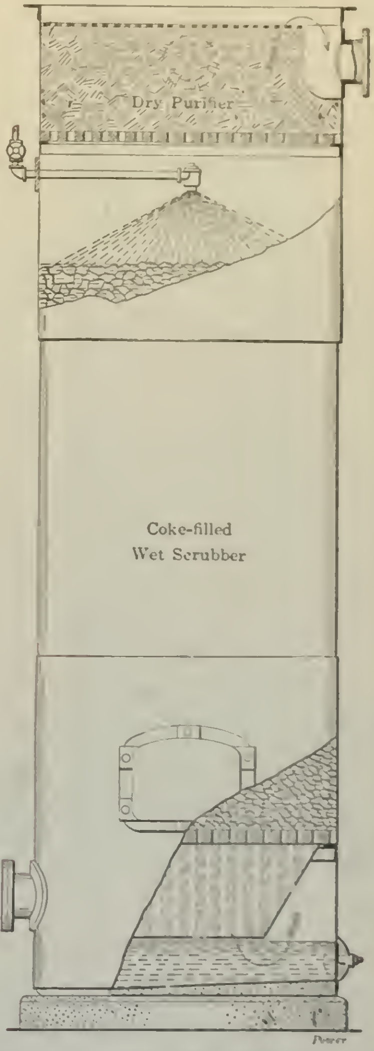

Fig. 8. Combined Wet Scrubber and Dry Purifier

Combination Cleaners

In some cases the wet scrubber and dry purifier are combined in one structure, as represented in Fig. 8, where the dry filling is supported on a grid above the water spray in the top of the wet scrubber. This has the advantage of economy in floor space and the disadvantage of making the dry purifier more difficult to get at for cleansing the filling.

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*This is the second lecture of the series. The next lecture will deal with the principles of gasifying solid fuels.

In other cases the scrubber structure contains three different grades of cleansing material or fillings, a coarse filling near the bottom, a finer filling in the central part and the usual dry-purifier filling at the top. Fig. 9 illustrates this construction. The bottom filling is a wooden lattice work, the central filling is of crushed coke, divided into two sections, each having its own water spray, and the top filling is that of the ordinary dry purifier.

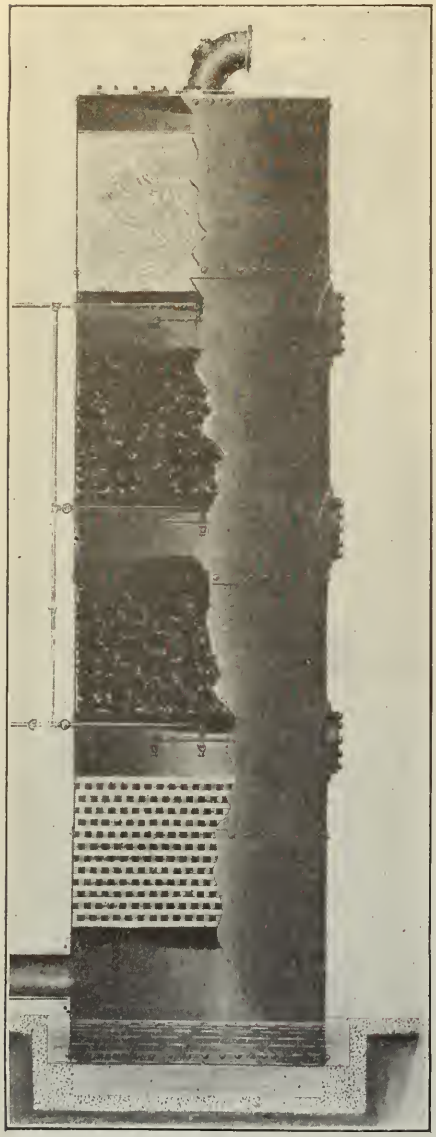


FIG. 9. COMPOSITE WET SCRUBBER, COMBINED WITH DRY PURIFIER

THE COMPLETE PRODUCER PLANT

In order to give the student a better grasp of the relations between the various members of the producer equipment, a complete plant is illustrated in Fig. 10, where the generator is indicated by the letter *G*, the scrubber by *S* and the dry purifier by *P*. The air intake is at *A*, the transfer pipe from the vaporizer to the ashpit, at *B*, and the air preheater or economizer at *E*. The hand blower *J* is for the purpose of supplying air to the fire in the generator at starting, when there is no other means of producing a draft, and the pipe *K* serves as a temporary chimney through which the gases formed by burning the coal escape to the atmosphere while the fire is being worked up. This pipe is commonly called the "purge" pipe, because it purges the producer of the gases made while the generator is being brought up to the regular working condition; these gases are not fit for use in an engine. A valve in the pipe is closed, shutting off the exit of the gases, when the generator has reached the proper condition for supplying gas to the engine. The generator shown here is equipped with a built-in vaporizer and an external economizer somewhat on the order of the one illustrated in Fig. 4.

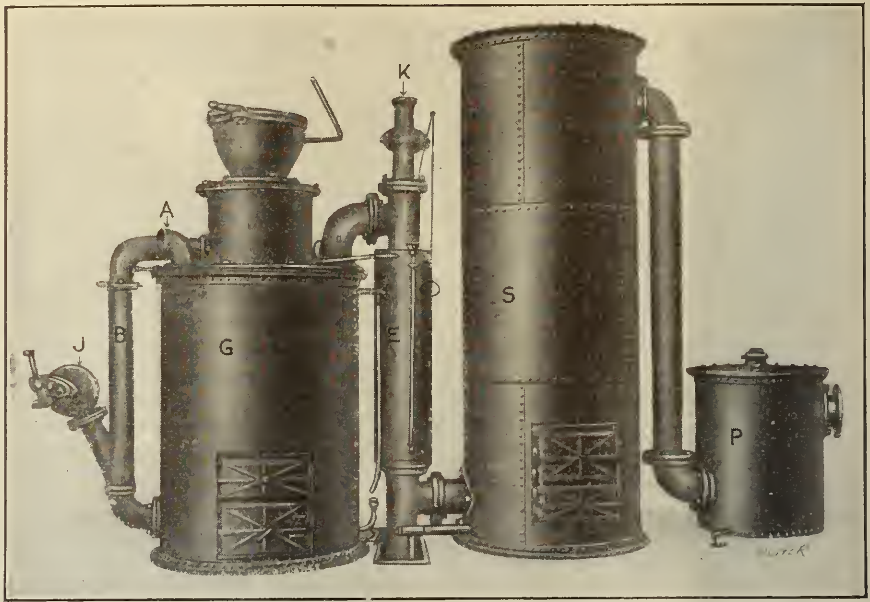


FIG. 10. A COMPLETE SUCTION PRODUCER PLANT

All of the apparatus thus far illustrated and explained belongs in the class of straight suction producers, in which the draft through the system is supplied by the suction of the engine piston during the intake strokes. The induced-draft or exhauster type of producer does not differ in any important detail from the general forms of construction here shown, but it has, in addition, a power-driven exhaust fan coupled to the outlet of the dry purifier to draw the air and steam into the generator and the gases out of it, thus relieving the engine of that work and keeping the gas under a slight pressure at the engine intake.

CORRESPONDENCE

Some Probabilities of the Gas Turbine

In Professor Heck's very able and interesting article on the "Inherent Difficulty of the Gas Turbine," published in the issue of May 3, the author points out that, whereas the constant-pressure cycle has the same theoretic efficiency, for the same degree of compression, as the constant-volume cycle of the ordinary gas engine, yet it involves a repeated handling of the gases at their minimum volume. This is a double task which the ordinary engine does not have to perform at all. Its compressed gases remain in the same locality during compression, combustion and expansion.

This repeated handling, in the constant-pressure cycle, necessarily involves increased cost of cylinder volume, increased friction and increased heat loss; and I believe that their relative importance is about in the order named.

Hence, if it is to be assumed that the gas turbine of the future must operate on the constant-pressure cycle, this appears to settle the possibility of the gas turbine competing with the piston gas engine. This is, I believe, Professor Heck's position, broadly speaking. Its truth the writer has learned from costly experience in building constant-pressure gas engines. But there is much more to the question than this, for the foregoing hypothesis rests upon three assumptions, none of which has yet been proven to be necessary.

First Assumption: That the degrees of compression must be equal for the constant-volume and constant-pressure cycles. This is manifestly unfair to the constant-pressure cycle, for if the constant-volume cycle carries compression only to 100 pounds, say, it nevertheless imposes upon the engine structure a maximum pressure of, say, 350 pounds, and the entire machine must be made strong enough to stand this stress. Nor, if the engines are to expand to the same terminal pressure, is there any difference in maximum volumes to compensate. Manifestly, therefore, the two cycles should be compared upon the basis of *equal maximum pressures.*

Upon this basis, the constant-volume cycle (that of the ordinary gas engine) would have a theoretic efficiency of 44.5 per cent. while that of the constant-pressure cycle would be over 60 per cent., assuming the pressure stated above. If the actual efficiency of the cycles be taken as naturally proportional to these figures, that of the constant-pressure cycle would be over 35 per cent. greater than that of the constant-volume cycle.. In other words, the constant-pressure engine might suffer practical losses 26 per cent. greater than those of the ordinary engine, and yet be of net efficiency equal to it. Indeed, the comparison can be stated more strongly yet. If the actual efficiency of the ordinary engine be one-half its theoretic efficiency, out of every 100 B.t.u. received it would actually convert 22 into work, losing also 22 of its theoretic possibilities, 44. But the theoretic possibilities of the constant-pressure cycle being 60 B.t.u., it might waste 38 of them and still come out just even with the ordinary engine. There is strong probability that it may be practicable to build within this limit.

Second Assumption: That the gas turbine must necessarily operate upon the constant-pressure cycle. This question cannot be discussed adequately without going into various constructive problems not in place here. But I wish to record my belief that the successful gas turbine will not, at least usually, utilize the constant-pressure cycle.

Third Assumption: That a gas turbine cannot overcome the difficulty of handling unusually large volumes any better than can a piston gas engine. In view of the phenomenal way in which the steam turbine has opened up the utilization, with great efficiency, of volumes so great that the piston steam engine is overwhelmed thereby, it is scarcely necessary to call attention to the promise of the gas turbine in the same direction. Again, however, I wish to record my belief that the gas turbine will not only be able to handle large volumes with sufficient ease to overcome the fundamental objections cited by Professor Heck—which, as I said, are real and important objections—but is very likely also to warrant the extension of terminal pressures into subatmospheric regions.

While we cannot, of course, literally "condense" gas-turbine exhausts, as we do steam, into volumes several hundred times smaller than before, yet the application of cold at lower pressures than atmospheric may none the less prove profitable. For it must be remembered that the piston gas engine suffers at the top of its cycle the same degree of inferiority to the steam engine which it does at the bottom. That is to say, at the top of the cycle the gas engine gets only a doubling or trebling of volumes or pressures of the "feed"—the charge pumped to top pressure. Yet on this very slim margin of profit, compared to the thousandfold increase in volume of feed enjoyed by the steam engine, it competes with the latter most successfully. Properly handled, that is, by a turbine, there is no fundamental reason why it may not ultimately do as well at the lower end of the cycle.

Neither the difficulties in expanding hot gases adiabatically in nozzles, reported by Doctor Lucke, nor the cyclic obstacles referred to by Professor Heck appear to me to constitute competent reasons for doubting the construction, in the near future, of a successful gas turbine. As to the often mentioned difficulty of getting a turbine to withstand flame, or near-flame, temperatures, I believe that to be quite irrelevant. The successful gas turbine will handle fluids no hotter than those in a steam turbine using moderately superheated steam. Considering the difficulties due only to temperature, which we experience in the use of even these standard machines, it seems to me futile to talk of building a turbine to stand gases from 1000 to 2000 degrees hotter. And it is quite unnecessary to try.

SIDNEY A. REEVE.

New York City.

Foundation Stresses

In a recent issue of this paper under the heading "Horizontal vs. Vertical Gas Engine," John S. Leese gives some calculations and figures on stresses to which an engine foundation is subjected, with which I cannot agree. Mr. Leese says:

"All forces have equal and opposite reactions, and a mean pressure of 80 pounds per square inch acting on a 24-inch piston in a single-acting horizontal engine has to be resisted by

$$\frac{0.7854 \times 24^2 \times 80}{2240} = 16 \text{ tons}$$

practically acting continuously on the foundation; or, if the explosive pressure be 320 pounds per square inch, 64 tons is the resistance needed."

Certainly "all forces have equal and opposite reactions," but it is hard to see how the reaction of the explosion ever should take place in the foundation. The engine frame alone is subject to these tremendous strains which are the results of the explosion, and it is at the crank shaft, where the opposite reaction occurs and where it is turned into useful work. But even though the foundation has not to stand as heavy a strain as indicated in Mr. Leese's article, it is nevertheless taxed to its limit by two minor reactions, and the local condition of the ground should always be made an object of careful study, where the erecting of a large gas engine is contemplated.

Take the above example of a 24-inch, single-cylinder, horizontal engine, which, I judge, must be about a 150-horsepower engine running at 140 revolutions per minute. The acceleration forces can be estimated closely enough by taking results obtained from tests on similar engines; in this case they will amount to about 50 pounds per square inch of piston area, hence the total force will be $0.7854 \times 24^2 \times 50 =$ about 22,600 pounds Of course, accurate calculations can be made only when all the details of the moving parts are at hand. This force would be acting continuously on the foundation, if the engine were not balanced at all, which is sometimes the case with vertical engines, while horizontal engines are always provided with balance weights of some kind. But as there is usually only a small space available between the main bearings, little more than about 50 per cent. of all the acceleration forces can be equalized by providing balance weights; the rest, say 10,000 pounds in our case, will act on the foundation, changing from positive maximum to zero and then to the negative maximum during one-half of a revolution.

In addition to this there is a second but minor force of reaction to which the foundation is subjected, due to the transmission of power by means of the belt. Assuming the diameter of the pulley to be 7 feet, the tension in the belt will be

$$\frac{150 \times 33{,}000}{3.1416 \times 7 \times 140} = \text{about } 1610 \text{ pounds}$$

which will have to be added to the acceleration forces when they act in one direction and subtracted when they act in the opposite direction. Adding these two forces, $10{,}000 + 1610 = 11{,}610$ pounds, which is the total stress that has to be resisted by the foundation. The fact that the explosions take place in the engine cylinder, thus driving the engine, has nothing to do with the phenomena here discussed, as they would occur regardless of how the power might be applied. It makes no difference in foundation stresses whether the power is generated in the cylinder by combustion or whether the engine is driven by a belt from some extraneous source, provided the engine be substantially built and of liberal dimensions, preventing vibrations due to excessive deflection of one or several of the engine parts.

W. A. ABEGG.

Los Angeles, Cal.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Accident to a Corliss Engine

One of the engineers in a plant which operated continuously found upon reporting for duty that the valve gear on the crank end of the Corliss engine was not working properly. Investigation showed that the cast-iron bonnet which supports the valve stem was broken at both top and bottom, as shown at *A* and *B*, Fig. 1. As the valve stem contained sufficient metal to make it extra strong, it did not fail under this severe test. The engine was shut down, the valve gear disconnected and the bonnet removed, the latter being utilized as a pattern from which to cast another, which was finished and installed as soon as possible.

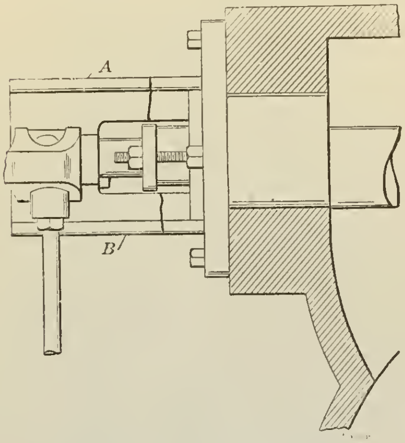


Fig. 1. Cracked Bonnet

A careful examination of the whole gear disclosed a crack in the head-end bonnet in almost the same relative location as the one in the crank end, but before it came apart a clamp was applied, as shown in Fig. 2. Holes were drilled in the clamp corresponding to those in the side of steam chest, the short cap screws which formerly held the bonnet in place were replaced by longer ones as shown at *C* and *D* and a bolt put through ears on each side. Although this prevented adjustment of the valve-rod stuffing box, it enabled the engineer to get started after a delay of nearly two days.

It seems that the bonnet must have been strong enough for all legitimate purposes, especially as the fractured metal showed a clean break with no sign of having been previously broken. It is possible that mismanagement caused the trouble. Suppose that when attempting to start the engine, the engineer did not admit steam enough to pass the outside center; in other words, with the crosshead as far away from the cylinder as possible, the crank stopped on the center. The engine being fitted with a powerful device for turning the flywheel by hand under such conditions, and being anxious to get the machinery in motion, the engineer may have operated this device while the throttle valve was open. In the meantime steam would have collected in the steam chest and cylinder, and the crank having barely passed center with full pressure acting on the piston, the latter would start with a strong jerk very different from its usual motion. With the steam valve held to its seat by boiler pressure and requiring considerable force to move it, a sudden shock as above described would have caused something to give way, which in this case proved to be the bonnet.

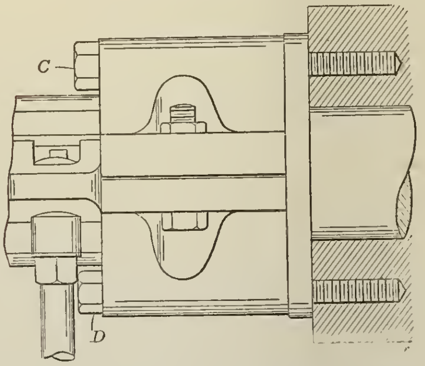


Fig. 2. Bonnet with Clamp Attached

After the valve gear had been repaired and all parts connected ready for use, steam was admitted, but the engine did not run smoothly and failed to furnish the required power. An indicator was applied and the diagrams shown in Fig. 3 were secured. Casual examination of these seems to show negative pressure, but careful measurements with a planimeter gave a mean effective pressure of about two pounds for the right hand, and eight pounds for the left hand, or an average of five pounds for a complete revolution. This would not be sufficient to run the engine and shafting, but the main pulley is fitted with a clutch; hence, none of the shafting was running because this clutch was thrown out, therefore this mean effective pressure of five pounds was enough to give the engine and main belt full speed.

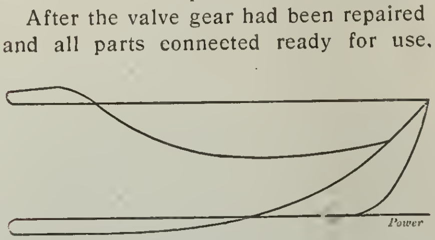


Fig. 3. Diagram Taken with Valve Reversed

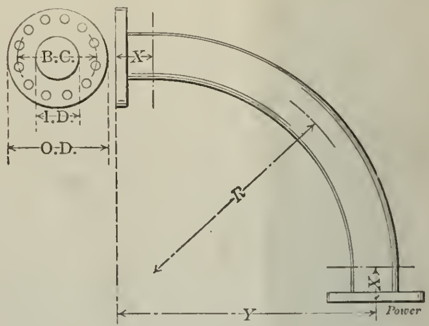
Each steam valve on this engine is driven by a T-head on the valve stem; hence, they are removed by taking off the bonnets on the side opposite to the valve gear, and drawing them out without disturbing the gear. When the crank-end valve was replaced after the repairs were completed, it was reversed by mistake and remained open during the whole stroke, as the diagram shows. After it had been properly inserted, another set of diagrams was taken and showed satisfactory results.

W. H. Wakeman.

New Haven, Conn.

Pipe Bends

The pipefitter of a number of years ago was accustomed to bend pipe because he was unable to obtain fittings to meet his needs. We at the present time occasionally bend pipe for this reason, but more often for other considerations which he never thought of, such as expansion, frictionless joints, etc.



Standard Bend

Bends are better able to withstand high pressures than any kind of fitting. They also take up expansion much better and give less trouble than expansion joints.

In making bends, certain rules should be observed. Referring to the sketch, the radius *R* should be at least five times the diameter of the pipe, and more if possible, as it decreases the friction and increases the flexibility: that is, the amount the bend will close or open without rupture. This not only refers to quarter bends but to all bends. The distance *X* is the amount of straight pipe which must necessarily be on the ends of all bends in order to thread them or make the joint. The table gives the smallest advisable amount that may be allowed for this distance but these values can be increased to any length.

The distance *Y* from the center to the face of the bend is, of course, the sum of *R* and *X* and a great many engineers and steamfitters give this measurement only, letting the manufacturer choose the radius and tangent.

The size of the pipe and description, whether standard or extra heavy, should always be given; and in regard to this I believe a great many make a mistake when they order bends made from extra-heavy pipe. The weak part of a bend is the joint or the manner in which it is fastened at the ends, and as we increase the thickness of the pipe we decrease the flexibility or spring of the bend.

For this reason I would advise the use of standard-pipe bends unless the conditions are unusual.

The following tests made by a prominent manufacturer will probably make this point clearer: An 8-inch quarter bend made from standard-weight pipe with a 40-inch radius and fitted with extra-heavy cast-iron flanges sprung 1⅛ inches when one of the flanges broke. An 8-inch quarter bend made from extra-heavy pipe, with the same radius and working under exactly the same conditions, sprung only ¾ inch when the flange broke. This showed that the standard bend was the stronger.

SIZES OF PIPE BENDS.

Size of Pipe, Inches.	Radius, "R."	Tangents, "X."	Length of Pipe in Bend of 90°.
2½	12½"	4"	2′- 3¾"
3	15	4	2′- 7¾"
3½	17½	5	3′- 1½"
4	20	5	3′- 5½"
4½	22½	6	3′-11½"
5	25	6	4′- 3¼"
6	30	7	5′- 1⅛"
7	35	8	5′-11"
8	40	9	6′- 9"
9	45	11	7′- 8¾"
10	50	12	8′- 6½"
12	60	14	10′- 0¼"

If flanges are desired on the bend, the *I D* (inside diameter) and *O D* (outside diameter) should be stated when ordering, as well as the *B C* (bolt circle) and number and size of the holes.

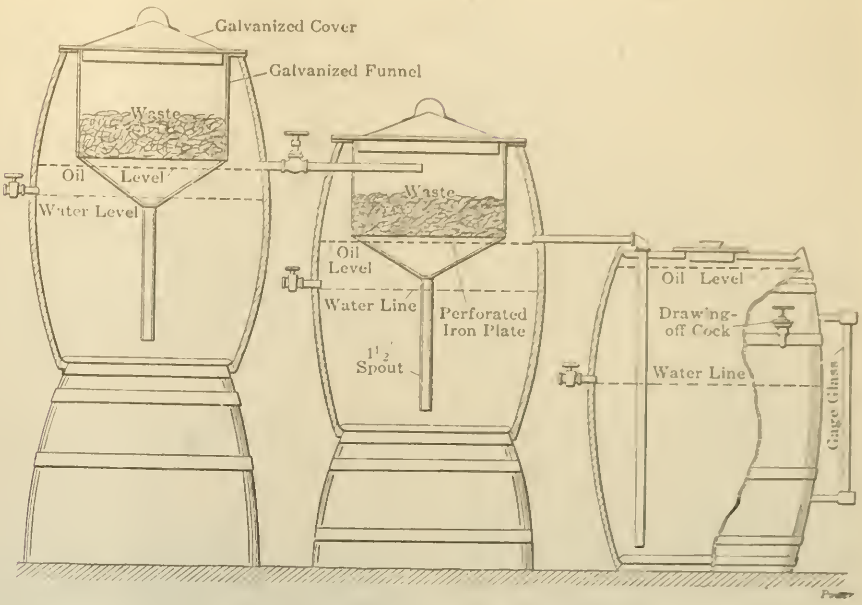
Bending a piece of pipe naturally weakens it, but with the radius given in the attached table the bend is from 15 to 20 per cent. stronger than the threaded joint. Welding or shrinking on steel flanges greatly increases the strength of the joint, but screwed joints are strong enough for any of the pressures we meet with in ordinary practice.

H. G. SCHULER.

Kansas City, Mo.

A Homemade Oil Filter

The homemade oil filter shown in the accompanying illustration has been found by an actual test of two years to be very efficient.



OIL FILTER MADE FROM BARRELS

It is made of empty barrels and other scrap material and performs its duty as well as any costly filter.

In almost any plant enough discarded material can be found to make a filter of this kind, and if the engineer is handy with the soldering iron the services of a tinsmith will not be required to make the funnel and covers.

C. F. HALL.

Yazoo City, Miss.

Running Corliss Valve Gear Noiselessly

Many men in charge of Corliss engines are apparently painstaking and particular regarding the care and operation of the equipment under their charge, excepting the valve gear. The writer has entered a great many engine rooms and found well kept machinery operating smoothly and quietly, with many visible evidences of care on the part of the engineer, yet with the valve gear chattering and rattling.

In many cases, of course, the noisy valve gear is due to lack of pride or to neglect on the part of the engineer. But there are more cases perhaps where the engineer has never realized that with a little patient and well directed effort this important part of the engine can be made to run almost without noise. Once having felt the satisfaction and pleasure of efforts thus rewarded, many who are now satisfied or indifferent would become enthusiastic on the subject.

In some instances by simply keying up the lost motion a great improvement would be effected. In this connection it is of first importance to make certain that the eccentric and reach rods are free from lost motion at the rocker arm, as a little looseness at this point reverberates loudly through the frame.

But there are other causes of a pounding valve gear not susceptible to this treatment, a poor fit of the wristplate on its bearing being a common cause. Frequently new engines develop this defect in their initial trial. A little lost motion at the center will be multiplied at the rim and may be seen by standing at the end of the valve gear and watching the vibrations of the wristplate at each stroke. Under these conditions the wristplate will ring or chatter badly. This can be cured only by bushing and fitting in a proper manner. The skeleton type of wristplate is a great improvement in this respect over the disk type and engineers having to do with the purchase or acceptance of an engine with the disk type should not accept one with the above described fault.

The next thing in importance as contributing to noisy operation is "side lash." or a lateral motion of the rods on the pins. Some builders make provision for taking up this lost motion by cutting the pins a trifle short and shimming out with circular paper liners between a flange washer and the end of the pin. Few valve gears will run noiselessly unless cared for in this respect.

As there is very little movement in the lower ends of the dashpot rods they may be keyed up quite snugly, and with these free from lost motion at both ends the steam hooks can be set quite closely. This is desirable, as when a hook is allowed excessive travel, a disagreeable noise is plainly audible as it engages the steam arm. Of course, care must be taken not to set the hook dangerously close.

Owing to their design, some hooks resist very stubbornly attempts to make them run noiselessly. It is essential that they fit well on their pins, both reciprocally and laterally. Grease applied by means of a compression grease cup in the pin bearing acts as a deadener of sound. The leather buffer inside the hook should be as thick as possible, allowing a safe hold for the hook. Frequently leather can be dovetailed into the rear side of the hook in such a manner as to strike against the pin holding the spring thus forming a srot of additional silencer, but this is not always necessary. If a click occurs at the time the hook strikes the knockoff block, the substitution of a fiber block will nearly always cure it. This style of knockoff block wears indefinitely and is considered by many builders as superior to steel.

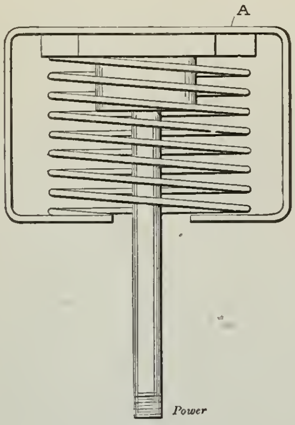
Compression in the cylinder aids in the attainment of a smooth-running valve gear for the reason that it tends to equalize the pressure on the steam valves, thus decreasing the friction due to boiler pressure in the steam chest. The writer does not remember having seen this point touched upon in the discussions about compression, although it will, perhaps, be admitted as of some importance.

F. C. Hally.

Yazoo City, Miss.

Device for Inserting a Valve Stem

Often when inserting pump valves or other valves of a similar type, more or less trouble and inconvenience are experienced in getting the valve stem to catch the thread. This is on account of having to resist the pressure of the spring while at the same time turning the stem, and this often has to be accomplished while reaching through a handhole. Therefore, if a number of valves are to be inserted, the work will be facilitated by temporarily compressing the spring in some way.



Spring in Position

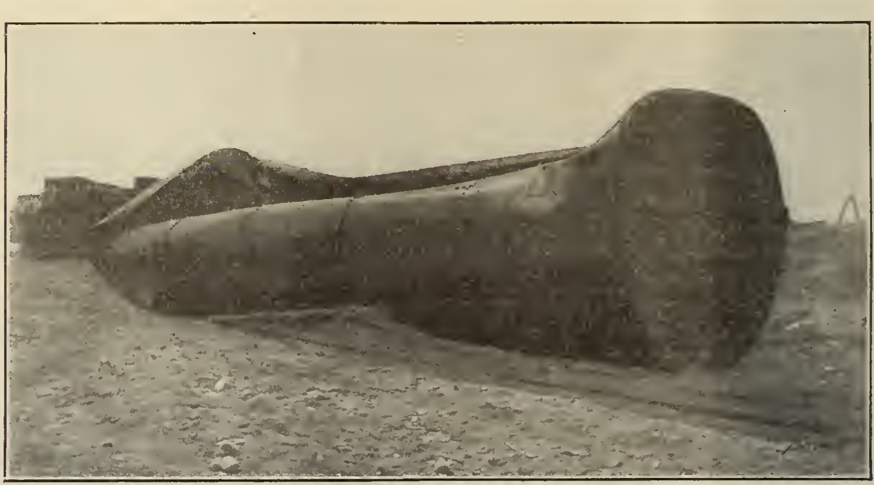
The accompanying sketch shows a very simple and handy means of doing this. The flat piece of wrought iron *A*, about 1/16x½ inch, is bent into the form of a rectangle with the ends not coming together on the fourth side. These ends are caught under the spring and the opposite side slipped over the head of the valve stem as shown. The stem can then be quickly and easily turned in and the piece of wrought iron as easily slipped off.

O. W. Middleton.

Chicago, Ill.

Collapse of a Cylinder Tank

The accompanying photograph shows a cylindrical tank which collapsed in the plant of Zinsser & Co., of Hastings-on-Hudson, N. Y. The accident was due to the attendant not following instructions and allowing cold water to enter the tank while it was full of hot vapor. This probably created a partial vacuum which the tank was not designed to withstand.



Collapsed Tank

If any readers of Power have any other theories to offer as to the cause of the failure, I should like to hear of their views.

H. Schneider.

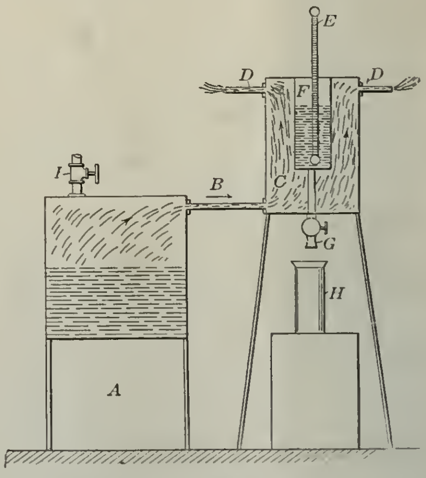
Hastings, N. Y.

Viscosity of Oils

The viscosity of oil is closely related but not proportional to its density. It is also closely related and in many cases inversely proportional to its lubricating qualities. These relations are not the same at ordinary temperatures as at high temperatures; therefore viscosity tests should be made at a temperature as near as possible to that at which the oil is to be used. A very common method is to take the viscosity as inversely proportional to its flow through a standard nozzle, while maintained at a constantly diminishing head and constant temperature. As the viscosity of water is nearly constant for all temperatures, it furnishes a good standard for comparison.

A simple apparatus, shown in the sketch, can be constructed easily and comparative results obtained. The tank *A* is partly filled with water through valve *I*. The water is heated by a blow torch or some other suitable means, and the steam passes through the tube *B*, into the chamber *C* and around the compartment *F*, containing the oil and the thermometer *E*. When the operating temperature of the oil has been reached, and kept fairly constant, valve *G* is opened and a flask *H* is filled to a certain hight, the exact time of flow being noted. This is repeated several times with each oil and the average obtained.

When the exact periods for filling the flask *H* to the given hight are obtained for the different oils to be tested, also for water, the comparative viscosity may be obtained by dividing the time of flow of the oil by the time of flow of the water. For example, it requires 102 seconds for a certain quantity of cylinder oil to flow through the nozzle *G* at a temperature of



Apparatus for Testing Viscosity

212 degrees, and it required 29 seconds for the same quantity of water.

$$\textit{Comparative viscosity} = \frac{102}{29} = 3.51$$

The flash and ignition point can be found in the following manner: Place about half a pint of oil in a porcelain evaporating dish and gradually heat. As the temperature rises apply a small flame from a pine stick, about ½ inch from the surface, until a flash is noticed to skim over the oil. At this stage note the temperature which will represent the flash point.

The oil can then be further heated until it will ignite of its own accord; this temperature can be taken as the ignition point.

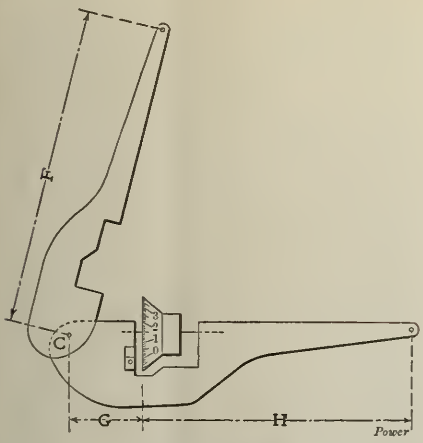
Do not place the thermometer directly in the oil but rather in a test tube containing mercury; this will lessen the danger of breaking it.

L. J. SCHRENK.

Detroit, Mich.

A Homemade Planimeter

There are many cases where a planimeter would save time and labor, but the amount of work is not sufficient to pay for its initial cost. The planimeter I constructed, a sketch of which is shown, had no cost at all, the material consisting of a few pins, a cigar-box cover, and half of a common thread spool.



PLANIMETER

The arms were made from the box cover; cutting out a space for the spool so that the axis of the spool, the tracing point, and the pivot *C* are in the same straight line. The spool is held in place by two pins which makes the friction of the wheel very small. The distance *G* is made as small as possible and *F* as long as is convenient. This gives greater roll to the spool and permits the divisions being made larger so that they are more easily read. The tracing point, also made from a pin with the head downward, may be set at any convenient length.

The wheel was graduated by tracing over known areas and the amount of roll for each square inch marked on the scale. These divisions were then divided into ten equal parts, thus making the instrument read to 1/10 square inch.

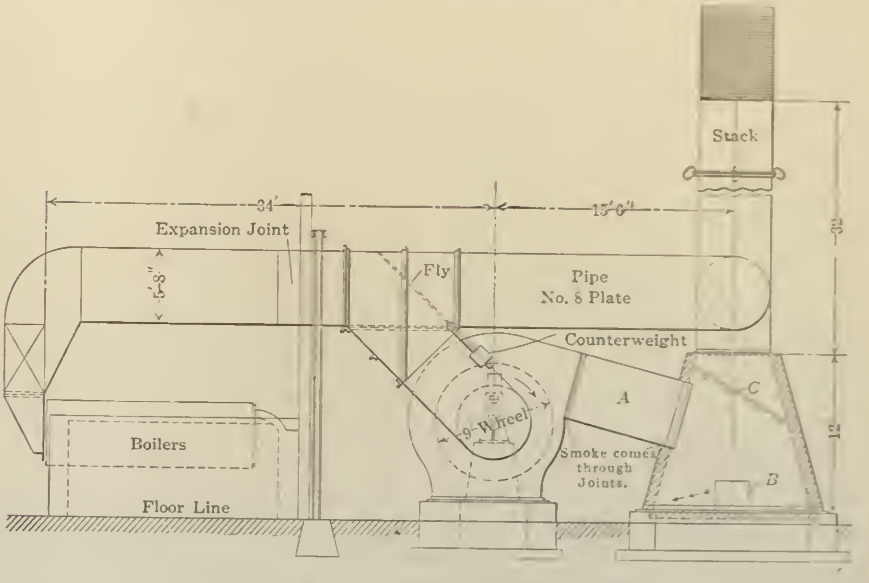
The cost of making such a planimeter is a minimum and if a little care is exercised in graduating the scale, fairly accurate results may be obtained.

J. E. ROCK.

Lansing, Mich.

Induced Draft Trouble

The writer recently took charge of a steam plant in which there were seventeen return-tubular boilers placed in two batteries of eight and nine respectively, the former consisting of five 60-inch and three 54-inch boilers 16 feet long. On account of damage to the smokestack of this battery it was decided to install an induced-draft system. This consists of a 7x7-inch vertical engine direct connected to a fan 9 feet in diameter with a 4-foot face. The engine varies in speed as follows: At 100 pounds pressure it makes 190 revolutions per minute; at 80 pounds pressure, 135 revolutions per minute, and at 60 pounds pressure, 105 revolutions per minute. The engine was supposed to run 300 revolutions per minute at 100 pounds pressure but 190 was the maximum speed obtained.



ARRANGEMENT OF INDUCED DRAFT OUTFIT

As the draft was too poor at this speed it was decided to install a more powerful engine, so that the fan could be speeded up to 300 revolutions per minute. A 9x14-inch engine of another make was procured and a 10x24-inch pulley keyed to the shaft; a similar pulley was also keyed to the fan shaft. The first engine was disconnected from the fan and the new one belted on. The governor on the latter engine was set at 300 revolutions per minute and the engine started. The results were disappointing as no better draft was obtained than in the former case. It was noticed that smoke poured from the joints in connection *A* (see sketch) between the fan and the stack, and if door *B* in the stack base was opened, smoke and ashes came out in the direction indicated by the arrows. The spark arrester *C* was removed as it was thought ashes might be held against it by pressure sufficient to choke it, but the results were precisely the same.

Here is a chance for the readers of POWER to determine the cause of the trouble and to state how they would remedy it with the least expenditure of time and money.

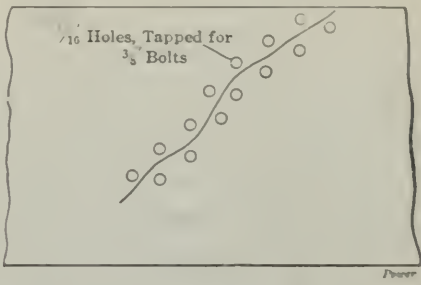
WILLIAM FAULTHIER.

Seattle, Wash.

Sewing Cracks

The method of sewing cracks in iron as illustrated in the accompanying sketch has been used by the writer for a number of years and always with successful results. It is particularly applicable in cases of cracks in flue sheets.

The method is to drill staggered holes on the opposite sides of the crack and tap them, using taper taps and not tapping all the way through. This will cause the bolts, when screwed up, to expand the inside of the crack and tightly close it.



METHOD OF MENDING CRACK

Wilson, N. C. R. W. GADD.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Increasing Capacity of Steam Boilers

The article by H. Kreisinger and W. T. Ray entitled "Increasing Capacity of Steam Boilers," which appeared in the May 24 issue of Power is useful in directing attention to possibilities of diminishing the size and cost of steam generators. It must not be thought, however, that this article and the tests referred to, which were carried out by the United States Geological Survey, have brought to light any facts not previously known about the laws connecting draft and fan power with heat transmission in connection with steam generators.

A great amount of research work has been done on this question of heat transmission in steam boilers although in most cases the tests have not been so thorough as has apparently been the case with those conducted by the United States Geological Survey. An inspection of the general index at the end of Professor Dalby's paper on Heat Transmission read before the Institute of Mechanical Engineers (London) in October of last year, will give an idea of the extent of the literature on the subject, and this index is not complete. It is a pity that many writers and experimenters on this subject did not, before writing or experimenting, make themselves more familiar with what had previously been done by others; although it would be a big task for any one person to read and digest all that has been written relevant to this subject.

As the reason why the heat transmission in steam boilers varies approximately as the square root of the draft may not appear evident, it may be useful to say a word on this point as, if a rule is understood, it is not only easier to remember but the rule is used with more confidence and satisfaction.

An elaborate treatise on the fundamental laws of heat transmission cannot be given on the present occasion but it may suffice to say that the rate of heat transmission between two fluids separated by a diaphragm (as, for example, between the hot gases and the water separated by the thickness of a tube or flue in a steam generator) can be expressed by an equation of the nature of

$$F = \frac{\theta}{r_1 + r_2 + r_3}$$

where,

F = B.t.u. transmitted per hour from one fluid to the other per square foot of surface. (Other units can, if desired, be used),

θ = Difference in temperature between the fluids,

r_1 = Resistance to the transfer of heat from the hotter fluid to the diaphragm,

r_2 = Resistance to the conduction of heat through the diaphragm,

r_3 = Resistance to the transfer of heat from the diaphragm to the colder fluid.

The equation, it will be seen, is of the same nature as Ohm's law for electricity and applies not only to steam generators but to surface condensers, tubular water heaters and the like.

In a steam generator under usual working conditions r_2 and r_3 are so small compared with r_1 as to be negligible or nearly so; the equation can therefore be expressed as

$$F \doteqdot \frac{\theta}{r_1} \text{ (} \doteqdot \text{ signifies approximate equality).}$$

The value of r_1 has been found by numerous experiments to depend greatly on the velocity, decreasing with increase in velocity; and in the case of the hot gases such as pass over the heating surface in a steam boiler, r_1 has been found to vary very approximately as the reciprocal of the velocity, so that the equation becomes

$$F \doteqdot \frac{\theta}{\frac{C}{V}} \doteqdot \frac{V\theta}{C}$$

where V is the velocity of the gases and C is a constant.

We have therefore arrived at the conclusion that the rate of heat transmission varies approximately as the velocity of the gases. But, as the kinetic energy of a fluid is proportional to the square of its velocity and as the kinetic energy must equal the energy expended in producing that kinetic energy, it follows that the velocity given to each pound of hot gas in a steam generator by the fan or other source of draft must vary as the square root of the energy usefully expended by the fan (or its equivalent). A certain irregularity is caused by the reduction in the gas velocity due to friction, but it will be seen that, generally speaking, the heat transmission would be proportional to the square root of the fan power, or the fan power would vary as the square of the heat transmission, if the weight of the gases passed through the boiler per minute were kept constant. If, however, the aggregate area for the flow of the gases is kept constant, the weight of gas passed per minute must increase with the velocity, so that the fan power varies as the cube, or third power, of the heat transmission. The product of draft and volume of gas per minute is a measure of power, and hence, as the volume per pound varies little unless with enormous drafts, the draft varies approximately as the square of the heat transmission or the heat transmission varies approximately as the square root of the draft.

Being clear as to the reasons for variation in heat transmission we can then try to benefit by applying the knowledge; but it is here that we are handicapped by want of data.

It is easy to draw up tables as given in the article above referred to; and such tables may certainly be sufficiently correct to be of service; but, having prepared the tables, we are still very far from knowing what is the best fan power to provide for in any case.

If a boiler plant is wanted to generate a certain weight of steam per hour at a certain pressure, then the question arises as to what is the best fan power to provide. To solve this problem we must consider the initial cost of the boilers, the initial cost of the fans with the engines or electric motors driving them, and the coal corresponding to the work absorbed by the fans. We must also consider the question of any variation in the boiler efficiency due to altered conditions as regards draft or arrangement of heating surface and any variation in engine efficiency due to alteration in the dryness fraction of the steam due to change in the evaporation rate. We must compare any increase or decrease in the standing charges which must be allowed for interest on initial expenditure and for depreciation, etc., with any decrease or increase in the running costs due to coal, wages, repairs, etc. We must, in fact, prepare a curve showing the variation in total annual expenditure with variation in fan power. This curve falls with increase in fan power up to a certain critical point and then rises; but the ascertaining of the critical point with reliability is a matter of considerable difficulty. The critical point is certainly not the same for all types of boiler and for all conditions. Weight is an important consideration in warships but is of comparatively little consequence on land. The importance of floor space varies under different conditions; and coal (or other fuel) varies greatly in price according to locality.

The most serious difficulty arises, however, in determining how the fan power affects the initial cost of the boiler per pound of steam generated per hour and the depreciation and repair costs. Increase in fan power reduces the necessary heating surface; but a little consideration will show that the reduction in cost of boiler is not proportional to the reduction in heating surface and must vary with the type and design of the boiler. Grate area, steam-liberating surface, and thickness of tubes and plates must all receive consideration. As regards the danger of overheating of plates, little appears to have been published on this subject which is of use in the present consideration, but there is, no doubt, that the risk of trouble due to overheating increases with increase in the rate of heat transmission. This point is of special importance when the feed water contains oil. Data on the subject by those who can speak from experience would be of great value.

R. M. Neilson.

Glasgow, England.

Blowoff Tank

In Power for May 17 particulars are given of an accident to a blowoff tank, the cover of which was fractured, presumably when the boiler was being blown down. There have been accidents of this kind before and in one which happened a few years ago, if the writer remembers correctly, there was loss of life.

It seems that an accident of this kind should be attributed not so much to the weakness of the tank as to the insufficient area of the vent pipe. It is evident that if the outlet pipes were only of the same total area as the blowoff from the boiler the pressure in the tank would very soon equal that in the boiler. It is necessary to make the outlet area sufficiently large to discharge all of the water and steam entering it while the pressure in the tank remains low.

These tanks are not usually safe under pressure of more than about 5 pounds per square inch and are often used in places where it is undesirable to blow down directly into the drain; the practice adopted being to blow down into the tank and allow the water to cool before running it off into the sewer. The outlet to the sewer enters near the bottom of the tank and is closed when the boiler is being blown down. With any arrangement, however, it seems safest to consider the outlet to the sewer as having to discharge water only and to calculate the size of vent pipe so that it will be capable of discharging all the steam which is formed with a pressure in the tank not to exceed, say, 5 pounds per square inch.

To perform this calculation we have to determine first the rate at which water will be discharged under a given pressure through the blowoff tap. The formula

$$v = \sqrt{2\,gh}$$

gives the velocity of a jet of water issuing under a head h but assumes that the water is flowing in a steady and unbroken jet. The conditions which obtain when blowing down a boiler, however, are very different from this, as the water is much hotter than 212 degrees and is therefore evaporating as the pressure falls so that the jet is actually a mixture of steam and water and the weight passing the blowoff tap per minute is materially less than is given by the above formula. The shape of the blowoff tap and the resistance of the pipes and bends also restrict the flow.

Actual tests made by noting the fall in the gage glass when the blowoff was fully open showed that with a pressure of 100 pounds in the boiler about 2000 pounds per minute was the rate of discharge through a 2-inch blowoff tap, which is only about one-fourth the amount which would be given by the above formula.

Taking this figure, 2000 pounds per minute, therefore, we next have to find how much steam is formed by the reduction of pressure from 100 to 5 pounds gage pressure.

Let x equal the weight of the steam formed per pound of water blown into the tank and assume that there is no loss of heat in the process. We have, then: The total heat of one pound of water at 100 pounds pressure equals 338 B.t.u. The total heat x pounds of steam at 5 pounds pressure equals $1183x$ B.t.u. The total heat of $(1 - x)$ pounds of water at 5 pounds pressure equals $228\ (1 - x)$.

Equating, we have,

$$1183x + (1 - x)\ 228 = 338,$$
$$955\ x = 110,$$
$$x = 0.115$$

pound.

The vent pipe therefore must be large enough to discharge

$$2000 \times 0.115 = 230$$

pounds of steam per minute at a pressure of 20 pounds absolute. Rankine's approximation for the rate of flow of steam gives results which have been checked and found to agree very closely with what is obtained in practice. The approximation is

$$q = p \div 70,$$

where q equals the weight of steam discharged per unit of area per second and p equals the internal absolute pressure. The external pressure is immaterial so long as it does not exceed $\frac{3}{5}\ p$. The rate of flow from the vent pipe works out by this formula to 17.16 pounds per square inch of area per minute. We have already found that the weight of steam to be discharged is 230 pounds per minute and consequently the size of the vent pipe necessary to do this under the conditions assumed must be

$$230 \div 17.16 = 13.4$$

square inches or, say, 4[illegible] inches in diameter.

In the example shown the total area of the vent pipe and outlet to the sewer is only 6.28 square inches so that the pressure in the tank must have been considerably more than 20 pounds gage pressure before it could pass all of the steam which was formed. To find the exact pressure in the tank, the calculation will have to be gone over again as the amount of steam formed at this pressure will be less than when the pressure is 20 pounds absolute. The calculation can be most readily made by a process of trial and error and it will be found that the pressure in the tank will have to be about 35 pounds absolute to give a rate of flow of steam through the outlet pipes equal to the amount formed by the reduction of pressure from 100 pounds to the pressure in the tank.

This pressure would probably not be sufficient to fracture the cover though it is higher than is quite safe for a cast-iron plate ⅞ inch thick and 2 feet, 6 inches in diameter without ribs. The calculation has been made moreover on the assumption that the outlet pipe is free to act as a vent as well as the vent pipe proper, whereas if the tank becomes full of water the outlet pipe will be covered and the calculation would then have to be made on the assumption that the vent has to discharge all of the steam. Under these conditions it is quite possible that the pressure in the tank would be high enough to burst it and it is quite likely that this is what occurred. The nature of the fracture clearly illustrates how the plate is weakened by the center manhole; the radial cracks evidently having started at the opening.

Wilbur Arnold.

Manchester, England.

The Selection of an Engine

Cornelius T. Myers in his discussion of the selection of an engine, in the May 24 issue, states that it is well recognized among engine builders of experience that high piston speeds mean better steam economies, for the larger the volume of steam worked through a given cylinder the smaller is the percentage of condensation therein.

Upon investigation it will be seen that this statement is incorrect.

In the first place, the given proof does not hold, for, if the engine is worked with a later cutoff than that giving the economical number of expansions, more steam is passed through the cylinder but the engine uses more steam per horsepower than it would if it were working with the economical number of expansions.

In the second place, take two engines, one of low speed, say 500 feet per minute, and the other of high speed, say 800 feet per minute, both developing 200 horsepower when working at 40 pounds mean effective pressure, and both having cylinders with the same percentage of clearance volume.

The cylinder dimensions of the low-speed engine would be,

$$\frac{33{,}000 \times 200}{40 \times 500} = 330$$

square inches in area, or about 20.5 inches in diameter. Taking the revolutions per minute as 100, the stroke would be 30 inches.

The cylinder dimensions of the high-speed engine would be

$$\frac{33\,000 \times 200}{40 \times 800} = 207$$

square inches in area, or about 16.25 inches in diameter, and, taking the revolutions per minute as 160, the stroke would be 30 inches.

Engine builders know from their own tests that, if these two engines were working at the same mean effective pressure, the larger engine would be the more economical. This may also be seen from published tests of engines.

The reason advanced for this is that with the increased cylinder dimensions the cylinder surface exposed to the steam is less, and in view of the fact that cylinder condensation is increased by increase of steam surface this seems to be the feasible explanation.

That the cylinder surface is the greater in the high-speed engine is readily seen, for the speed is increased 60 per cent.; but the surface per stroke decreases approximately only in proportion to the diameters, that is, 20 per cent.

Thus there is a choice of two engines both capable of developing the same economical and maximum powers and equally reliable; one, a high-speed, being of lesser weight, has its low cost to recommend it; the other, a low-speed, has its better economy to recommend it.

It is not only the buyer or his agent that has this choice, for the engine builder has to decide upon which engine to submit his tender, and he far too often only has the meager details of the inquiry or his knowledge of the destination of the engine to guide him.

In Mr. Miller's list of high-speed tandem-compound engines the maker *D* offered the biggest engine, at the best steam consumption and at the lowest price per horsepower. This, however, was passed over for maker *C*, who, besides not making such a good offer, was unwilling or unable to give steam consumption.

In the list of high-speed Corliss or four-valve engines the maker *D* offered the engine capable, as Mr. Myers states, of developing greater economical and maximum loads under all conditions at a lower price than maker *B*. This, however, was passed over for the reason Mr. Miller states.

Mr. Miller's final choice could also have been investigated if the various makers had been able to give the steam consumptions under the three conditions specified.

Both Mr. Miller and Mr. Myers are to be congratulated upon their contributions to this subject, for in many cases it is only considered when the results of a new plant have been disappointing, and it is then that the gas engine often makes a convert.

A. V. Clarke.

Gainsborough, England.

Central Station versus Isolated Plant

To my mind Mr. Naylor's statements in a recent issue of Power, in regard to Marshall Field & Co.'s plant being captured by the central-station people proves once again the almost inconceivable apathy displayed by the consumers of central-station current and, in particular, the operating engineers of isolated plants. According to a recent statement of the New York Edison Company to the Public Service Commission, their average rate for lighting was 7.13 cents per kilowatt-hour and for power 6.77 cents per kilowatt-hour.

The situation is simply this: The small consumers at 10 cents per kilowatt are forced to pay 3 or 4 cents per kilowatt more than the average, and the large consumers, who can afford to operate their own plants, and who, as a matter of fact, consume but a small proportion of the total output, enjoy the special privilege of paying 3 or 4 cents less than the average rate. In other words, the central-station people, who receive their valuable franchises by the consent of the people through their respective legislatures are allowed to use a rate discrimination of between 300 and 400 per cent. If this rate discrimination was adjusted on a fair basis between the central-station people and the public, as are railroad rates, gas rates, surface-car rates and those of numerous other public utilities, the "nigger in the woodpile" would be captured and annihilated instead of the private plant.

It seems to me that Mr. Naylor is certainly stretching his argument to the breaking point when he compares private water works and private gas plants with the private electric light and power plants. Mr. Naylor might see a ray of sunshine through his gloomy vision of the isolated plant's future in view of the fact that the extra cost of installing generating machinery in the average building is not a "million dollars or more," and especially as the engines for several months in the year incidentally act as reducing valves for the heating system. In lieu of more definite information, I presume Marshall Field & Co. buy their steam for heating purposes from an outside company, or else heat by electricity, as one reason advanced by Mr. Naylor for shutting down the plant is that they have no sidewalk room for delivery of coal and disposal of ashes.

I submit that my knowledge of Chicago plants is quite limited, but I am surprised to learn through Mr. Naylor that the element of "noise, dirt and danger" carries such weight in favor of central-station service. Why not make the isolated plant clean, quiet and safe, as a modern plant should be that is operated by a competent engineer who has the confidence of his employer, especially as discarding the electric generator units only relieves the engineer of one of the many responsibilities in connection with the modern plant. Every cloud has a silvery lining, however. The central station is a necessity and a benefit all around. But it must be compelled to play fair. In its energetic fight to displace the isolated plant, it has put the operating engineer on his guard, and consequently numerous plants are being operated with greater economy with a tendency to install a higher grade of machinery than in the past.

It is the old story of the survival of the fittest, and it is up to the operating engineers to realize and cope with the existing conditions. If they direct their concerted efforts through the proper channel, they are bound to come out victorious in the end.

Harry J. Marks.

New York City.

Boiler Room Ventilation

I would like to say a few words in regard to the letter by J. L. Shell in the June 14 number as he asks for the opinion of other engineers on the subject.

In the first place, keeping the boiler-room doors shut will, in a measure, keep the room temperature higher and decrease the loss due to radiation as the manager claims. But, as Mr. Shell says, the temperature would have to go to 373 degrees entirely to stop the radiation.

Now suppose the manager had Satan himself to fire in that temperature, as natural draft depends on the difference in the weight of a column of cold and a column of hot air; hot air inside the stack, and the cold air having free access to the ashpit, the devil, having considerably warmer air than normal in the fire room would get a considerably reduced draft. Also, as no fresh air came in, the resulting poor combustion due to an insufficient supply of oxygen would cause heavier loss of economy than would open boiler-room doors.

The radiation losses from the piping in the boiler room would not be 5 per cent. even with uncovered piping, but the economy of the boiler can easily be reduced 10 to 15 or 20 per cent. by poor conditions of fire and air supply.

By using pipe covering and also having a plentiful supply of good fresh air, more coal will be saved than with closed doors and bare pipe. Some places have practically tight fire rooms, but air is forced into them and out through the stack by pressure, and the air is fresh.

The manager shows very little humanity. Also he does not appreciate an engineer. Mr. Shell will not improve his prospects nor increase the manager's appreciation of engineers by staying there. I heartily approve of his resolution to let the cleaning of fittings go until a more convenient season.

W. C. Thorne.

Vineland, N. J.

Boiler Explosions

In Power for May 31 there appeared an article by Peter Van Brock under the heading "Boiler Explosion." It seems to me he leaves an impression which might prove disastrous if it were followed when he says that there is little or no danger of an explosion so long as the "coal tosser" keeps his water up to the mark. In the first place, he says that it is the engineer's business to see to it that the internal pressure does not exceed one-fifth of the strength of the boiler. We are perfectly willing to agree with him on this point. If we were always able to know that such conditions existed in our boilers we would never have any nightmare about boiler explosions; but, unfortunately, it is beyond the ken of man to know the exact condition of all parts of his boiler plant at all times. There are myriads of causes for the initial rupture, which is usually followed by a disastrous explosion. It may be due to contraction and expansion strains, poor workmanship, poor material, poor design, ignorance in operation and, no doubt, there are some cases where the rupture is caused by a weakening of the plates by overheating due to low water.

Brother Van Brock says that when the water level gets below the fire plate or tubes the steam gets enormously hot and dry and the pressure builds up very rapidly with an explosion as the result. On this point we ask, what is the purpose of superheaters? Is it to build up the pressure? We think not. If the pressure built up with the temperature we would still have saturated steam and not superheated steam.

Suppose that in a horizontal tubular boiler the water level falls 8 or 10 inches below the fire line. The conditions inside the boiler are not changed except that a more or less superheating of the steam has taken place without any change in pressure. The heat transfer from iron to steam is not nearly so rapid as from iron to water; consequently, the steam being unable to "soak up the heat," the plate between the fire line and the lowered water line becomes overheated, and, since steel when heated above 600 degrees Fahrenheit loses its strength very rapidly, the answer is easy. When the plate becomes too weak to stand the strain it parts, relieving the boiler of its pressure. The water contained in the boiler when relieved of its pressure suddenly becomes an explosive matter and forthwith proceeds to deface the landscape.

Years ago, probably at about the time when the ferry- and tug-boat boilers which Brother Van Brock tells us of exploded, it was customary in cases of boiler explosions to lay it all to low water, thus shielding the owners of boilers that should have been in the scrap heap and placing the blame on the engineer.

In recent years when men in charge of steam boilers became better acquainted with the physical properties of boilers and their contents, the ideas about explosions being due to low water began to fade away and more common-sense views of the matter were taken with the result that we are building better boilers, subjecting them to more rigid inspections, exercising more care in their operation, and we believe that, considering the number of boilers in use, there are fewer explosions and less loss of life and property.

Brother Van Brock says that at one time he carried 205 pounds pressure in a boiler that was allowed only 100 pounds by the inspector, and would willingly have carried another 100 pounds had it been necessary, safeguarding (?) himself by carrying 4½ inches of water over the fire tubes. In the light of the foregoing it is not hard to place the brother from the Hawkeye State.

In regard to low water, I would like to ask Brother Van Brock the following question: In the Manning, Climax, Porcupine and numerous other vertical boilers with about one-fourth or one-third of the heating surface above the water line, why is it that the steam does not become enormously hot and dry and build up the pressure to the bursting point? For example, take some of the small vertical fire-tube boilers in which the water line is not more than 4 or 5 feet from the fire, if the exposure of a plate or tube in one boiler has such a disastrous effect, why is not the same effect produced in another boiler? Again, how often do we hear of locomotive engineers being discharged for burning the crown sheets in their boilers? Is it to be supposed that the crown sheet is burned when it is covered with water? We think not. If not, why does not an explosion occur?

We should remember that low water leads to dangerous conditions and that excessive steam pressure must not be tolerated. We should refuse to operate a boiler when we know it to be unsafe. Our boilers should be thoroughly examined at regular intervals as to cleanliness, condition of fittings, setting, supports, braces, etc. A boiler explosion with high water is much more disastrous than with low water. We must not allow sudden opening of large valves. If these precautions are taken, we need not worry much about explosions, and in case an explosion does occur we will know that it was from some cause that could not be foreseen, and consequently is not the fault of the operating force.

R. L. Bevis.

Terre Haute, Ind.

Recent issues of Power and other technical papers have reported the wrecking of an engine by the bursting of the flywheel, resulting from a governor that failed to govern at the time most needed, and two disastrous boiler explosions from unknown causes.

Professors of expert knowledge may speak authoritatively on the cause by judging from certain of the effects visible to them. But vague speculations and theories founded thereon are not knowledge, and are of no practical value.

Shortness of water is often cited as a cause of explosions in steam boilers. It is a question if shortness of water ever was the prime cause of any boiler explosion while the water in the boiler was being depleted at a regular, steady rate.

I have seen a vertical boiler which a number of men assured me that they had seen red hot on several occasions; the engine stopping for lack of steam drew attention to the boiler. A hurried departure of all hands from several small factories in the vicinity followed each time. The boiler seemed to be no worse for its experiences. An old manufacturer in Detroit assured me that he had tried on several occasions to see if his boiler could be ruptured through shortness of water and was satisfied that it could not.

John W. Payler.

Detroit, Mich.

Lubricator Remedy

Referring to the remedy for lubricator trouble suggested by Mr. Wentz in the May 31 issue, I wish to say that I have used the following with good results:

After filling the lubricator, blow out the water in the glass thoroughly, cut off steam, take off the cap and pour in a little soapy water. Any kind of soap will do; I have used common laundry soap. I find that this remedy gives better results than the use of glycerin or salt.

Probably many readers have used this remedy. It is recommended to those who have not.

Martin McGerry.

Hilton, N. Y.

Oil Mill Superintendents Convention

The seventeenth annual convention of the Oil Mill Superintendents Association was held at the Menger hotel, San Antonio, Tex., on June 7, 8 and 9. Most of the members were present, and an enjoyable time was spent by all.

The convention was called to order by President T. J. McNulty, of Brookhaven, Miss. A number of interesting papers on engineering and other subjects pertaining to the operation of the mills were read during the three days' meetings, and much benefit was derived by the members from the practical points brought out in them and the discussions which followed. Some of the papers concerning the engineering side of the question were as follows: "Packing, Its Use and Abuse," by W. E. Sanders; "Indicating and Valve Setting," by J. C. Newberry; "Boilers and Their Care," by Walter Leonard; "Rope Driving," by W. B. Hosford.

Effective use was made of the blackboard in illustrating the subjects touched upon by the papers, especially in explanation of the indicator diagrams, and from the interest shown by the members in this subject it was plain to see that all thoroughly appreciated the value of the indicator and the necessity for its use in their plants.

The supplymen's organization, which is connected with the association, is only two years old, this being the second year that they have made an exhibit. This association is known as the Oil Mill Machinery Manufactures and Supply Association, and from the showing they have made this year it is a "husky" youngster, and sure to grow rapidly. At present there are 43 members, and a number of new applications on file. Those of the members who had exhibits of their products this year were: Philip Carey Company, Jenkins Brothers, Johns-Manville Company, Peerless Rubber Manufacturing Company, Cleveland Rubber Works, Crandall Packing Company, Buckeye Iron and Brass Works, Strong, Carlisle & Hammond.

The new officers elected for the coming year were: President, F. E. Voorhies, LaFayette, La.; vice-president, J. C. Newberry, Gonzales, Tex.; secretary-treasurer, B. C. Newberry, Lyons, Tex., reëlected; assistant secretary, Mrs. B. C. Newberry, Lyons, Tex., reëlected.

The State vice-presidents elected were: Texas, Ed Taylor, San Marcos; Arkansas, Guy Conley, Paris; Mississippi, W. R. Crout, Hazelhurst; Louisiana, F. A. Parodi, Bastrop, reëlected; Oklahoma, G. C. Reed, Stroud; Tennessee, Charles E. Graham, Memphis, reëlected.

The officers of the Manufactures and Supplymen's Association for the past year were: President, F. M. Smith, Dallas, Tex.; vice-president, A. C. Langston, New York; secretary-treasurer, W. H. Marshall, Chattanooga, Tenn. When the time came for electing the officers for another year, all of the present ones were reelected by acclamation. The next convention will be held at Houston, Tex., in June of next year.

Baltimore High Pressure Fire Service

The accompanying view shows one of the three new high-pressure fire-service pumping engines, built by Allis-Chalmers Company, to supply the high-pressure fire-protection lines of the city of Baltimore, Md. The three pumps are complete in the shops and are awaiting the completion of the new pumping station which is being built by the city.

These pumps will take their supply from the regular city lines, but additional provisions have been made for a supply direct from the harbor.

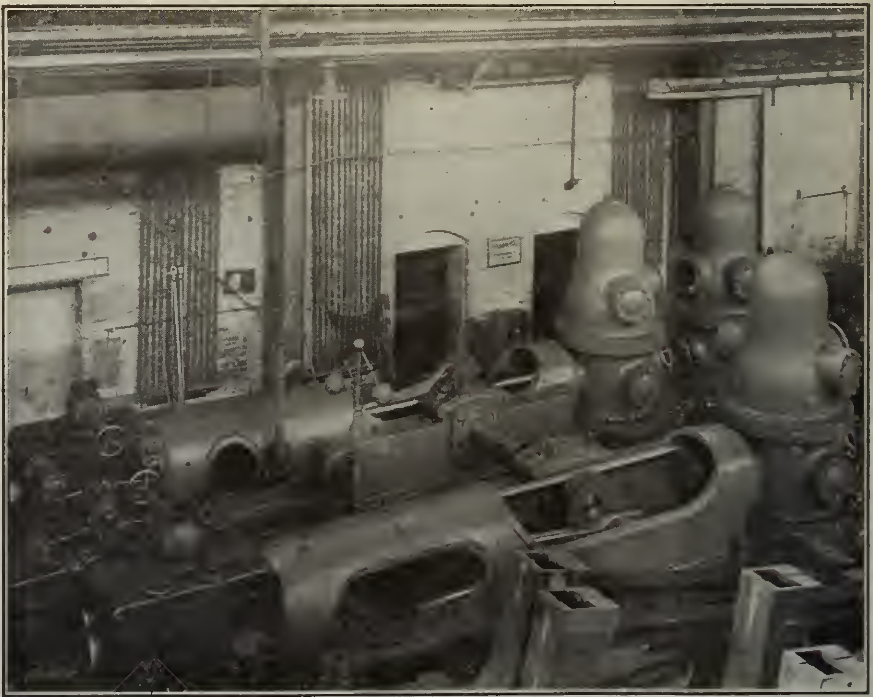
The pumping engines are of the horizontal, twin-duplex, crank-and-flywheel type with steam cylinders 22 and 22 by 36 inches in size and pump plungers 13¼ inches in diameter. Each pump has been designed to deliver 3000 gallons of water per minute against a total head of 150 pounds, but they are also capable of pumping against a maximum pressure of 300 pounds per square inch.

The type of machinery selected by the city of Baltimore is interesting, as it differs from that used by other large cities for fire protection.

The city of New York has pumping stations equipped with Allis-Chalmers motor-driven centrifugal pumps. Philadelphia, in which city the first high-pressure system was installed, decided upon geared triplex pumps, driven by three-cylinder, vertical gas engines. The city of Philadelphia has enough independent sources of gas supply to be convinced of its absolutely reliability. Both of these systems have been extremely satisfactory in their operation.

The next large city to install high-pressure fire-protection apparatus was San Francisco, Cal., immediately after the earthquake and fire which destroyed all of the business section of the city. Steam-turbine-driven centrifugal pumps were selected for both the land stations and the fire-boat equipments. Oil is used for fuel under the boilers.

When the new system is in service the insurance rates in Baltimore will be much lower than they are at present and a great conflagration will be practically impossible.



Allis-Chalmers Pumping Engine for High-pressure Fire Service in Baltimore

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y. Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* ,875,000 *copies of* POWER.

Our circulation for June, 1910, *was* 39,000.

July 5................ 34,500
July 12.............. 34,500

None sent free regularly, no returns from *ews companies, no back numbers.* *Figures* *re live, net circulation.*

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Power
and The Engineer

Editorial

The Institute of Operating Engineers and Existing Organizations

The following letter addressed to an officer of the National Association of Stationary Engineers who declined to take an active interest in the Institute for fear that it might detract from the older association, is interesting as developing more fully the character and purposes of the Institute and its real relation to existing organizations:

The National Association of Stationary Engineers is a fraternal organization, with ritual, signs, tokens and passwords. Its meetings, smokers, picnics, balls and conventions afford opportunities for members of the craft to extend and strengthen their acquaintance, to discuss the doings in the profession and to listen to a line of salutary talk from the leaders, to have a good time, including that significance of "good" which includes beneficial, and for the stronger personalities to achieve preferment and distinction.

Its lectures, discussions and prize contests are all good, in that they encourage men to think, to debate, to contest in intellectual achievement and induce habits of thought and study. But the educational work which it does is incidental. It begins nowhere and leads nowhere (except as all education, however desultory, leads upward). A man may be a member of the National Association of Stationary Engineers all his life without being able to do an example in decimal fractions.

As a fraternal society the National Association of Stationary Engineers is a helpful influence in the field and has done, and will continue to do, a lot of good along the lines projected by its founders.

The proposed Institute is a college with classrooms all over the country, in night schools, in homes, in college extension courses, in the institutions which have already been provided and in those the opening or establishment of which it will inspire, where men may learn as they earn. Its laboratories are the engine and boiler shops and the power plants of the country. Its curriculum will be such that when a man has mastered it he will have all that a technical graduate has of knowledge applicable to power-plant operation, and will not have to plead for primers and story books because he cannot read the standard literature of his business. He will have years of practical experience, acquired at the same time that he was acquiring his book lore, and developed and ripened by the application of that book lore as it was attained. And he will have such an education as will enable him to meet cultured people upon their own plane, and to take a gentleman's interest in the doings of the world.

When he passes the grade of a mechanician or hand workman, and has the fullness of years, experience and study necessary to constitute him one of the highly specialized and multifariously trained beings that the real operating engineer according to the Institute's standard will be, he becomes a full-fledged member of the Institute and receives a certificate, which can be made to mean more than a degree of mechanical engineering from any institute in the land to a man whose life work is to be the operation of power plants.

The time is not so far distant when anybody who had money enough to buy a pair or two of foceps and the strength to use them could be a dentist. If one wanted to get a tooth pulled, he usually went to the barber. Now, dentistry is a profession. Its members are well remunerated and can afford refinements, conforts and advantages that the mechanic cannot offer to his family. But nobody can be a dentist who has not graduated from a college of dental surgery.

The vocation of operative engineering is in much the same slack unorganized chaotic condition that dentistry was a hundred years ago. A man takes a job to run an engine for $1.50 a day because he cannot get a job to lay bricks at $4 a day, and he becomes "the engineer." And when the engineer is spoken of the popular conception does not jump to the thoroughly trained and highly specialized individual whom we know as the responsible engineer in charge of complicated power plant, but to a slouchy member of the laboring class who makes clean money by working in a grimy steam plant and who is not content to take the money home with him and leave the grime behind. It is true that a lot of the operating engineers are gentlemen in every sense of the word. They dress like gentlemen, behave like gentlemen, talk like gentlemen, practice the amenities of life and are agreeable social companions. But these are weighted down by a mass who have never got, and never will get, beyond the mere hand worker or craftsman stage, and when you say engineer the man on the street thinks of a paper cap, a monkey wrench and an oil can and not the indicator, gas analysis and power-plant economics.

When a man holds a certificate of membership in this Institute, it will mean that he is one of the first kind of men. We will make the term "operating engineer" mean something, and something more than a hand laborer, and while it is not expected to restrict the practising of operative engineering to those holding the degree of the Institute, as the practice of dentistry is confined to those holding the degrees of the colleges of dental surgery, we can make it mean so much, and so demonstrate to employers that this kind of a man is worth so much more than a mere laborer that operating engineers will be employed as heads of departments and at salaries commensurate with their responsibilities and preparation, and the Institute will be the natural source of supply for such men. Employers, commissions in charge of public work, and others are already coming to it for suggestions as to high-class men suitable for the large opportunities which they have to offer.

The Institute would naturally exert its activities in every legitimate way toward the amelioration of the condition of the workers and attendants around the plant —the men on watch in the boiler and engine room, who would naturally be its apprentices and journeymen—to a reduction of their time to a reasonable working day that they might have time and capacity left for self-improvement, to an improvement of the conditions surrounding their employment to include as much of comfort and convenience as is consistent with the character of the work, to induce a better class of young men to look to the vocation as a career and encourage them to stay in it after they get started.

The Institute will be to the vocation what the civil, mechanical, electrical and mining engineering societies are to those professions. It will have meetings, as do these great national societies, at which papers of real value will be presented and discussed, giving to their authors an opportunity to become known and distinguished along the lines of their particular research or practice, and to the profession the results of that study and experience. Their proceedings would become as important and valuable in this particular field as are the proceedings of the learned societies in the fields of their own activities.

In addition to prescribing for its undergraduate members, courses of study adapted to the several grades, the Institute would exert its activities in the direction of cultivating opportunities to pursue these lines of study. Advantage would be taken of the opportunities offered by such institutions as the Franklin Union at Boston, and various similar institutions in other cities, of the expressed willingness of a number of technical schools and colleges to place their facilities at the service of the working man when a practicable plan therefor is evolved, to procure courses in night schools, correspondence schools, Y. M. C. A. classes, etc., especially planned to carry out the requirements of the curricula and to meet the needs of the advancing students, and in general to do anything that it can do to systematize the work of getting education that will be helpful to the mechanic.

The interest and promised support and coöperation of prominent educators toward this end is an encouraging sign.

You would not expect a young man to give up his membership, or lose his interest in the National Association of Stationary Engineers because he took a course in power-plant engineering in the Franklin Union, would you? On the contrary, if every member of the National Association of Stationary Engineers in Boston would take such a course, we would have a higher and better standard of membership when they got through, and if with this technical instruction there could be combined that indefinable something which comes with education, and the scholarly association of man with man, we would have gone far in raising the plane of the membership, as we have been talking and writing of raising it and seeing it raised for the last quarter of a century.

You Will Never Know

In reporting the boiler explosion which recently occurred at the Midvale Steel Works, Philadelphia, Penn., in which two men were killed and several seriously injured, one of the daily newspapers assigned the following cause for the disaster:

"The cause of the explosion, according to officials of the plant, was carelessness on the part of the two men who were killed, * * * the workmen, while cleaning the boiler, had unknowingly loosened too many steam jets, and the vapor, pouring so strongly against the dust pans, caused the explosion."

This explanation impressed us so profoundly that we were constrained to appeal to the steel company for a few more details. Our request for information elicited the following reply signed by the secretary of the company:

"We beg to advise you that it is the policy of this company never to give out any information relating to our work. We regret that we cannot make any exception in this case."

It is needless to say that this broadgage attitude is thoroughly respectable and highly conservative. It indicates acute business acumen. No danger of suffering from adverse criticism when no one outside is qualified to form an opinion as to who is responsible. No opportunity for others to profit by the mistakes that have been made.

The Midvalve Steel Company is heartily to be congratulated upon the broad mindedness of its managers.

The National Association of Stationary Engineers

POWER yields to nobody in its interest in and good wishes for the National Association of Stationary Engineers. Its senior editor has been a member for a quarter of a century and three more of its editors are members in good standing. We have published and lauded its achievements, taken part in its committee and research work, and urged its advantages to those who were eligible for membership. A canvass of the members would reveal the fact that a large percentage of them got their first knowledge of the association from POWER and that the impression produced by what they read was favorable to the organization.

We have not hesitated, however, when the occasion seemed to demand it, to criticize the methods followed and to suggest how the means and activities of the association could be employed to better advantage. The expensive annual convention is a case in point. The bulk of the income of the national organization, upward of twenty thousand dollars, is disbursed every year in mileage that some two hundred delegates may travel across the country, elect a new set of officers, and decide where the next convention is to be held. Add to this mileage the hotel and other expenses of the delegates and a fair compensation for four days' time of two hundred high-grade men and we challenge any advocate of the continuance of the system to show anything that any recent convention has done which has been worth the expenditure to the membership.

Bill Banger, in his impressions of the last convention, brought this point out forcibly and we are glad to see that our views are shared by the president of the Wisconsin State Association, who in a forceful address, which is being presented at all the State conventions, demonstrates the small amount of benefit acruing to the membership from the big annual convention and urges smaller representation through State associations and the use of the fund now spent in transporting a large number of men to great distances for no useful end to educational and other beneficial purposes

---

John E. Carroll, of carbonic-acid-engine fame, was arrested in Boston, Mass., on June 23, charged with having used the United States mails in a scheme to defraud.

Inquiries of General Interest

Speed of Rope Drive

What are the velocities at which different sizes of rope should run to transmit one horsepower?

R. E. A.

For transmitting one horsepower, ropes should run at speeds as follows:

Diameter.	Speed.
½ inch	1000 feet per minute.
¾ inch	450 feet per minute.
1 inch	250 feet per minute.
1½ inch	115 feet per minute.
2 inch	75 feet per minute.

This table is based on the assumption that the working stress shall not exceed 200 pounds for a cross-sectional area of 1 square inch, which is about one-thirtieth of the strength of the rope and allows a wide margin for wear.

Volumes of Air Compressed

How many volumes of free air must be compressed to obtain any desired pressure?

H. D. W.

As many volumes as 14.7 is contained times in the number denoting the absolute pressure to which the air is compressed.

Hard and Soft Patches

What is the difference between a hard and a soft patch on a boiler, and should a patch be put on the inside or outside of the shell?

C. W. A.

Soft patches are bolted to the sheet and are usually made tight with red lead or similar packing as calking is impracticable. Such patches are seldom used with high pressures.

Hard patches are riveted to the sheet and made tight by calking. Sometimes hard patches are put on with tap bolts which are screwed into tapped holes in the sheet and riveted. A patch should always be put on the inside as otherwise it will form a sort of pocket for the collection of scale and sediment.

Temperature and Pressure

If two vessels of the same or different sizes, one half full and the other quarter full of water and heated to the same temperature, will the steam pressure be the same in both? And if so, why?

C. J. M.

If both are heated to the same degree the steam pressure in one will be equal to that in the other, for the pressure of steam always depends on the temperature and not on the amount of water. If the temperature is 212 degrees the pressure in the vessels will be 14.7 pounds per square inch. If the temperature is raised to 338 degrees the pressure will be 115 pounds.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Direction of Rotation

What is meant by the expression "running with the sun" as applied to waterwheels?

It is a term used by millwrights and means to the right or "right handed" as the hands of a watch or clock. Clockwise is a term frequently used to express the same thing.

Density of Water Under Pressure

Is it true that sea water is so dense at great depths that iron will not sink through it to the bottom?

M. M. H.

No! It is not true. Sea water at great depths is under tremendous pressure but being incompressible is no more dense than at the surface and iron will sink as readily at one depth as another as there is the same difference in weight for equal volumes.

Pressure on Safety Valves

I was denied a license the other day because I said that if there were two safety valves on the same boiler, one 2 inches in diameter and the other 3½ inches, the pressure would be the same on each. Please let me know what was wrong in my answer.

A. S.

The pressure *per square inch* is the same against each; but there are more square inches in one case than in the other, so that the total pressure against the 3½-inch valve is about three times as great as that against the 2-inch valve. There was evidently a misunderstanding about what was meant by the pressures being "the same."

Force of Expanding Water

If I confine one cubic foot of water at a temperature of 32 degrees and heat it to 212 degrees, how much pressure will it exert in trying to expand?

P. H. H.

It would exert a pressure equal to that required to compress it from the greater volume to the smaller and as this has never been accomplished it is impossible to tell what the pressure would be.

Diagonal Stays

How is the required cross-sectional area of a diagonal boiler stay determined?

U. S. R.

Multiply the area of a through or direct stay necessary to support the given surface by the length of the diagonal stay and divide the product by the perpendicular distance from the surface supported to the center of the palm of the diagonal stay.

Steam Pump and Injector

What makes a steam pump feed the boilers? Why is it that an injector will not feed a boiler of high pressure when it is hooked to a low-pressure boiler? I have one high-pressure and two low-pressure boilers and my injector won't work on the high pressure because it is hooked to the low-pressure boiler and I use a power pump on the high-pressure boiler.

E. G. W.

The ordinary boiler-feed pump forces water into the boiler from which it takes steam, because the steam piston has a larger area than the water piston, and the steam pressure on one side of the steam piston is not balanced by the water pressure upon the opposite side of the water piston, and it is this unbalanced pressure that drives the pump.

Injectors have a limited range outside of which they will not work. An injector using steam at 100 pounds pressure might force water into a boiler against the pressure of 150 pounds per square inch, while the same injector using steam at 40 pounds pressure might not be able to force water against a pressure of 60 pounds.

To feed a boiler carrying a high pressure with an injector taking steam from a boiler carrying a much lower pressure would require that the injector be especially designed for this work, as exhaust injectors are.

Horsepower of Engine

I have a 10-inch engine with a stroke of 32 inches running 170 revolutions per minute. The boiler pressure is 80 pounds. What is the horsepower of the engine?

J. C. A.

The piston speed is 906 feet per minute, and (neglecting the area of the piston rod) the engine will develop 2.15 indicated horsepower for each pound of mean effective pressure, which if taken at 35 pounds would be 75.25 horsepower.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Nelson Blowoff Valve

The body of this valve is provided with the inlet *A* and outlet *B* which is controlled by the reciprocating piston *C*. In open position the piston provides a curved port, as shown at *D*. The piston has a V-shaped recess directly over the curved port to retain the packing *E*, and is adjusted by the gland sleeve *F*, which is also provided with similar V-shaped recess. The gland sleeve is carried by the piston *C*, and adjusted by the octagonal nut *G* which is screwed onto the upper end of the piston and is rotated by the double-handled wrench *H*. The wrench cannot be removed from the valve without disassembling, but may be lifted so as to reëngage the octagonal nut *G* in its various positions. The gland sleeve *F* has a flattened portion to engage a similar flattened portion below the thread on the end of the piston *C*, and engages with the guide ribs on the yoke which prevent the rotation of the piston so that the curved port of the piston will register with the inlet connection when in an open position.

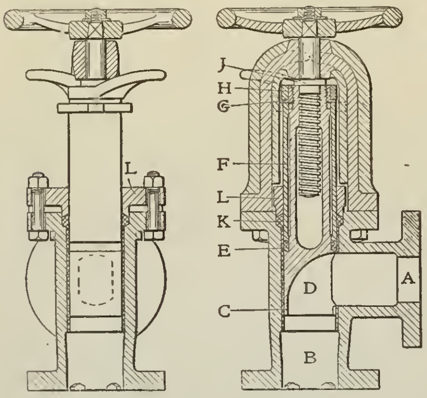


Fig. 1. Valve Open

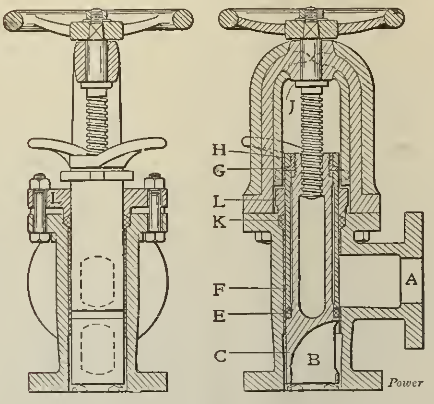


Fig. 2. Valve Closed

The travel of the piston is limited in its open position by the collar *J*, and in its closed position by the lugs at the outlet connection.

Around the gland sleeve, above the inlet connection, a recess is located in the valve body for the packing *K*, which is adjusted by the gland *L*.

When opening or closing the valve, piston, packing, gland sleeve, octagonal nut and wrench travel in unison. The packing *E* in the V-shaped recess passes the inlet connection in opening or closing, but the packing *K* is stationary. The packing *E* is adjusted by the wrench *H* when the valve is in a closed position.

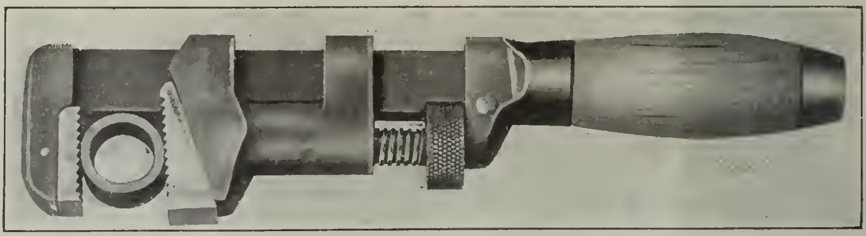
The valve is made in all sizes, flanged and screwed, straightway and angle, and is designed to withstand a working pressure of 250 pounds with full pipe area and bronze fitted to withstand the destructive effect of the blowoff products.

A good feature is that the various parts and adjustments are in full view of the operator at all times.

This valve is manufactured by the Nelson Valve Company, Chestnut Hill, Philadelphia, Penn.

The Brosnihan Pipe Wrench

This pipe wrench can be adjusted and operated by one hand, leaving the other free to hold and guide the pipe. A firm and sure grip is obtained by means of wedge and screw attachments.



Brosnihan Pipe Wrench

The movable wedge or sleeve jaw being held against the pipe by a spring, grasps it instantly on the downward movement of the handle bar. It can be set so that it will not crush thin pipe and it can readily be released and removed from the work.

The jaws are made of tool steel, hardened, and tempered in oil, and the bar sleeve and screw are case-hardened. Extra jaws can be procured to replace those worn out, making an old wrench as good as new.

To use the wrench place it on the pipe and turn the thumb screw until the jaws press against the work. Then ease on the handle and turn the screw until the sleeve jaw projects 1/16 to ⅛ inch over the end of the sleeve, and then use as any ordinary wrench. For thin pipe set the wrench so the sleeve jaw, when holding the pipe, will rest on the bar.

This wrench is manufactured by the Brosnihan Wrench Company, 31 Hermon street, Worcester, Mass.

Liberty Four Valve Strainer

This strainer is made in two types, one with top drive, Fig. 1; the other with side drive, Fig. 2. The side drive is used on vacuum work where air leakage is not permissible when cleaning the strainer. The screw of the side drive passes up through the lower valve and is surrounded by a tube so that there can be no leakage past the threads of the screw where it passes through the lower valve. For pressure work and service in which a small leakage of air can be neglected, the top drive is generally employed.

The four-valve strainer, as the name implies, has four valves, two valves per set, one of which is hollow and in which is placed the strainer basket. When in use, the water is always passing through one of the strainer baskets, entering at the top, the other one being, if necessary, cut out for cleaning purposes. The body of the strainer is made of one casting.

In the condition shown in the illustrations the water is passing through the right-hand strainer, the left-hand strainer being raised and ready to be removed for cleaning, which can readily be done by removing the cover and slipping the strainer over the valve stem. When the dirt has been removed, the basket and the cover are replaced, and the basket is ready for service. The four-valve strainer is designed for use where the water does not contain very large quantities of foreign matter and is recommended only for such cases.

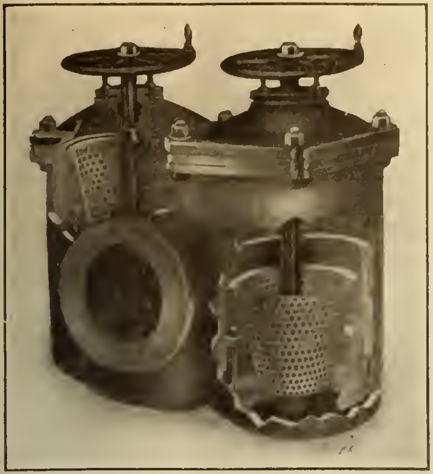


FIG. 1. TOP DRIVE

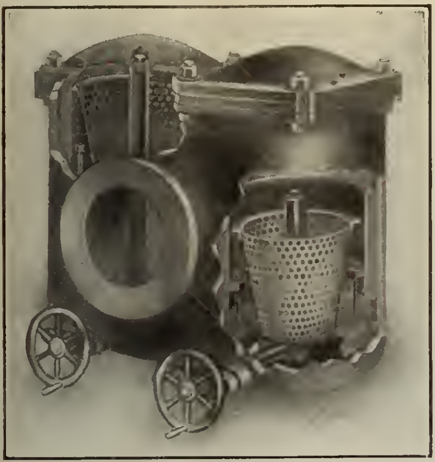


FIG. 2. SIDE DRIVE

The strainer is manufactured by the Liberty Manufacturing Company, 6705 Susquehanna street, Pittsburg, Penn.

Foster Fixed-focus Pyrometer

The chief argument against the usual forms of radiation pyrometers is that they require expert knowledge in handling them. The design of the Foster fixed-focus pyrometer overcomes this restriction. It requires no intricate manipulation or adjustment in its use.

The manner in which the instrument is used is shown in Fig. 1. The operator is observing the temperature of a boiler furnace and exploring the temperature distribution in order to determine if it is high enough and sufficiently uniform. The instrument indicates instantly on the dial the temperature of the hot body at which the tube is pointed. It is not necessary to open the furnace doors to take the temperature if a peep hole 1½ inches in diameter is available.

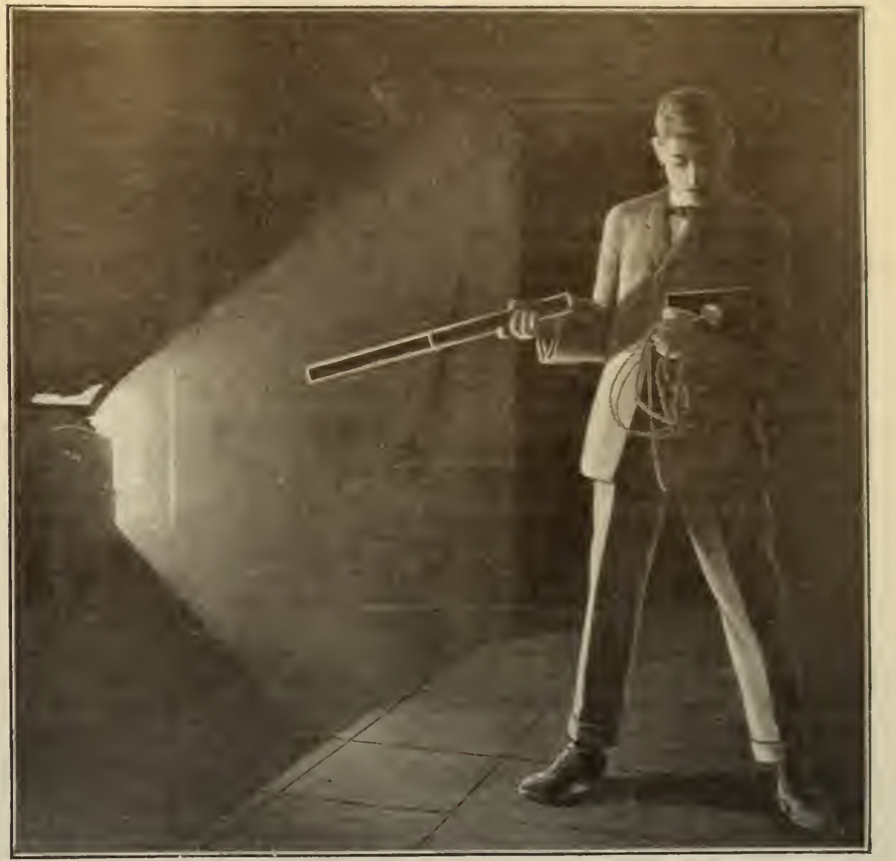


FIG. 1. THE FOSTER FIXED-FOCUS PYROMETER IN USE

The construction of the instrument is shown in Fig. 2, which is a diagram of the receiving tube. At the front end of the tube there is a diaphragm *E F* and at the opposite end a concave mirror *C*. The line *A B* represents the surface of any hot body and the dotted lines indicate how this will radiate heat to the mirror *C*. This mirror has one focus at the aperture *E F* and the other at the sensitive device *D*. This sensitive device is the junction of a small thermocouple. The mirror *C* concentrates upon *D* the heat image of the aperture *E F* filled with the radiant heat from the hot bodies and this heat image raises the temperature of the themocouple, giving rise to an electromotive force. Connection is made to an indicating millivoltmeter by means of a flexible cable and the indications of this millivoltmeter are calibrated to read in degrees of temperature direct.

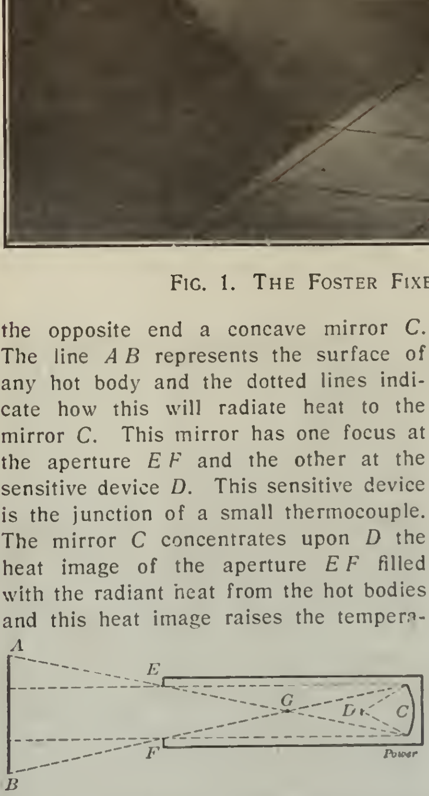


FIG. 2. DIAGRAM OF THE RECEIVING TUBE

The relative position of the mirror *C*, the sensitive device *D* and the aperture *E F* are all fixed so that the focusing is done once for all and is never changed in use.

In order that the aperture *E F* may be entirely filled with radiant heat from the hot body *A B* it is necessary that the tube be brought within a certain maximum working distance. This working distance varies with the size of the hot body and is measured from the apex *G* of the cone *G A B*. The position of this apex is shown on the outside of the receiving tube by a clamping ring which may be seen in Fig. 1 at the center of the tube.

The rule for the working distance is that the center ring must be within ten times the diameter or smallest dimension of the hot body. Any distance less than this maximum working distance is satisfactory and will not affect the reading. Looking at the diagram in Fig. 2, it will be seen that if the hot body *A B* were brought nearer to *G* the only result would be that the outer edges of it would not be in the measurement as they could only radiate heat to the walls of the tube and these are made nonreflexing.

The instrument is calibrated to give direct temperature reading without corrections.

It was devised by Charles E. Foster and is manufactured by the Taylor Instrument Companies, Rochester, N. Y.

Wisconsin N. A. S. E. State Convention

The tenth annual convention of the Wisconsin State association opened Friday, June 17, with an address of welcome to the delegates by Mayor John C. Voss, and a response by A. B. Balzer, State president, of Sheboygan. About sixty delegates from all parts of the State were present at this session, while additional members of the State body arrived in the city during the afternoon. In his address of welcome Mayor Voss extended a cordial greeting to the visitors and commended them on the purposes for which they met annually, concluding his remarks with an outline of the many beautiful buildings and recreation spots in the city and the advantages to be enjoyed by the residents of "Only One Oshkosh." Mr. Balzer, in his response, outlined briefly the history of the association, telling of its organization in the East 28 years ago and the progress it has made since.

Isaac H. Crawford, who acted as chairman of the opening exercises, then introduced Fred W. Raven, of Chicago, secretary of the National association. Mr. Raven said: "The objects of this association are truly embodied in fraternalism. As the preamble states, the association was organized for this purpose only, and not for the furtherance of strikes or to interfere in any way between its members and their employers in regard to wages. Its meetings are for educational purposes and for the securing of the enactment of engineers' license laws." Short addresses followed by F. H. Turner, of the *Practical Engineer*, and F. B. Fulman, of Muncie, Indiana.

W. A. Converse, of the Dearborn Drug and Chemical Company, gave an interesting talk on "Boiler Feed Water," before the convention Friday evening The talk was made more interesting by the use of charts, illustrations and chemical apparatus. Mr. Converse gave many demonstrations of much value to the engineers, which if applied to their routine work, would mean a saving of money and incidentally help to work out the idea of conservation of natural resources.

Officers elected for the following year were: A. A. Schroeder, of LaCrosse, president; A. La Budde, of Ashland, vice-president; S. W. Marty, of Oshkosh, secretary; John F. Murphy, of Madison, treasurer; Richard Gaepel, of Milwaukee, conductor; E. S. Wells, of Stevens Point, doorkeeper, and W. Classmann, of Milwaukee, State deputy. The new officers were installed by F. W. Raven, who congratulated them upon the advancement they had made in the association, and complimented the retiring officers on the work they had accomplished during the past year.

A special delegation from Milwaukee attended the picnic at the Power Boat club, which was one of the principal features of the entertainment program. Fishing and dancing were indulged in and a ball game between the Milwaukee and Oshkosh members proved an interesting event, in which the Oshkosh engineers succeeded in winning by a score of 4 to 3. A dinner was served at noon and a bountiful supper was prepared in the evening.

Saturday afternoon the visitors were entertained by George M. Paine, president of the Paine Lumber Company, on his steam yacht "Nia." A trip was made to Winneconne and Lake Butte Des Morts, returning to the Oshkosh Yacht club. Refreshments were served by the host during the course of the ride. On returning to their respective homes all reported one of the most pleasant visits ever enjoyed at a convention city and many valuable ideas were taken away.

The place of next meeting will be decided upon later by the officers. It is understood that La Crosse and Fond du Lac both are contesting strongly for the honor.

Conn. State N. A. S. E. Convention

The fifteenth annual convention of the Connecticut State association of the National Association of Stationary Engineers was held at New Haven, Friday and Saturday, June 24 and 25, with headquarters at the Garde hotel.

At eleven o'clock on Friday morning, a preliminary meeting was called to order by State President George A. Cleveland

Delegates and Visitors at the Connecticut National Association of Stationary Engineers State Convention in the engineers' hall in the Boardman building. Mr. Cleveland welcomed the delegates and guests and appointed the several committees, after which the meeting adjourned until nine o'clock on Saturday morning.

On Friday evening the convention assembled at the engineers' hall to participate in a social session. The festivities began with a brief address of welcome by Mayor Frank Rice. National President William J. Reynolds spoke earnestly in favor of State associations, and also gave a general outline of the good work of the National Association of Stationary Engineers. Past National President Herbert E. Stone praised the ladies' auxiliary and the life and accident department, and dwelt particularly on the fraternal side of the organization. Edward Maroney's short talk was well received. Joshua O'Leary recited and Jack Armour entertained with songs and stories. Refreshments were served. State President Cleveland was the toastmaster and H. G. Nichols was the pianist.

At the business session on Saturday morning, the principal work was in regard to the securing of a suitable State license law; the delegates voted unanimously for such a law. After several routine matters were disposed of the election of officers took place, resulting as follows: President, Charles H. Ostrander, Meriden No. 1; vice-president, Frank L. Chapman, Norwich No. 6; secretary and treasurer, Henry Vanderbergh, Hartford No. 5; conductor, John H. Sherman, New Haven No. 10; doorkeeper, Thomas Reed, New Haven No. 2; trustees, George P. Thomas, Norwich No. 6; Patrick J. Grace, Bridgeport No. 4; George A. Cleveland, New Haven No. 2. The officers were installed by Past National President Herbert E. Stone, assisted by Thomas P. Burke.

The State association voted to recommend to the national convention as the next State deputy James A. Laudon, Torrington No. 17.

Two new associations were admitted to membership during the past year.

The convention voted to recommend to the national convention in Rochester that Springfield, Mass., be the meeting place for the 1912 national convention.

The next annual meeting will be held in Hartford in June, 1911.

On Friday afternoon the convention took special trolley cars to Tabard Inn, Lighthouse point, where a shore dinner was served. The principal feature of this outing was a baseball match between the engineers and supplymen.

The exhibitors were: Dearborn Drug and Chemical Works; M. T. Davidson Company; Harrison Safety Boiler Works; Quaker City Rubber Company; Lake Erie Boiler Compound Company; General Electric Company; Keystone Lubricating Company; Perfection Grate Company; Paul L. Crowe; The Iron Works Company; Garlock Packing Company; Albany Lubricating Company; C. E. Squires Company; William R. Winn; Hartford Mill Supply Company; *Practical Engineer;* Connecticut Metal Boiler Cleaner Company; Ashton Valve Company; A. W. Harris Oil Company; H. W. Johns-Manville Company; McLeod & Henry Company; Home Rubber Company; Greene, Tweed & Co.; Jenkins Brothers; American Steam Gauge and Valve Manufacturing Company; J. H. Williams & Co.; Birch, Riley & Co.; C. S. Mersick & Co.; Lunkenheimer Company and POWER AND THE ENGINEER.

Excursion of the Combined Associations of Brooklyn

The third annual excursion of the Combined Association of Engineers of the Borough of Brooklyn, N. Y., was held at Cranberry lake, N. J., on Sunday, June 26. Fully fifteen hundred excursionists, comprising the engineers and their families, boarded special trains for the lake, where a most delightful day was spent.

Flywheel Explosion

Shortly after 7 o'clock on Thursday morning, June 23, a flywheel explosion occurred in the shops of the Morgantown & Kingwood Railroad Company at Morgantown, W. Va. Harry Murphy, foreman of the car-repair and wood-working department, was killed.

The engine to which the accident happended was an old simple affair which did not have any governor. It drove a circular saw for cutting up heavy timber.

The regular sawyer was not at the plant on the day of the explosion. Murphy started up the engine and, not being familiar with its control, it is thought that he turned on a full head of steam which caused the unloaded engine to race.

The flywheel was completely demolished. It was 6 feet 8 inches in diameter and 13 inches across the face.

Except for a few holes in the roof and side walls, where the fragments of the wheel penetrated the building, the property damage was small.

PERSONAL

C. T. Anderson has been appointed manager of the Chicago office, 1616 Fisher building, of the C. W. Hunt Company, New York, builders of coal-handling, conveying and hoisting machinery.

W. E. Crist, formerly president and general manager of the Crist Valve Manufacturing Company, is now the general manager of the valve department of the O. M. Edwards Company, of Syracuse, N. Y.

Charles L. H. Wilke, formerly sales manager of the Jeanesville Iron Works Company, has been appointed district manager of the Wheeler Condenser and Engineering Company, with offices at 816 Commonwealth Trust building, Philadelphia, Penn.

C. W. Hunt Company, New York, builders of coal-handling, conveying and hoisting machinery, has opened offices in the State Bank building, Richmond, Va., and also at 607 Rhodes building, Atlanta, Ga., with W. F. Lee, for several years preliminary engineer to the company, in supervision.

Walter H. Derriman, B.Sc., A.C.G.I., A.I.E.E., a well known electrical patent attorney, of London, England, has entered into partnership with the firm of Dicker & Pollak, chartered patent agents, of 37 Furnival street, Holborn, London, E. C., England, and will personally conduct their electrical patent work. The name of the firm has been altered to Dicker, Pollak & Derriman.

The firm of Frazier & Fox, engineers, Cleveland, O., has been dissolved. The business and obligations of the firm have been assumed by J. W. Frazier, who will continue the engineering work under the name of "The J. W. Frazier Company" with offices in the Rockefeller building.

J. H. Fox, the other member of the firm, will be chief engineer for the Pittsburg Plate Glass Company.

NEW PUBLICATIONS

TESTS OF TIMBER BEAMS, by Arthur N. Talbot, issued as Bulletin No. 41 of the Engineering Experiment Station of the University of Illinois, is a detailed report of the tests of a large number of full-size timber bridge stringers. The test beams included new, seasoned, and creosoted wood of several species. The report also gives the results of shear and bending tests on specimens cut from the large beams. Important conclusions concerning strength and other properties of structural timber are given. Copies of Bulletin No. 41 may be obtained gratis on application to W. F. M. Goss, director of the Engineering Experiment Station, University of Illinois, Urbana, Illinois.

A resolution has been introduced in Congress creating a commission to investigate and report upon the advisability of holding an exposition commemorative of the semi-centennial of negro freedom.

An exposition is to be held on Put-in-Bay island during 1913 in commemoration of the one hundredth anniversary of the battle of Lake Erie.

NEW INVENTIONS

Printed copies of patents are frnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

COMPOUND ENGINE. Charles Baden, Hackensack, N. J. 961,094.

TWO-CYCLE GAS ENGINE. Oscar H. Dorer, Irvington, N. J. 961,111.

INTERNAL COMBUSTION ENGINE. William W. Morse, Newark, N. J. 961,152.

STEAM ENGINE. Henry W. Weaver, Hutchinson, Kan. 961,186.

REVERSIBLE STEAM TURBINE. Frank C. Beyerle, San Diego, Cal., assignor of one-half to R. M. Mobius, San Diego, Cal. 961,205.

TWO-STROKE CYCLE MOTOR. Armand Peugeot, Tony Huber, and Henri de Lostalot, Billancourt, France. 961,315.

FLOAT TIDE MILL. Uriah S. Jackson, Ossipee, N. H. 961,410.

EXPLOSIVE ENGINE. Ernest S. Bowen, Geneva, N. Y., assignor of one-half to Walter Lester Fay, Geneva, N. Y. 961,581.

WINDMILL. John A. Swanson, Canton, S. D. 961,658.

INTERNAL COMBUSTION ENGINE. John W. Bradley, Seattle, Wash., assignor of one-half to Jacob E. Zickrick, Seattle, Wash. 961,677.

WINDMILL. Lee C. Matthews, San Francisco, Cal. 961,705.

BOILERS AND FURNACES

OIL BURNER: John C. Banks, Chillicothe, Tex. 961,386.

STEAM GENERATOR. Robert J. Galbraith, Albany, Oreg. 961,490.

STEAM GENERATOR. George M. L. Moore-Irvine, Newry, Ireland. 961,516.

FURNACE GRATE. Herman A. Poppenhusen, Evanston, Ill. 961,528.

POWER PLANT AUXILIARIES AND APPLIANCES

WATER PURIFIER AND CLARIFIER. John E. Angell, St. Louis, Mo. 961,091.

COOLING TOWER. Edwin Burhorn, Hoboken, N. J. 961,100.

WATER METER. Ernest E. Gamon, Newark, N. J. 961,116.

MECHANICAL JOINT FOR VACUUM TUBES. Daniel McFarlan Moore, Newark, N. J., assignor to Moore Electrical Company, New York, N. Y., a Corporation of New York. 961,151.

GOVERNING MECHANISM FOR EXPLOSIVE ENGINES. Peter Paulson, Marshfield, Wis. 961,156.

SWIVEL HOSE COUPLING. James Shotwell, Watsonville, Cal. 961,170.

VALVE DEVICE. John Slorach, Cleveland, Johannesburg, Transvaal. 961,173.

VALVE MECHANISM. Charles W. Everson, Brooklyn, N. Y., and George F. D. Trask, Orange, N. J., assignors to said George F. D. Trask. 961,220.

NIPPLE AND CLAMP FOR PIPES. Charles E. Maxfield, Cleveland, Ohio, assignor of two-fifths to Egbert F. Hammond, Cleveland, Ohio. 961,247.

CARBURETER. Ernest N. Broderick, Detroit, Mich. 961,284.

STEAM REGULATING VALVE. William F Kiesel, Jr., Altoona, Penn. 961,300.

REVERSING GEAR. Victor E. Randall, Battle Creek, Mich. 961,316.

TURBINE BLADE. Henry H. Watt, Chicago, Ill. 961,328.

PISTON PACKING Charles E. Ross, Columbus, Ohio. 961,373.

CARBURETER. Thomas Leggett Sturtevant, Quincy, and Thomas Joseph Sturtevant, Wellesley, Mass., assignors to Sturtevant Mill Company, a Corporation of Maine. 961,423.

VALVE. Theodore Shade, Sacramento, Cal., assignor to Lightning Valve and Dredge Company, Stockton, Cal. 961,466.

CARBURETER. William C. Carter, St. Louis, Mo. 961,481.

SHAFT PACKING. Chester Comstock, Ridgewood, N. J. 961,484.

SHAFT BEARING. George C. Hicks, Jr., Connersville, Ind., assignor to the P. H. & F. M Roots Company, Connersville, Ind. 961,496.

OILING DEVICE. Robert T. Horlick, Chicago, Ill., assignor of one-half to Edward T. Horlick, Chicago, Ill. 961,497.

VALVE. George G. Little, Gulfport, Miss. 961,505.

BOILER BRACE. Charles Neblett, Bellevue, Ky. 961,522.

VALVE FOR CARBURETERS AND OTHER APPARATUS. Arthur E. England, Boston, Mass., assignor of one-half to James A. Yantis, Malden, Mass., and one-half to Frank P. Woodbury and Willis Du Bois Pulver, Salem, N. H. 961,590.

GATE VALVE. Adalbert W. Fischer, Philadelphia, Penn., assignor to Schutte & Koerting Company, Philadelphia, Penn., a Corporation of Pennsylvania. 961,594.

OSCILLATING PISTON PUMP. Michael Keller, Eureka, Utah. 961,618.

IGNITION MAGNETO MECHANISM FOR AIR STARTED INTERNAL COMBUSTION ENGINES. Charles E. Sargent, Corliss, Wis., assignor to Wisconsin Engine Company, Corliss, Wis., a Corporation of Wisconsin. 961,648.

ROTARY BEARING FOR SUSPENDED SHAFTS. Frederick R. Barnheisel, Chicago, Ill. 961,670.

METHOD OF BURNING POWDERED FUEL Henry R. Barnhurst, Allentown, Penn., assignor of one-half to Henry Gregory Barnhurst, Catasauqua, Penn. 961,672.

SAFETY VALVE. Louis Schutte, Philadelphia, Penn.; Guarante Trust & Safe Deposit Co., and Mary D. Schutte, executors of said Louis Schutte, deceased, assignors to Schutte & Koerting Company, Philadelphia, Penn., a Corporation of Pennsylvania. 961,735.

COMBINED CHECK AND GATE VALVE. John H. Stickel, Handy Harrisville Station, Penn. 961,738.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC FUSE. Anthony J. Smith, New York, N. Y., assignor of one-third to William J. Smith, New York, N. Y. 961,175.

ELECTRICAL BLOCK TRAIN CONTROL SYSTEM. John B. Tonnar, Clayton, Wash. 961,177.

ALLOY FOR ELECTRICAL RESISTANCES. Wilbur B. Driver, East Orange, N. J. 961,217.

ELECTRIC CUT OUT. Thomas E. Murray, New York, N. Y. 961,308.

SYSTEM OF ELECTRICAL DISTRIBUTION. Granville E. Palmer, Boston, Mass. 961,313.

TIME CONTROLLED ELECTRIC SWITCH. Fred M. Tottingham, New York, N. Y., assignor of one-half to Elie J. Moneuse, Sheepshead Bay Station, Brooklyn, N. Y. 961,558.

INSULATOR. William Louis Reusch, Yorkton, Saskatchewan, Canada. 961,646.

ELECTRIC SWITCH. Harry C. Grant, Bayonne, N. J. 961,693.

ARC LAMP. James C. Piper, Indianapolis, Ind. 961,733.

INCANDESCENT-LAMP SYSTEM. Yasuke Kawasaki, Vancouver, British Columbia, Canada. 962,435.

VAPOR ELECTRIC APPARATUS. John T. H. Dempster, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 962,497.

ELECTRIC HAMMER. Burton M. Dutton, Richmond, Va., assignor of one-half to William H. Oxenham, Richmond, Va. 962,499.

RECEPTACLE OR SOCKET FOR ELECTRIC LAMPS. Walter S. Ryan, New York, N. Y., assignor of three-fourths to Louis Hengerer, New York, N. Y. 962,589.

APPARATUS FOR ELECTROPLATING PIPES, ETC. Daniel Hayes Murphy, New Castle, Penn. 962,655.

DYNAMO ELECTRIC MACHINE. Miles Walker, Manchester, England, assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 962,683.

ARC LAMP. Thomas J. Anderson, Houston, Tex., assignor to the Anderson-Lacy Electric Headlight Company, Houston, Tex., a Corporation of Texas. 962,692.

ELECTRIC-MEASURING INSTRUMENT. Robert D. Hickok, Atlanta, Ga. 962,759.

ELECTRIC HEATING UNIT. Herbert M. Smith, Pittsfield, Mass., assignor to General Electric Company, a Corporation of New York. 962,793.

MOTOR CONTROL. Hugo Heymann, Berlin, Germany, assignor to General Electric Company, a Corporation of New York. 962,835.

STARTING DEVICE FOR ELECTRIC MOTORS. David H. Plank and Alfred C. Finney, Schenectady, N. Y., assignors to General Electric Company, a Corporation of New York. 962,856.

STORAGE BATTERY. Charles Frederick Washburn, New London, Conn. 962,870.

THERMO ELECTRIC FURNACE REGULATOR. James H. Bobbitt, Harvard, Neb. 962,884.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York: sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

Vol. 32 NEW YORK, JULY 19, 1910 No. 29

A HABIT often acquired by young engineers and one that should be discouraged from the start is that of continually running to the superintendent with every little trouble that is discovered.

Some do it under the delusion that their superiors will get the impression that they are alert and "onto their job." In such cases, the fallacy needs no comment.

More often, however, it is done through lack of confidence, not that they are incompetent, for they may know very well what to do—perhaps better than the superintendent—but they hate to assume responsibility.

Backbone is as essential to an engineer as water is to a fish. If he does not make use of it in little things it is bound to fail him in an emergency, and then the inevitable happens.

The engineer is supposed to be a specialist in his line, and is hired as such.

Hence he should accept the responsibilities attendant upon his position.

If unwilling to do this it is clearly evident that he has chosen the wrong vocation, and he should lose no time in changing over to some line more fitting to his temperament.

The superintendent has his own duties to perform with a thousand and one things to carry in his mind, and should not be bothered with minor operating details.

He may tolerate this for a short time, but it will soon become a nuisance and he will then look around for an engineer who will go quietly about his business and by his manner impress those about him with the confidence that he can successfully hold down the job and cope with all ordinary difficulties.

True, the superintendent may take more than a passing interest in the engine room, and may have a keen eye for details. When he comes around, let him find everything running smoothly and volunteer information only when asked.

Then show by your answers that you are thoroughly familiar with everything, but *do not talk too much* and make only such statements as you are prepared to back up.

This is one of the many rungs in the ladder to success.

It is applicable not only to the engineer, but to the oiler, fireman and other members of the operating force, as well, in their relations with their superiors.

Cooling the Condensing Water

By Warren O. Rogers

In a plant in Chattanooga spray nozzles and a cooling tower of 5000 gallons capacity per minute are used for this purpose. Johnstown, Penn., also has an interesting installation of spray nozzles.

An uptodate system of cooling condensing water by means of both spray nozzles and cooling tower, is in operation at the power plant of the Chattanooga Power and Light Company, Chattanooga, Tenn. In order that the reader may have a comprehensive idea as to the amount of condensing water used, it will not be amiss to make mention of the main steam units in the plant.

Arranged along one side of the turbine room, Fig. 1, are four 500-kilowatt and two 1000-kilowatt Curtis turbines, 2300-volt, 120-ampere, three-phase, 60-cycle units. The electric generators are all of the same make and output. There is also one 26 and 52 by 48-inch Hamilton-Corliss engine, direct connected to a General Electric direct-current generator of 1000 kilowatts capacity. Two 75-kilowatt Curtis turbines of the horizontal type run at a speed of 2400 revolutions per minute, noncondensing, and are direct connected to General Electric direct-current generators of 75-kilowatts capacity. In addition there is a steam-driven exciter unit, with an engine of the marine type driving a General Electric direct-current generator at a speed of 450 revolutions per minute. In the basement below the condensers are placed the dry-vacuum and motor-driven centrifugal pumps. The large turbines exhaust into Wheeler condensers and the four small units into Worthington condensers.

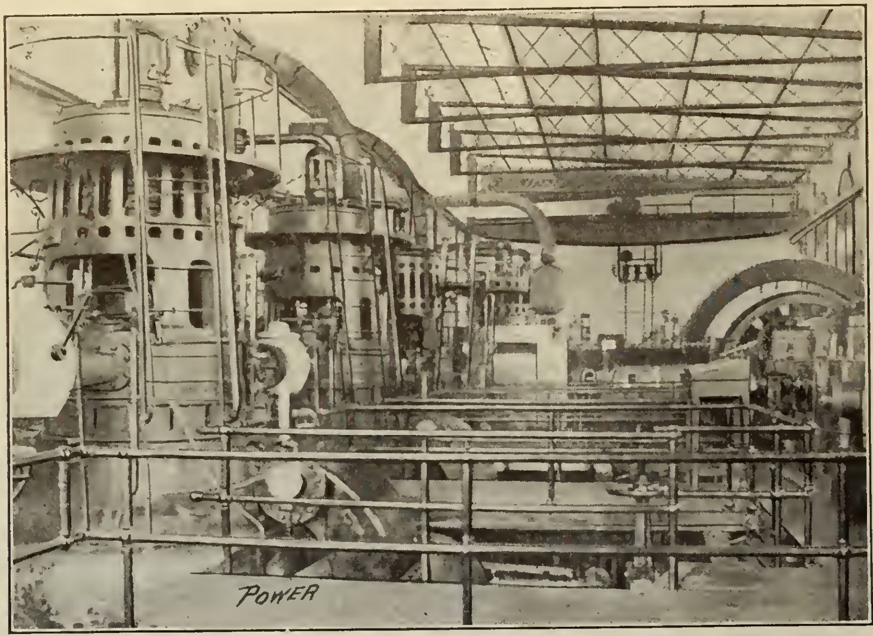


Fig. 1. Showing Steam Units Which are Run Condensing

The auxiliaries all run noncondensing, the steam being delivered to the feedwater heater.

Water for condensing purposes is taken from a pond 100 feet wide by 150 feet long, having an average depth of 4 feet, with the exception that at the intake point it has a depth of 10 feet. At the present time the condensers are furnished with circulating water by means of two steam-driven and one motor-driven centrifugal pumps of 3500 gallons capacity per minute, and one motor-driven centrifugal pump of 4500 gallons capacity per minute. It is the intention of the present chief engineer to substitute the smaller motor-driven unit with a second one of 4500 gallons capacity per minute.

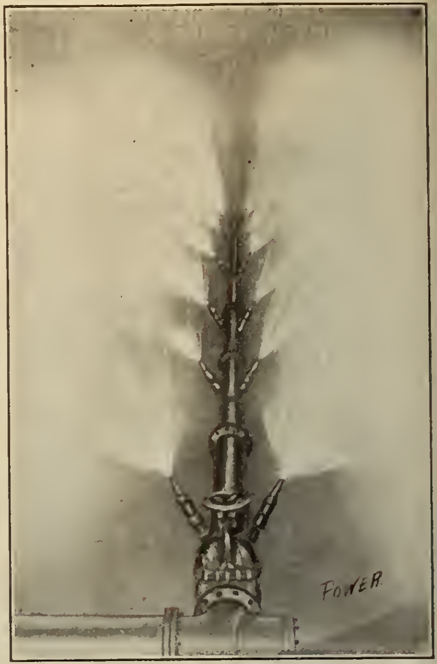


Fig. 3. Arrangement of Spray Nozzles

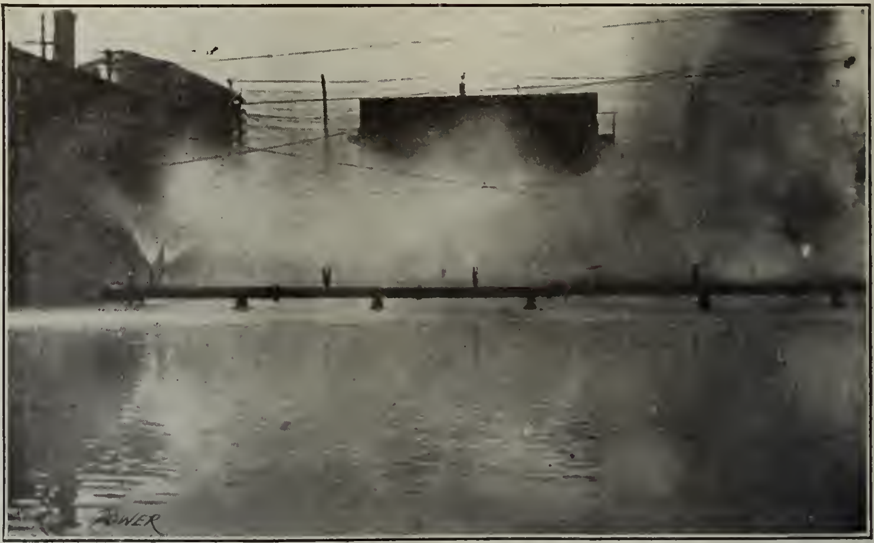


Fig. 2. Spray Nozzles, Cooling Tower and Pond

As the water is forced through the surface condensers it passes to the spray jet nozzles, Figs. 2 and 3, at a pressure of 18 pounds per square inch, and at a temperature of from 125 to 140 degrees,

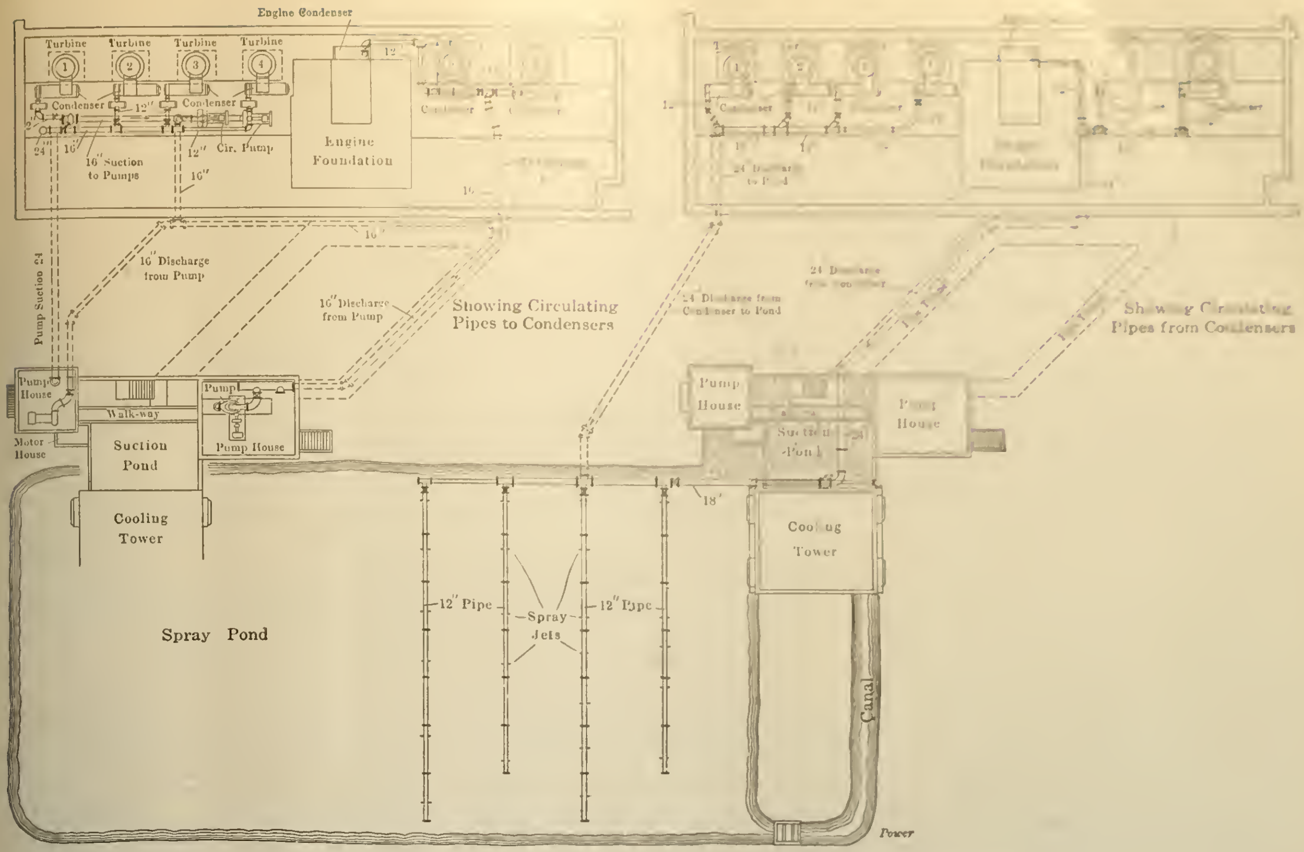


Fig. 4. Diagram of Circulating Pipes to and From the Condensers

the difference in temperature being due to changing load conditions. The water in the hotwell varies in temperature from 140 to 150 degrees, so that it is 10 to 15 degrees hotter than at the nozzles.

In the cooling pond are arranged four branch pipes containing forty-eight 1½-inch Schütte & Koerting spray nozzles, there being 10 nozzles on two lines and 14 nozzles on the remaining two branch pipes. These pipe lines connect with a main header placed along one side of the pond, which in turn connects with the pipe leading from the condensers, as shown in Fig. 4.

Although the water comes to the spray nozzles at a temperature of about 125 degrees, the water in the pond remains at a temperature of about three degrees above the existing temperature of the atmosphere, both winter and summer.

In connection with the spray system of cooling the condensing water, a Wheeler cooling tower capable of handling 5000 gallons of water per minute has been installed. Both systems are used, depending upon the seasons of the year; for instance, during the winter months the spray nozzles and pond are used almost exclusively, a small amount of water being allowed to pass through the cooling tower, but the fans for circulating the air are not in operation. In the summer months, when the atmospheric temperature is high, the cooling tower is used.

Another interesting installation of jet-cooling sprays is in operation at the power plant of the Cambria Iron and Steel Works, Johnstown, Penn. The features of this system are the location of the spray nozzles and the manner of handling the water in cooling approximately 27,000 gallons per hour.

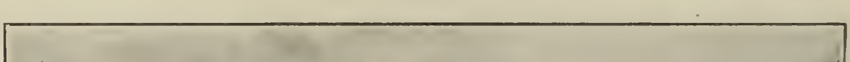


Fig. 5. Spray Nozzles and Cooling Steps

The condensing water is pumped to the spray nozzles through a 24-inch pipe by means of a centrifugal pump located in the power house at the foot of the hill. The horizontal pipe shown in Fig. 5 is made up of twenty 18-inch tees. Each tee is fitted with one 1½-inch Schütte & Koerting spray nozzle. These nozzles are set at two angles, every other nozzle being in line, as shown in Fig. 6. This brings ten of the spray nozzles at an angle of about 60 degrees, and the second set 30 degrees lower. The relative positions of the two sets of nozzles are shown at *A*, Fig. 7.

The spray-nozzle pipe reaches across the upper end of the cooling slope, which is 72 feet 6 inches wide, and 124 feet long. It is constructed of concrete steps located on a sidehill, as shown in Figs. 5 and 6. These concrete steps are 12 inches high by 12 inches wide. On top and at the edge of each step a strip of wood, 2 inches thick by 4 inches wide, is secured.

After the water is discharged from the spray nozzles, it begins its downward course to the bottom of the cooling slope, tumbling from step to step, and, striking the wooden obstructions on the outer surface of the step, is broken up as shown in Figs. 5 and 6. This gives the air a greater opportunity of cooling the water than if the step surface were smooth.

The difference in the temperature of the water is clearly shown by the amount of vapor arising from the water at the top and bottom of the slope. The water comes from the jets at a temperature of 120 degrees and is taken from the reservoir at the bottom of the slope at 80 degrees, a difference of 40 degrees.

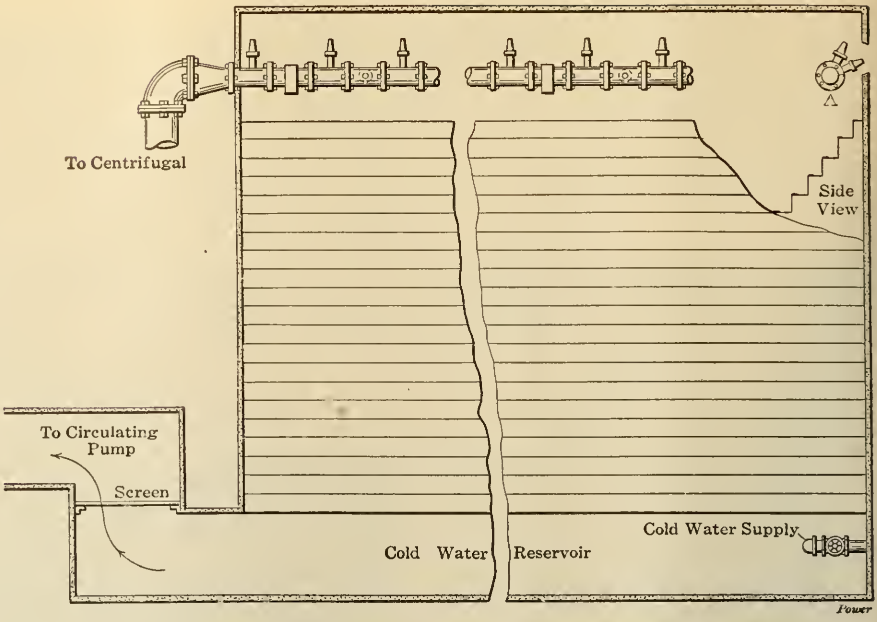


Fig. 7. Diagram of Cooling Steps and Spray Nozzles

At the bottom of the slope is a reservoir extending the entire width of the step and about 10 feet beyond at one end, as shown in Fig. 7. It is 5 feet wide and 3 feet deep at one end, and 4 feet deep at the other. Between this oblong reservoir and the intake to the circulating pump is a set of strainers which prevent the passage of refuse to the centrifugal pumps. The inlet to the circulating pumps is 6 feet deep. The reservoir at the bottom of the slope is provided with a 6-inch outlet for drainage to the sewer. A fresh-water connection is also made with the reservoir for replenishing the water lost by leakage and evaporation. The circulating pumps are in duplicate, and of the two-stage type. The system covers considerable space but since its installation has given the best of satisfaction.

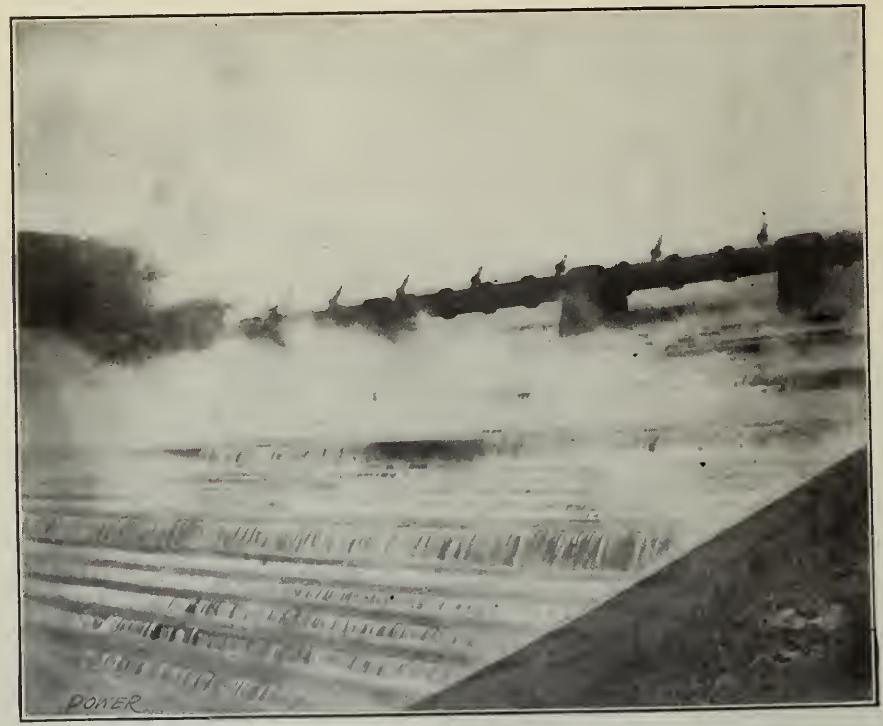


Fig. 6. Closer View of Cooling Steps and Spray Nozzles

Coal Dust Explosions

A paper by Doctor Thorton and Mr. Bowden on "The Ignition of Coal Dust by Electric Flashes" was recently read at Newcastle, England. Experiments have shown that dry coal dust in bulk is a nonconductor of electricity, but when made into a paste with water it is liable to cause a short-circuiting flash if placed between electrodes. If a flash is produced—as by opening a switch—in a cloud of dust an explosion is produced. The character of the dust cloud and the strength and voltage of the current broken affects the readiness with which the dust is ignited, and at medium voltages direct current appears to be more dangerous than alternating. In view of the conclusive experiments on coal-dust explosions produced by the discharge of a cannon, as carried out at Altofts, the results described were to be expected.

O'Brien stepped into the door of Casey's engine room one morning and found that worthy member tinkering with a balky pump and swearing softly to himself. Thinking that he saw a good chance to "put one over" on him, he remarked: "Casey, ye'd bether kim over t' th' wather works tomorrer, they air goin' t' pull aff an intherestin' ingineerin' axperimint."

Casey looked up, and being caught off his guard by the innocent look on O'Brien's face, "bit" easily.

"What air they goin' t' do, O'Brien?" he asked.

With a broad grin O'Brien replied: "They air goin' t' dehorn th' hydraulic ram."

And the next moment he was busy getting out of the door just a few inches ahead of Casey's No. 10 brogan.

How to Make a Flue-Gas Analysis

By Julian C. Smallwood

The Orsat flue-gas analyzing apparatus and the manner of operating it is thoroughly described. Directions are given for preparing the chemical solutions which are used in the apparatus. Valuable hints are offered to assist the beginner to avoid common errors which influence the accuracy of the results.

Now that the steam-engine indicator has become important in maintaining economy of steam distribution, the operating engineer will the more readily believe that a knowledge of what goes on in his furnaces may result in economy of steam manufacture. Since a saving accomplished in the boiler room is of just as much value as a similar one in the engine room, it is well worth while to give some close attention to the place where the stuff is produced.

Without actually testing the boiler we cannot inform ourselves whether or not its heating surface and grate are acting efficiently. It may be easily determined, however, whether or not the coal is being properly burned, and, if not, how much waste is involved. With this knowledge it becomes simple to eliminate the loss. It is the purpose of this article to describe how such determinations are made and what can be done with them.

The process known as combustion is simply the quick combination of a burnable material with oxygen; the combination makes what we call heat. Coal is composed of some burnable material, chiefly carbon, and some that is not so, which appears as ash. The oxygen which the carbon combines with is in the air. What we know as air, however, contains more than oxygen; in fact, the oxygen composes only about one-fifth part of the air, the rest being nitrogen and a small proportion of water vapor and other gases. As a result of the combination of oxygen and carbon, either one or both of two gases may be formed, namely, carbon dioxide and carbon monoxide. These gases are commonly referred to as CO_2 and CO, respectively. As will be explained later, if monoxide is formed the combustion is incomplete and therefore all of the heat possible has not been obtained by burning the coal.

It may be well at this juncture to say something about heat. If it is understood, to start with, that heat is a purely relative thing, it will not be difficult to grasp the way in which it is measured. Thus, if you step into a room in which the atmosphere is at a temperature of 60 degrees on a hot day in summer it is felt to be cool, but if you come into a room having the same temperature in the winter it feels warm. Because of this relativeness of heat we cannot say that any material contains a certain amount of heat in the same sense that it has weight or length. But we can say that a body has a definite amount more heat than another body or has a definite amount more heat in one condition of temperature than another. This enables us to measure heat, and its unit is defined thus: The unit of heat (the British thermal unit, or, more shortly, B.t.u.) is the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit.

Using this unit, if one pound of water at 60 degrees is raised in temperature to 70 degrees it will have 10 B.t.u. more heat in it in the second condition than it had in the first. If ten pounds of water are heated through the same range of temperature 100 B.t.u. are added to it.

If a pound of coal is burned completely in such a way that all of the heat generated is absorbed by a known weight of water, then, by observing the rise of temperature of the water, it may be determined how much heat it is possible for the coal to produce. This sort of thing is done to determine the *heat value* of the coal.

All substances do not absorb equal amounts of heat during the same rise in temperature. Thus, if a pound of iron is heated from 60 to 61 degrees the amount of heat it has taken up is about one-eighth of that which the same weight of water would have absorbed. This quantity, one-eighth, is called the *specific heat* of the iron and for different materials it has different values. In any case the amount of heat in B.t.u. added to or taken away from a body is,

Weight in pounds × *range in temperature* × *specific heat.*

Now in burning coal for a boiler it is essentially necessary, in order to get all of the good out of it, to furnish enough oxygen to combine with all of the carbon so that it may be converted to carbon dioxide. In doing so we have to supply air containing four times as much nitrogen, which absorbs some of the heat of the coal and carries it up the chimney. This cannot be helped, but if we supply too much oxygen and the accompanying nitrogen, so much more heat is carried up the chimney which might otherwise be saved.

Further, and more important, if there is incomplete combustion we are not getting all of the good out of the coal that might be obtained. This loss does not show in the residue—burnable material appearing with the ash does not show *incomplete* combustion but *noncombustion*—but it does show in the flue gas. This, then, is the important thing to watch, and a knowledge of the proportions of its constituents enables one to make many useful calculations.

The most important constituents of flue gas to be determined are: Carbon dioxide (CO_2), showing complete combustion; carbon monoxide (CO), showing incomplete combustion; and oxygen (O), showing an excess of air or leakage in the flue. The remainder may be assumed to be nitrogen (N), although there is a number of other gases, originating from both the coal and the air, which form a small per cent. of the residue.

It may be noted that it is desirable to have the proportion of CO_2 present in the gas reasonably high for this indicates little excess of air. There are a number of automatic CO_2 recorders on the market, which enable one to keep track of the percentage of this gas. They should be regarded, however, as giving only a general indication of the efficiency of combustion for they convey no idea of the possible losses due to incomplete combustion. To judge only by the amount of CO_2 present is about as accurate as it would be to estimate the value of a pile of coins by counting only the gold ones. A general idea may be obtained by so doing, but possibly a very inaccurate one. It is erroneously thought that the higher the percentage of CO_2 indicated by these recorders, the better is the combustion. If the CO_2 is high it may be that too little air is supplied to the coal. If this is the case, incomplete combustion follows and a condition may be obtained where the resulting loss overbalances that from too much air.

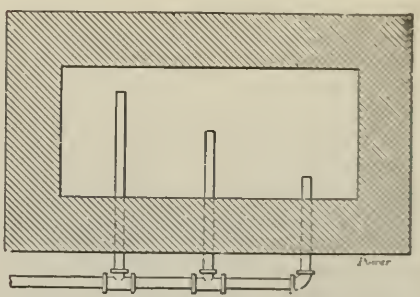


FIG. 1. ARRANGEMENT OF SAMPLING TUBES

The apparatus for analyzing flue gases depends upon the separate absorption of the gases named above by certain chemicals. A measured volume of the gas is brought into contact with one of these chemicals which removes from it the CO_2 but does not take up any of the other constituents. If the difference in volume resulting is divided by the original volume of the flue gas, the result is the proportion of CO_2. In this way the proportions of the three principal constituents are determined. The sum of these quantities subtracted from one gives the part of nitrogen, approximately. If the original volume contains 100 units as, for instance, cubic inches, the number of cubic inches of each gas then represents its per cent. of the total; that is, parts of one hundred.

The first step necessary in making an analysis is to obtain a representative sample of the gas. A short length of iron pipe may be inserted in the uptake so that it extends about half way across; through this the gas may be drawn. As the proportions are apt to vary at different points in the section of the uptake, it is better to use more than one pipe as shown by Fig. 1. Whatever combination of pipes for tapping is used, it should be placed as near as possible to the furnace to prevent dilution by air leaking through the brickwork. Great care should be taken to make the pipe joints tight for the same reason.

To obtain a sample various devices are used, the object of all of them being to collect the gas as it leaves the furnace in such a way as to exclude adulteration with anything else. Fig. 2 illustrates a convenient and easily made apparatus for this purpose. The bottle *S* is used to receive the sample. Its rubber stopper is pierced by glass tubes *T T′*, the shorter one of which is connected by light rubber hose *H* to the pipe leading from the uptake. The other bottle *R* is provided with the glass tubes and the hose connections as shown. The combination is placed upon a shelf near the outlet pipe *P* and a trifle below it. To obtain a sample, *S* is filled completely with water and *R* to about one-quarter of its depth. By blowing into *R* through *B*, water is forced through the hose connecting *R* and *S*, into *S* and up the hose *H* leading to the flue. In this way all of the air may be displaced from *H*. If there is any air in the other hose or in *S*, this, too, will pass through *H*. As soon as the pressure is removed from *B* the water will begin to siphon from *S* into *R*, and thus suck gas from the flue through *H*. The siphoning will continue until the water levels in the two bottles are the same. The gas should be collected very slowly; to do this the tube *B* may be choked with a cork pierced by a small hole. This causes the air to be displaced from *R* slowly, and consequently the water from *S*. When the sample is obtained the pinch cock *C* is closed and the bottles removed *together*. This is done so that when the gas is removed from *S*, water will be drawn in from *R* to supply the diminished volume in *S*.

Having obtained a sample of flue gas the next step is to introduce it into an apparatus for analysis. The most widely used one for this purpose is, perhaps, the Orsat, and this will be described. Referring to Fig. 3, *F* is a glass vessel into which the flue gas is drawn and measured. The gas is then caused to flow from *F* into the flask *D* in which is a chemical capable of absorbing the CO_2. It is then returned to *F*, measured, and the reduction of volume noted. This procedure is repeated for the determination of the other constituents, the vessels *M* and *O* being used.

To describe the operation in more detail, it is first necessary to displace all the air or other gas in the chamber *F* and the tube *T* leading from it. To do this a small bottle *B*, filled with water and connected with *F* by a rubber hose *H* as shown, is elevated. The water then flows from it into *F* and through *T*, but does not enter *D*, *M* or *O* because the glass cocks *d*, *m* and *o* are closed. When water appears at the outlet to the left of *f*, the hose leading from the sample bottle is attached at this point and the pinch cock on it released. Care should be taken in doing this that as little air as possible be entrained in the free end of the hose. The small bottle *B* is now lowered to the position indicated by the dotted lines, allowing the water in *F* to flow back into it. To fill the space vacated by this water, the flue gas enters through *T* into *F*, the whole volume of which should be filled with the gas. The cock *f* is now closed and the sampling bottle and connecting hose removed. One minute should be allowed for the walls of *F* to drain. The bottle *B* is then raised a trifle until the water level reaches the lowest graduation, 100. The pinch cock *p* is now closed. As the gas has been compressed by this rise of water level, the pressure should be relieved by opening *f* for an instant, allowing a little of the gas to escape. It should be noticed that if the pinch cock *p* is open the gas in *F* will occupy more or less volume according to whether it is expanded or compressed by lowering or elevating the bottle *B*. It is therefore necessary when reading the volume to have the water level in *B* the same as that in *F*, and this may be accomplished by holding the bottle *B* so that the levels coincide when the readings are made.

We have now secured a sample of gas in the apparatus which is neither expanded nor compressed. The amount of it by the graduated scale is 100 "c.c.," c.c. referring to the French unit of volume, cubic centimeters. This unit is about one-sixteenth of a cubic inch.

The next step is to introduce the sample into the flask *D*. To do this *d* is opened; the bottle *B* is elevated until the water level in *F* rises to the mark *n* and the pinch cock *p* is closed. The water entering *F* forces the gas into *D*; the entering gas in turn presses the chemical out of *D* up into *D′* through the connecting pipe at the bottom. Flask *D* contains a number of glass tubes which remain wet with the chemical, thus furnishing a large area to absorb the CO_2. The liquid entering *D′* thereby displaces the air previously contained. This air passes into the rubber bag *R* and expands it. The function of this bag is, first, to keep the chemical from contact with the outside air (such contact would rapidly use it up); and, second, to provide a means, if necessary, to force the chemical back into *D*.

Having filled *D* with the sample, it should remain there at least three minutes to allow time for the thorough absorption of the CO_2. The gas is then returned to *F* by lowering the bottle *B* and opening the pinch cock *p*. If necessary, the rubber bag is pressed so that the level of the chemical in *D* returns to the mark *a*. When this is done the cock *d* is closed; the bottle *B* is held alongside of *F* so that the water levels are the same, and a reading taken. If, for example, the reading is 87 c.c., the volume of CO_2 that *was* in the sample occupied,

$$100 \text{ c.c.} - 87 \text{ c.c.} = 13 \text{ c.c.}$$

or 13 per cent. of the entire volume.

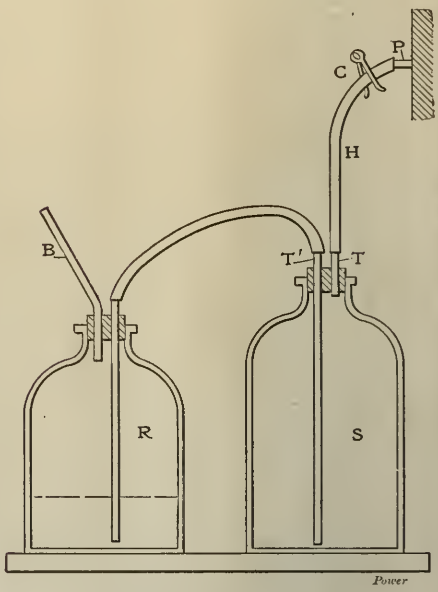


Fig. 2. Sample-collecting Apparatus

The procedure described is repeated in each of the flasks *O* and *M*. The chemical in *O* removes the oxygen from the gas; that in *M*, the CO. What is left is called nitrogen. The difference between the volumes of the gas before and after entering *O* gives the volume of oxygen. In a similar manner the volume of CO is obtained.

The chemicals used are as follows:

For carbon dioxide, one-half pound of

caustic potash dissolved in one pint of water. The caustic potash is a solid and is sold in the form of sticks. It should be kept in a corked bottle and should not be freely handled as it attacks the flesh. This chemical, as used, will absorb forty times its own volume of CO_2.

For oxygen, one pint of the caustic potash solution as just described, to which is added five-eighths of an ounce, avoirdupois, of pyrogallic acid. This is a powder and may be poured into a bottle containing the measured caustic potash and water. This bottle should then be kept corked to prevent the contents spoiling. The absorptive capacity of the mixture is twenty times its own volume.

For carbon monoxide, the necessary solution may be prepared by placing in an open-mouthed pint bottle, such as one used for milk, one ounce of copper oxide and a bundle of copper wire a little shorter than the bottle in length and in diameter about half that of the bottle. To this should be added commercial hydrochloric acid sufficient to cover the wire. The bottle is then corked. When the solution becomes colorless it is ready for use. The absorptive capacity is twice the volume of the mixture.

When pouring these chemicals into the proper vessels of the apparatus, the rubber bags and the corks are removed from them, and the cocks *d, o, m,* and *f*, Fig. 3, opened. The chemicals should be poured in until the level is about half way up in the connected flasks as shown at *k k*. By manipulating the bottle *B* all of the CO solution, for instance, may be drawn into the flask *M*, the cocks *d, o* and *f* being closed and *m* open. The cork and attached rubber bag may then be replaced and the cock *m* closed.

The foregoing outlines the procedure of flue-gas analysis by the Orsat apparatus. A little study of and practice with the actual apparatus will result in its complete understanding. There are, however, some common mistakes easily made and not readily detected that may influence the results materially. The following gives a few points in practical operation:

Carbon dioxide should be removed first, oxygen next and monoxide last. If this order is not kept the results will be wrong. The chemical that removes oxygen will absorb carbon dioxide also; therefore the latter should not be present when oxygen is dealt with. A similar error would be produced by testing for carbon monoxide first.

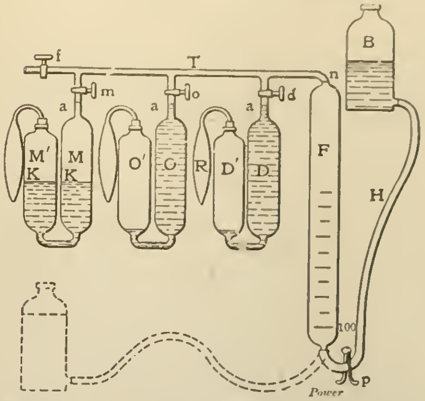


Fig. 3. Diagrammatic Arrangement of the Orsat Apparatus

The analysis should be made at a uniform temperature of not less than 60 degrees. The apparatus, when in use, should not be exposed to a changing temperature, or placed where drafts or sun may strike it. The resulting changes of temperature cause expansion or contraction of the gas under test, and any such change in volume will make the results untrustworthy.

The water that is brought into contact with the gas should be prevented from absorbing any part of it. Water has the property of absorbing the gases and if any part of a sample is removed by water, the analysis will not represent the gas as it came from the flue. To prevent the water used in the apparatus from affecting the results, it should be previously saturated with flue gas so that it is incapable of absorbing more. This may be done by connecting the sample bottle, containing a charge, with the apparatus and alternately raising and lowering the bottle *B*, Fig. 3, so that the gas is worked in and out of chamber *F*. The water used in the sample bottle may be saturated with the gas for the same purpose; or, if preferred, a drop of oil introduced into the sample bottle will cover the surface of the water in it and separate it from the gas.

When removing the sample bottle *S*, Fig. 2, from the flue it is well to siphon back into it some of the water from *R*, the pinch-cock *C* being closed. This puts the gas in *S* under pressure and if any leakage takes place past the rubber stopper it will be outward and will not affect the sample as would be the case if air leaked inward.

After the chemicals have been used for some time they become weak. If a record is kept of the volume of the gas each has absorbed, it will be known when its absorptive capacity is reached; it should then be discarded.

It is well, after making a measurement, to run the gas back into contact with the chemical it has just left, and allow it to remain a minute. Upon measuring again, if the volume is the same, the result is satisfactory. If, however, a check reading of this kind shows less volume than the previous one, it must be because the gas has not had sufficient contact with the chemical to remove all of the constituent being measured. The treatment should then be repeated until a constant volume is obtained. This precaution is particularly necessary in determining oxygen as the solution acts slowly.

In measuring the water level in *F*, Fig. 3, it will be noticed that where the water meets the sides of the vessel it clings to it and is slightly higher than the level at the middle; that is, a rounded surface is formed instead of a flat one. The lowest point of this surface should be sighted on the graduating lines and the corresponding graduation read.

Oil Grooves and Bearing Lubrication

By William Kavanagh

A hollow crank pin and one of the means by which it is oiled is shown in the dotted lines, Fig 1. A hole is bored centrally and longitudinally up to the collar *C*, as indicated by the dotted lines. The diameter of hole *A* is of sufficient size to admit the entrance of a brass tube *B*, which is filled with oil or other lubricant, the length of the tube being nearly as long as the hole in *A* in the crank pin. Both ends of the brass tube are closed, and to one end is attached a small ring, so that the tube can be withdrawn by means of a small hook. Three holes are drilled in the tube as shown at *X*. The end holes are vent holes, the center one being the filling hole.

In order to fill the tube with oil the screw plug at *D* is removed and the tube loaded with oil or other lubricant. The plug *B* should be screwed up medium hard. The oil from the tube finds its way out through the holes *X*, through the hole *E*, and then passes in between the pin and brasses. The tube is filled before starting the engine.

When the engine is running over, the tendency of the crank-pin box on the strap end is to wipe off nearly all of the lubricant, especially when the boxes are not properly fitted, as shown in Fig. 2, which shows the sharp edges of the brass at *E E*.

When the engine is running under, the brass on the connecting-rod end acts as a wiper in the same manner if not properly fitted.

At *F F*, Fig. 2, the sharp edges shown at *E E* are removed for either direction of rotation.

A great variety of oil channels are cut in boxes, according to different ideas. Some fitters prefer a cross, shown at *A* in Fig. 3, others a V or a T channel, shown at *B* and *C*, respectively.

Where high-speed bearings give trouble from heating, a double or single spiral may be cut in the shaft, which, if properly done, will aid materially in reducing heat. In cutting the spiral, care should be taken not to extend it outside of the box, as in such case oil will be thrown. One way of cutting a spiral is shown in Fig. 4, the lines *A* and *B* determining how near the end of the box the spiral may be cut.

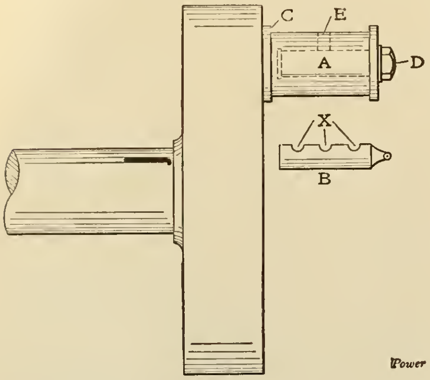


Fig. 1

Most ring oil bearings that heat do so from two or three causes aside from excessive friction or load. The rings may cease rotating; lack of oil in the oil well, the oil becoming dirty, or the use of a heavy oil, especially if there has been any cylinder oil added to the regular machine oil used in the bearings, will cause heating. When the oil is thick and sluggish it stops the rings rotating and prevents lubrication. When a ring-oiled bearing runs hot, perhaps the oil is worn out and should be changed. All of the old oil must be run out and the well washed out with some good mineral oil. After filling the well with clean oil, it will be found that the bearing, in nearly every case, will cease running warm.

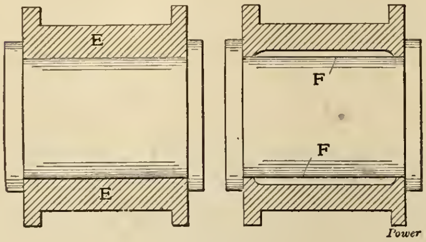


Fig. 2

Babbitt metal melts easily and at a sufficient temperature becomes soft and is easily crowded out of the box in the direction of strain. With belt-driven shafts the babbitt metal will become thin in the direction of the pull of the belt, and, therefore, the soft-metal bearing is not very reliable, especially where a bearing must run. In the writer's plant all soft-metal bearings were consigned to the scrap heap long ago, and only hard brass sleeves are used. They permit of a high temperature before trouble is given by heating and, besides, give a warning which can be heard in time to prevent trouble. If the oiler wishes to save about 50 per cent. of his labor and waste, let him use waste after wetting in clean water; the rapidity with which the oil is wiped off a surface is astonishing.

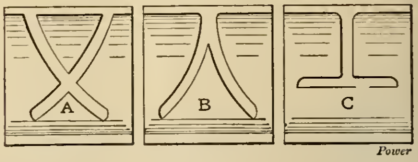


Fig. 3

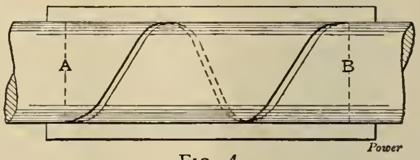


Fig. 4

Leather for Belting Purposes

By Frederic Dannerth

> *The best belting is obtained from a particular section of the hide and must go through numerous processes before it is suitable for the transmission of power. The various processes which transform the raw hide into the finished belt are enumerated, and some valuable hints are given on the use of belt dressing and the care of belts.*

Before entering upon a discussion of the requirements of leather belting it may be of interest to review briefly the processes through which such leather passes before it is in the proper condition to be made into belting.

First, the skin is removed from the animal by a machine and is soaked in lime water to loosen the hair by swelling the pores. Then it is beamed, that is, all the hair and outer skin are removed by means of blunt knives operated by hand, and after being washed it is bated; this is usually accomplished by the use of lactic acid, its object being to remove any lime which may remain on the skin. It is next pickled in a bath containing sulphuric acid and sodium chloride, which treatment enables the fiber to take up the tanning material much better than would otherwise be the case. The skin is tanned in a bath containing white-oak bark, at a temperature of 80 degrees Fahrenheit and is then laid away for a period of about six months, in ground bark with just enough water to cover it. In trimming the leather, all of the shoulder is removed to within 4 feet from the base of the tail, as is also the belly, the loose character of the fiber rendering this leather unsuitable for belting.

After being removed from the tanning process and trimmed, the leather is reeled in fresh water and shaved by a machine which removes all the loose particles of flesh projecting from the surface. A scouring process removes the "bloom" caused by excess of tanning material and also puts the leather in condition to take up the necessary grease. Next it is jacked and sammied and is placed on stretching frames where it remains until perfectly dry. Upon being removed from the frames it is cut into the desired widths with an allowance for trimming. Here it might be mentioned that all leather for first-class belting must be cut from "center stock" which should show the demarcation produced by the backbone and which should not be wider than 30 inches nor longer than 4½ feet.

A scarfing machine and a lapping machine are both used to put a lap on the leather, while another machine removes the grease from the flesh side; this latter operation is called whitening and is very necessary as the cement otherwise would not grip the leather. The several pieces of leather are now ready to be cemented together, an operation requiring a pressure of about 20 tons.

If the finished belt is to be wider than 30 inches, it is necessary to make it two or three ply. For example, a 60-inch belt might be constructed as follows: The first ply, consisting of three pieces of leather each 20 inches wide and lapped longitudinally, the length of this section being about 4 feet; the second ply, consisting of one 36-inch center united on each side with one 12-inch center, all 4 feet long; while the third ply consists of two 30-inch centers lapped longitudinally and, like the others, 4 feet long. The accompanying illustration may help to make this clear.

"Belting butt" is the term applied to

that portion of the hide which remains after those portions of the hide unsuitable for belting (neck, belly and flank) have been cut away, this trimming constituting about half the weight of the hide, as before mentioned. The stuffing of the leather with oils or grease is carried out after the leather has been placed on the stretching frames and is done to render the butt flexible. In some cases this stuffing is supplemented with a loading of glucose, sugar of lead and fats. The result of this is that we may find two pieces of single belting, one weighing 26 and the other 18 ounces per square foot. High-grade belting should not wrinkle nor "pipe" if bent back and forth repeatedly at a sharp angle. Fat wrinkles are such as appear at the ends of laps and are not objectionable; in fact, they are seen in the best grades of belting. Another point to be observed when examining belting is its "trueness," as a good belt should lie quite flat if placed on the floor and should be of uniform thickness. By carefully sorting the butts it becomes easy for the belt manufacturer to produce various grades of belting, but after the belt is once made up it is difficult to recognize poor or low-grade beltings at a casual glance.

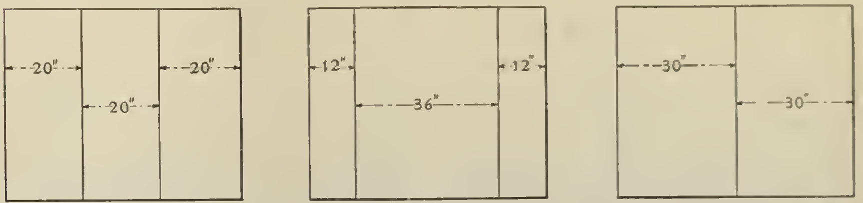
After once having purchased the belt and placed it in position it will be noticed that there are two principal causes of loss in power transmission, namely: *tightness* and *slippage of belts.*

The slippage may be ascertained by taking the speed of the various lines of shafting at full load and again at no load. *Chase* has calculated the actual money loss for a mill with 1000 looms, turning out 5000 pieces of print cloth per week, with a belt slippage of 29 per cent. prevailing, and he finds the weekly loss to be 100 pieces valued at $2 each. One of the immediate results of a tight belt is a hot bearing; hence, many engineers run the belts so that slippage is just overcome and then, at a certain time of the year, a systematic shortening of the belts in the mill is undertaken.

If any belt is wasting power in friction in a manner which can be easily remedied, that belt is obviously too tight. A belt may be considered sufficiently tight if it does not slip and has sufficient pulley contact to keep within the strength of the leather and withstand the strain placed upon it.

Belts cut from center stock run smoothest and straightest, and are therefore recommended for use with high speeds (30 feet per second and over). In 8-inch widths such belts can be produced in thicknesses of 6/32 to 7/32 of an inch; this is due to the fact that the hide along the line of the backbone is rather thin. Belts which are to run at slower speeds may be cut from the part directly adjoining the absolute center, for in this case the tension on the fibers in the different parts of the belt will have opportunity to equalize. Such belts not cut from absolute centers should not be 8 inches in width, otherwise the tension on the two edges of the belt will be unequal and the belt will not run true. In cases where loose pulleys and fixed pulleys are employed, the greater tension on one edge of the belt is utilized by placing the belt in such a position on the loose pulley that the tighter edge adjoins the fixed pulley. The advantage of this lies in the fact that it is easy to run the belt on to the driving pulley but difficult to throw it back to the loose pulley.

The quality of a belt depends upon its strength, durability and its driving power, but these cannot be determined by the thickness or the weight. The ability to transmit power is determined by the tensile strength of the leather and its elasticity as well as the character of the drive. The strongest part of leather is located about one-third the distance from the flesh side but the action on the belt is greater on the inner half of the leather. For this reason and also because the leather is thus used in its natural position, the flesh side of the leather is placed preferably against the pulley.



First Ply Second Ply Third Ply

Method of Cutting Plies for 60-inch Belt

There is at this time a distinct difference of opinion among engineers regarding the value of belt dressings. While some insist that all belting and especially leather belting should be treated periodically with such preparations, there are others who maintain that they should be avoided or used sparingly. It seems, however, that this second group base their statements upon their experience with low-grade dressings containing rosin and similar harmful substances. *S. von Bach* recommends that leather belts be degreased periodically and that they should then be given a small amount of belt dressing free from rosin.

Belts after being installed attract dust which prevents proper contact and so interferes with adhesion, or they become glazed and then slip. These defects may be remedied temporarily by tightening the belt, an operation which results in greater strain on the bearings and a misuse of power for overcoming the increased friction. Commercial belt dressings are used for making the belt stick to the pulleys and to preserve the belt. The substances used in manufacturing these preparations include stearic acid, degras, rosin, castor oil and fish oils.

Properly curried leather belting, used under the right conditions, will seldom become dry and hard. In certain cases a dressing containing 70 per cent. tallow (free from acid and having a melting point of about 95 degrees Fahrenheit) mixed with 30 per cent. castor oil has given excellent results. If it is found necessary to use a preparation of this kind it should be used sparingly as the use of oil and grease tends to make the leather stretch. Such a mixture as indicated above will soften the fibers and at the same time increase the adhesion of the belt. If it is desired only to correct slipping this may be done by applying a few drops of castor oil to the pulley, without fear of injuring the leather.

The small amount of slip which is bound to occur on the pulley side of the belting produces heat, which causes that side to become dry and hence further slipping. Careful tests seem to indicate that the coefficient of friction is influenced directly by the absence of air or other substances between the belt and the pulley and that the best results are obtained when the belt is in direct contact with the pulley. For this reason the belt dressing should be used very sparingly in order to avoid any reduction in the gripping power of the belt. All dressings of a sticky or oily nature will attract dust and this gathers in clots on the pulley or on the belt with the result that the belt runs unevenly, at the same time decreasing the area of contact. If a newly installed belt has stretched excessively, it should not be shortened immediately but should be treated on the pulley side with beef tallow. This causes the belt to slip somewhat but the frictional heat soon liquefies the fat which is then gradually absorbed by the leather. The swelling of the leather due to this absorption causes a shortening of from 1 to 2 per cent.

In conclusion it may be said that a good dressing should not decompose, become rancid with age, be influenced by moisture, dry up nor cause the belt to stick to the pulley, but should soften the pulley side of the belt, penetrate the belt, leaving no film on the surface, lubricate and protect the fibers from crumbling, prevent the generation of electricity by the belt and increase the coefficient of friction. Also, it should contain no mineral acids. A dressing fulfilling these specifications will not injure the belt and may be used with impunity.

Notes on Western Anthracite Coal

By A. G. Christie

> *As is well known, some types of mechanical stoker are better adapted to burn certain kinds of coal than others. This is confirmed in the present case where much difficulty was experienced in attempting to burn No. 2 buckwheat coal on a Roney stoker while a mixture consisting of No. 1 buckwheat and pea coal was found to give satisfactory results.*

Pennsylvania anthracite coal has been used for so long a time and under so many varying conditions that engineers now know what type of furnace to construct, what class of grates to use, and how the fire should be handled in order to give the best efficiency with any given size of coal. This coal is almost free from hydrocarbons and hence burns practically without smoke. In mining, this coal usually has fairly well defined partings so that the bone can be separated readily from the good coal; the ash as a rule is friable and forms a very small percentage of the coal. The anthracite coals found in western Canada, and particularly in Alberta, have quite different characteristics from the Pennsylvania anthracite.

The Alberta coalfields, as far as they have been prospected, cover an area about 400 miles long and from 100 to 200 miles wide. Almost every variety of coal is found within this area from the anthracite coals of the Bow river section and the newly discovered Brazeau river sections to the brown lignite of the prairie. The coking coals of the Crow's Nest district are well known and are used to a great extent by the smelters in British Columbia, Idaho and Montana.

At present the only anthracite deposits being worked are located at Bankhead, Alberta, a few miles from Banff, the famous scenic resort. These deposits lie under the eastern slope of Cascade mountain and, as is common in most Western mines, only the seams of the upper formations are being worked. This may account for the fact that the coal is more truly a semi-anthracite than a pure anthracite, better grades of anthracite being known to exist in the lower seams. The coal is hard and does not weather, nor does it possess the shiny black appearance of Pennsylvania anthracite. At times there is considerable shale, often in very thin seams in the coal itself. An analysis of the coal is as follows:

	Per Cent.
Moisture	2.1
Volatile matter	8.1
Fixed carbon	79.3
Ash	10.5
Sulphur	0.6

Attention is called to the high ash content and the per cent. of volatile matter.

A process has been developed at this mine by means of which all the small coal and dust is briquetted, the coal-tar pitch being used as a binder. The briquets are "pillow shaped," ¾ inch long and about the same width and thickness. They are excellent for domestic purposes, leave little ash and command a high price.

Attempts to burn this Bankhead anthracite were made at the plant of a large cement company located within the Rocky mountains, about 25 miles from Bankhead. A very fine grade of semi-bituminous steam coal was mined in great quantities only ten miles away at Canmore, but the whole output of the mines is taken by the Canadian Pacific Railway for use on its locomotives.

The boiler plant consisted of seven Babcock & Wilcox boilers, set in batteries of two each with an extra boiler set alone, each with 4980 square feet of heating surface and equipped with superheaters to give 100 degrees Fahrenheit superheat. These boilers furnished steam for driving the turbo-alternators and reciprocating engines which produced power for the mill and quarry.

When the question of furnaces came up, it was decided to install Roney mechanical stokers for the following reasons: First, the manufacturers were willing to guarantee that these stokers would burn Bankhead No. 2 buckwheat coal satisfactorily and produce the rated capacity of the boilers; second, the cement company owned large coal areas of both anthracite and soft coal and had plans under consideration for opening these, hence, a furnace was desired which could handle either class of coal, and the Roney stoker was deemed capable of doing this; the third and most important reason was the belief that cheaper labor could be used in the operation of mechanical stokers and with more satisfactory results. This judgment was fully justified in actual operation. At first only four boilers were equipped with the old style of stoker having flat plates, though later the remaining three boilers were equipped with the new-style stoker having vertical fins pinned to a yoke. The arch over the furnace was extended by two rows of brick in excess of what was then considered good practice.

After passing through the regular baffles the flue gases passed down the back of the boiler and into a concrete smoke flue which was built under the boilers and led to either of two 16-foot induced-draft fans placed at one end of the boiler room. These fans were engine driven and discharged the gases through a short 10-foot steel stack. The engines were of a slow-speed type, direct-connected to the fans, either of which was capable of handling all the boilers. The speed of the fan was governed by a pressure-regulating valve on the steam line to the engine, which valve cut off the steam as the pressure in the boilers increased; by altering this regulating valve any desired draft could be obtained.

The illustration represents a section through the boiler room showing the relative arrangement of coal bunkers, boilers, stoker settings and smoke flues. Coal was drawn from chutes in the bottom of the bunker, transported on cars to the fronts of the boilers, and then shoveled onto the stokers by hand. The ashes were shoveled from the ashpits and loaded on similar cars which were pushed out onto the ash dump.

As is obvious, this arrangement necessitated a large amount of manual labor; in fact, the operating force outside of the boiler cleaners averaged eight men to a shift. It may be of interest to add that one man looked after the whole of the engine equipment and condensing machinery together with the switchboard, except for occasional supervision by the engineer in charge. The load on the plant, though steady compared to many railway central stations, varied considerably at all hours, and without notice; the day load was much heavier than that at night as the quarries, crushers and unloading conveyers were then in operation.

From Fig. 1 it will be seen that the only method of removing ashes was to shovel them from the bottom of the ashpit by a long-handled shovel. This removal of ash required a great deal of labor on account of the character of the ash. As has been remarked, the coal always seemed to carry considerable shale and bone. In burning, this seemed to sinter the various pieces of coal, and formed large masses of porous ash and clinker often of large size but as a rule very light in weight. However, the ash equaled volumetrically about 50 per cent. of the coal fired.

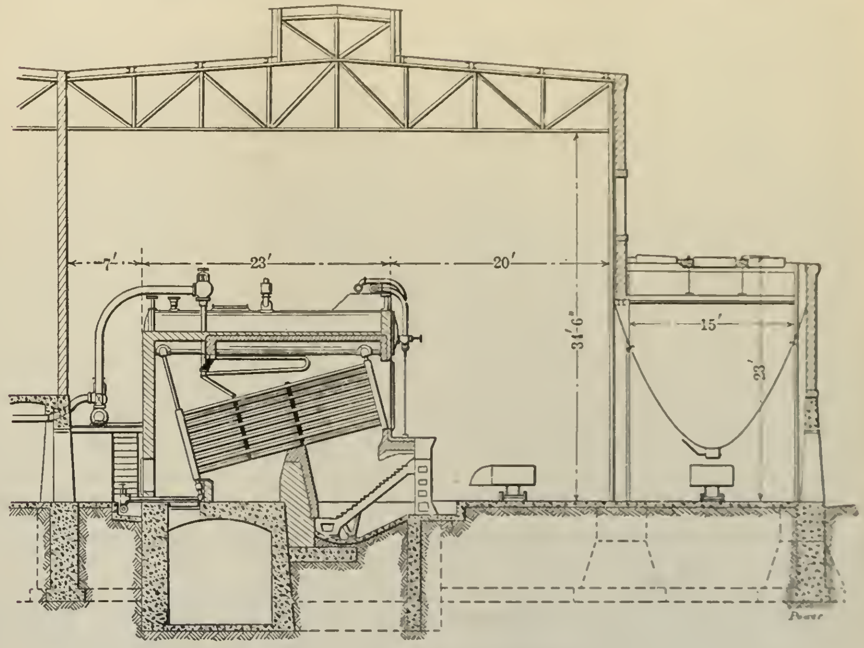
This outline gives a rough idea of the conditions existing in the plant. One point further needs to be emphasized and that is the poor class of labor available, which consisted either of newly arrived immigrants or of what is known in the West as "hobo" labor. The first class consisted largely of men from the British Isles who retained that inborn trait of altering habits and adjusting themselves slowly to new conditions. Many of the men employed in the boiler room were firing for the first time, and at times only foreigners were available, although some of these became very adept firemen. As labor was scarce, men of the hobo class were always being taken on. Often these men could fire well, but they could never be depended upon to stick to a job and usually left as soon as they had accumulated a little money. Hence, labor costs were high and results often discouraging as the operating staff were always breaking in new firemen, at which time the fires were often full of holes and large quantities of good coal were lost in cleaning the fires.

Naturally when the plant was first started, the cheapest coal, No. 2 buckwheat was tried. At first, difficulty was experienced in getting this coal to take fire all over the top of the grates; this trouble was attributed to the volatile matter being small and easily burned only at a high temperature.

The clinkering action of the ash, which has already been referred to, did not take place unless the fire was good and the temperature high. As there was no coking and as the draft necessary to burn this size of coal was high, trouble was encountered with the coal slipping to the dump grates at the bottom of the stoker. This produced several undesirable results: First, the coal piled up to such a depth at the bottom of the furnace that little combustion took place; second, as a result of the air passages being almost entirely cut off at the bottom of the furnace, the draft became more intense through the upper portion, and more of this small coal was carried to the bottom. Unless the draft were checked the grates would be swept bare and large volumes of cold air would pass into the furnace. It was impossible to put on any grate motion to provide air, as is usual on these stokers using soft coals, for this only made the coal work more quickly to the bottom of the furnace. The most serious trouble was the loss through unburned coal which frequently necessitated picking over the dump and returning the unburned portion to the hoppers. Another very troublesome feature was the formation of huge clinkers, caused by fresh burning coal sliding over partially burned-out ash; these were often hard to break up. The removal of these clinkers almost always required the opening of the side door and this produced a serious cooling effect upon the boiler in spite of a closed damper. Very often the poking necessary to remove these clinkers caused a slide of fresh coal from above.

It seemed probable that these troubles could be overcome if the angle of the grates with the horizontal were changed from 40 degrees to about 25 degrees. Accordingly, the grates under one boiler were lowered slightly in front and raised about a foot at the rear. This, however, did not prove as successful as had been anticipated; the very fine coal sifted back through the flat grate bars, for on account of the raising of the rear end these dipped down toward the front of the furnace. This loss almost equaled that met with in dumping at the original angle. Besides the grates had been made too flat, for the coal did not work down easily without considerable poking.

While this experimenting was going on, the mill was gradually being placed in operation. Since at all times one or more of the four boilers had to be out of service for cleaning, it was found necessary to install hand-fired grates under the single boiler, pending the arrival of the remaining mechanical stokers. As this was only a temporary arrangement, flat stationary grate bars were installed. Here, as in driers in different parts of the cement mill, it was found extremely difficult to keep a good fire and then only with the most skilful firing. Holes formed very readily, and after cleaning, it was hard to prevent this very fine coal from sifting through into the ashpit and leaving the grate bars bare.



SECTION THROUGH BOILER PLANT

About this time the coal company, which had had its briquetting plant in operation for some time, found that it was more profitable to make the No. 2 buckwheat into briquets than to sell it for power purposes. As the cement company had no fixed contract, the supply of No. 2 buckwheat was gradually reduced, and No. 1 buckwheat was supplied in its place. This cost 25 cents more a ton than the No. 2 buckwheat, but it was cheaper coal even then when completeness of combustion was considered. But during inactive periods at the mine in the spring and early summer, the cement company was forced to buy pea coal along with the No. 1 buckwheat then available. The firemen soon learned to burn the latter with very satisfactory results, while a mixture of two parts of No. 1 buckwheat and one of pea proved to be the best firing mixture.

When the second set of stokers of the improved type arrived, No. 2 buckwheat was again tried, and while better success was obtained than on the old type, the results were not satisfactory enough to warrant its further use owing to the sifting of the coal through the grate bars.

The methods employed in handling the fires with No. 1 buckwheat and pea coal were as follows: The dumps were always made with a very hot fire on the upper grates and the firebrick arch and side walls intensely hot. The damper was nearly closed and cooling thus avoided as much as possible during a dump. As soon as the ash and clinkers on the dump grates had been poked out, the grates were pulled up, and the clinker, which had been loosened previous to dumping, was pushed down with a slice bar. At the same time a heavy feed of fresh coal was put on the pusher bar, and the grates were always kept covered with coal. The presence of the hot firebrick arch caused a rapid ignition of the fresh coal and a good fire could be had within a short time. Then from time to time the fireman would break up the clinker and work it toward the dump grates, at the same time maintaining a steady feed of fresh coal. This would continue until dumping was again necessary. These sizes of coal were large enough to lie on the grates at all times, and did not avalanche under heavy draft. It was also noted that fewer and smaller-sized clinkers were formed on the improved type of stoker.

No. 1 buckwheat coal burns with considerable yellow flame but without smoke if the furnace temperature is kept hot. It has a tendency to form clinkers often of large size but not of a tenacious nature. Large quantities of dust are carried over the bridgewall and even into the fans. A very fine white dust collects on the lower side of the tubes above the furnace, sometimes to a depth of ½ inch in three weeks. This reduces the steaming capacity and requires careful and frequent cleaning.

The fans were usually run at a speed to maintain a draft of 0.6 to 0.8 inch of water at the fans and the dampers were regulated to give a draft of 0.3 to 0.5 inch above the grates. This was found to give the best results in ordinary operation, as an increase of draft tended to make the coal slide down the grates.

The daily records show that the average amount of coal burned ranged from 15 to 20 pounds per square foot of grate surface on the stoker. No tests were run on the boilers, but it is reasonable to assume that their average load under these conditions seldom exceeded their full rating.

During the early part of the summer the supply of coal from Bankhead became insufficient on account of the partial closing down of the mine. It was then decided that the cement company should open up some of its own bituminous-coal properties. While this mine was being opened, coal was secured from many sources, and a brief reference to these coals and their burning qualities on Roney stokers may be of interest.

Several carloads of Canmore coal were secured. This coal is used exclusively in the locomotives on the Canadian Pacific railway, and has proved an excellent steam coal. It is a semi-bituminous, noncoking coal and slacks on prolonged exposure to air. As the coal secured was principally slack and did not coke, it was not found to be a suitable coal for mechanical stokers. Another peculiarity noticed was that if Canmore coal became mixed with Bankhead coal in the furnace, hard clinkers of large size usually resulted.

For some time coal from Coal Creek and other mines in the Crow's Nest district was burned under the boilers. These are bituminous coking coals especially suitable for Roney stokers. Heavy drafts and high combustion per square foot of grate could be maintained even with slack coal. This coal filled every requirement had the freight charges not made the cost prohibitive.

Another coal used was known as "Hillcrest coal," coming from the Lethbridge district. This was a noncoking, bituminous coal which gave an intensely hot fire with little smoke and left the least ash of any of the coals. The ash was of a powdery nature and gave little trouble. This was probably the most satisfactory coal of all, but was too expensive when freight charges were paid.

The cement company, in developing its bituminous-coal property, opened up a mine in an outcropping which was very accessible for mining and shipping without the employment of hoists and other machinery. This coal proved to be a black lignite with high moisture and high volatile contents. It produced large volumes of dense black smoke, but was found to burn better on the new style of stoker than on the old style. It ran very high in ash and showed tendencies to coke. The coal, however, improved greatly in quality as the mine became deeper. Seams of high-grade coking coal are known to underlie the one being developed, but have not as yet been touched. When coals of lignite character are to be burned in Roney stokers, the furnaces should be set out considerably in front of the boiler and a reverberatory arch of considerable depth should be built up; then little smoke and good economy can be secured.

Bankhead coal is burned in a large number of plants throughout western Canada, on both stationary and shaking grates in hand-fired furnaces. In one place, a Dutch-oven type of furnace with shaking grates has proved satisfactory for almost any size of this coal. In another plant, steam blowers are used under the grates in conjunction with chimney draft and a very good economy obtained. Furnaces of the Parsons type should be very suitable for this coal. Bankhead coal is also used to a large extent in many gas producers of American, Canadian and European manufacture and has shown very favorable results.

The foregoing records a single attempt to burn a variety of anthracite coal on one type of mechanical stoker. It is possible that the type of furnace might have been altered to advantage, and that there are methods of handling the coal in these stokers which were not tried. It would seem justifiable to conclude that the Western anthracites do not adapt themselves to use in mechanical stokers, while practically all the Western bituminous and lignite coals are well suited to this type. It would also seem that a reverberatory furnace having grates equipped with a steam blower would not only provide a satisfactory means of economically burning this small anthracite coal but would also reduce the tendency to clinker.

Packing Condenser Tubes

By George H. Gibson

To pack condenser tubes so that they shall have free movement in the tube plate to provide for expansion and contraction without the possibility of leakage is not so simple a problem as it might seem, but it has been solved by the type of joint described in the following.

The tube plate of the Wheeler surface condenser is bored to two diameters, one large enough for the tube, the other ¼ inch larger and threaded to receive a screw ferrule. These ferrules, which encircle the ends of the tubes, have inwardly projecting lips which prevent the tubes from creeping out of the plate at either end. In order to secure a watertight joint between the tube plate and the tube, a certain grade of corset lacing used as a packing has been found to be the best packing.

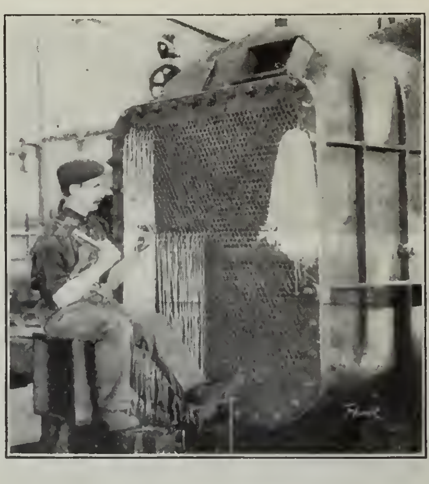


Fig. 1. Packing Condenser Tubes

This lacing requires a special treatment to render it entirely suitable. Some people use tallow for lubricating the packing, but if not of the best quality this may become rancid, breaking down into organic acids which corrode the tube. Too much lubricant causes binding while too little leads to breaking of the ferrules. The fiber of the corset lacing supplied by the makers of the above mentioned condenser, is thoroughly permeated at boiling temperature with just the right amount of paraffin, the exact amount being determined by weighing the lacing before and after treatment. A machine squeezes out any excess, so that it is difficult to tell by handling the lacing whether it has been paraffined or not. The corset lacing is then cut into the right lengths for packing the tubes.

This prepared lacing permits of rapid

work, as it gives just the right amount of properly lubricated packing for each joint.

There are several special devices for inserting the packing in condenser-tube plates, but the following simple method is most satisfactory:

The workman takes a handful of packing and standing or sitting in front of the condenser he pushes one end of a string into the annular space around the end of each tube by means of a small iron instrument, leaving the remainder of the string hanging down the face of the tube plate.

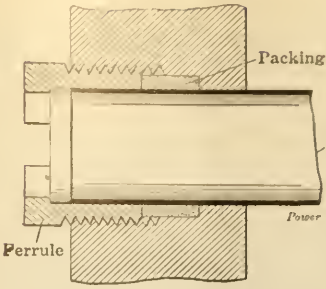


FIG. 2. DETAIL OF TUBE END

After inserting a good number of strings in this manner, he then takes a piece of brass pipe, about one foot long, having at one end a longitudinal slot sawed into it for a short distance on one side. Through this pipe he threads one of the strings of packing, the notched end of the pipe being held toward the tube plate. Then inserting the end of the pipe in the annular space around the end of the condenser tube he taps it with a wooden mallet, turning the tube through an arc of 90 or 120 degrees with his left hand between the strokes of the mallet. The method is shown in Fig. 1. In this way each layer of the lacing is packed evenly and, at the same time, not so tightly as to resist movement of the tubes. Too tight packing should be carefully avoided since it destroys the fiber of the lacing and does not secure a permanent water-tight joint. The packed tube is shown in Fig. 2. The above method takes time, but a tight joint is insured when the packing is in place.

Some Types of Torsion Meter[*]

By Edward P. Buffet

With the advent of the steam turbine in marine practice came the necessity of measuring the power delivered to the shaft. The steam-engine indicator was not applicable to this purpose and water brakes could not be used conveniently aboard ship. This led to the development of the torsion meter, based upon the principle that the arc of torsion of a shaft is proportional to the torque applied.

Dynamometers are divided into two general classes, namely, those of absorption and those of transmission. The absorption type is substituted for the resistance against which the prime mover works, and absorbs in friction or otherwise the power generated; the prony brake belongs to this class. Transmission dynamometers, on the other hand, are inserted somewhere between the source of power and the point at which it is utilized; consequently they may often be employed for taking measurements while the engine is in service. To this class belongs the special type known as torsion meters.

These meters are especially adapted to measuring the torsion in the shaft which transmits the power. While the original of this type dates back many years to the "pandynamometer" of Hirn, yet the development in general practice is comparatively recent and is the outcome of a need created by the steam turbine. With the advent of this type of engine, there arose a difficulty in accurately determining its power, as the reciprocating-engine indicator was not adapted to this use. In marine practice, efficiency measurements of the turbine and propeller often had to be lumped together, which was very unsatisfactory. This set engineers to work upon devising improved methods of applying the simple principle that a shaft when transmitting power will undergo a twisting action, which bears a definite relation to the power that is being transmitted. If we lay off a certain length upon the shaft, limited by two points in the same plane, and can devise some method of measuring the angular displacement of one point with reference to the other, while the shaft is transmitting power, this displacement, due to the twisting of the shaft, will be a function of the power transmitted. We shall not here concern ourselves with the mathematical calculations necessary to figure this out, further than to state the rule: *The amount of twist varies directly as the length between the two points, directly as the moment of the load applied, inversely as the rigidity of the material, and inversely as the fourth power of the shaft diameter.*

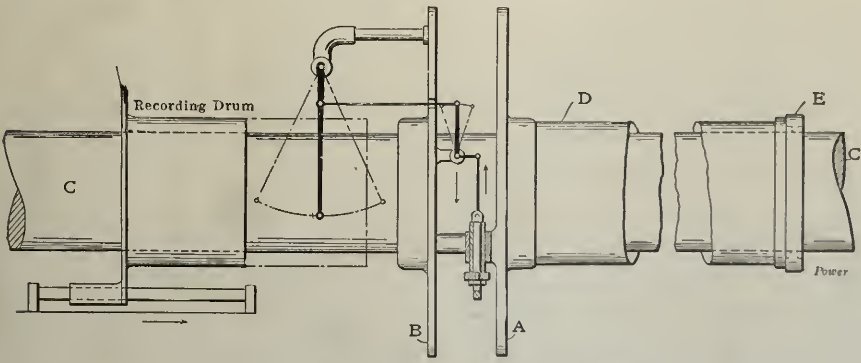


FIG. 1. FOETTINGER TORSION INDICATOR

The measurement of power by this means presents some decided advantages over the steam-engine indicator in point of reliability and the elimination of certain disturbing factors which often enter into the result. There is need, however, of care and precision in practice, since the actual amount of twisting in a shaft is usually very small; the twist of a propeller shaft, for example, rarely exceeds one degree in 10 feet of length, so that for a 12-inch shaft the circumferential displacement is only about one-eighth of an inch at full power. Yet with certain torsion meters it has been possible to determine powers to within one-fourth of one per cent., and possibly even that degree of precision has been exceeded.

A convenient division of the leading types of torsion meter can be made into three classes—mechanical, electrical and optical.

MECHANICAL TORSION METERS—THE FOETTINGER

Referring to Fig. 1, *A* and *B* are two annular disks attached to the shaft *C*; *B* directly connected to the shaft, and *A* serving as a flange of the sleeve *D* which extends to the right and is fixed to the shaft at *E*. Thus the length on the shaft availed of for torsional measurement is much greater than the distance between the two flanges themselves. A system of levers connecting these flanges amplifies their relative twist to move a pencil on an indicator drum. The illustration shows a simple form of drum consisting of a sleeve surrounding the shaft, fixed at the bottom so that it cannot revolve and arranged so that it may be slid along to the right under the pencil. As a matter of fact, a more refined sort of continuous drum is employed, this being driven by a gear from the shaft. At high speeds means are provided to compensate for the effect of centrifugal force.

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[*] Embodying material from an article in the *Revue de Mécanique* and various other sources.

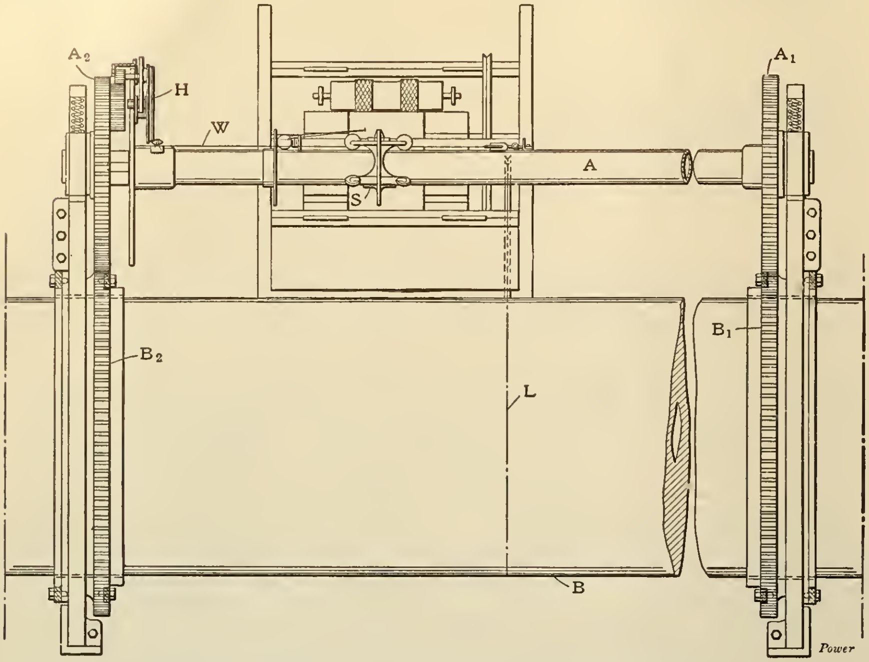


Fig. 2. Denny & Edgecombe Torsion Indicator

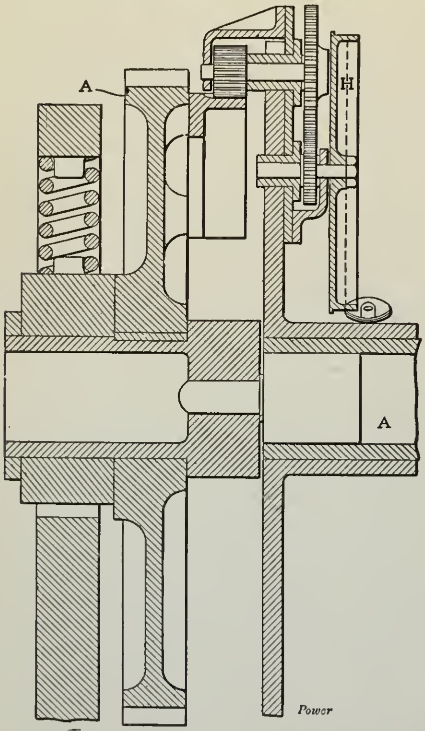


Fig. 3. Detail of Transmission in Denny & Edgecombe Meter

Investigations with this instrument have proved that shafts driven by reciprocating engines give considerable variations in torque, very important differences existing at high speeds between the diagrams produced by the torsion meter and corresponding ones plotted from steam-engine indicator cards and inertia force.

The Denny & Edgecombe

In the Denny & Edgecombe torsion meter, Figs. 2 and 3, a countershaft *A* is geared parallel to the main shaft *B* and runs at an increased speed. Equal gears B_1 and B_2, fixed upon the main shaft, mesh with smaller equal gears A_1 and A_2; the former is fixed to the countershaft and the latter is free to turn upon a bushing. It is evident that gear A_2 will revolve with shaft *A* as if keyed to it, except that, when shaft *B* is twisted, gear A_2 is angularly displaced about shaft *A*, a proportionate amount. Countershaft *A* and gear A_2 are connected by a train of gearing which, when they are angularly displaced, actuates an indicator pencil. The mechanism by which this is accomplished is shown, in Fig. 3. Let it suffice to say that over a pulley *H* passes a wire *W* (Fig. 2), which runs to a saddle *S* that rolls along shaft *A* and engages with a

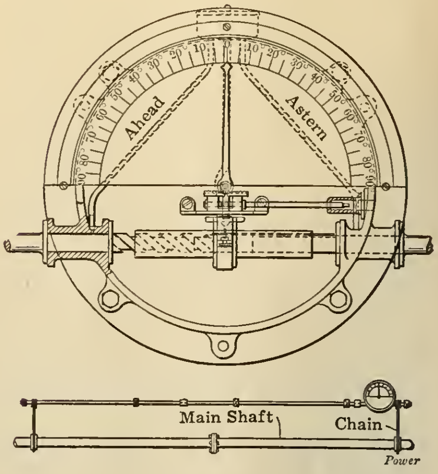


Fig. 5. The Collie Torsion Meter

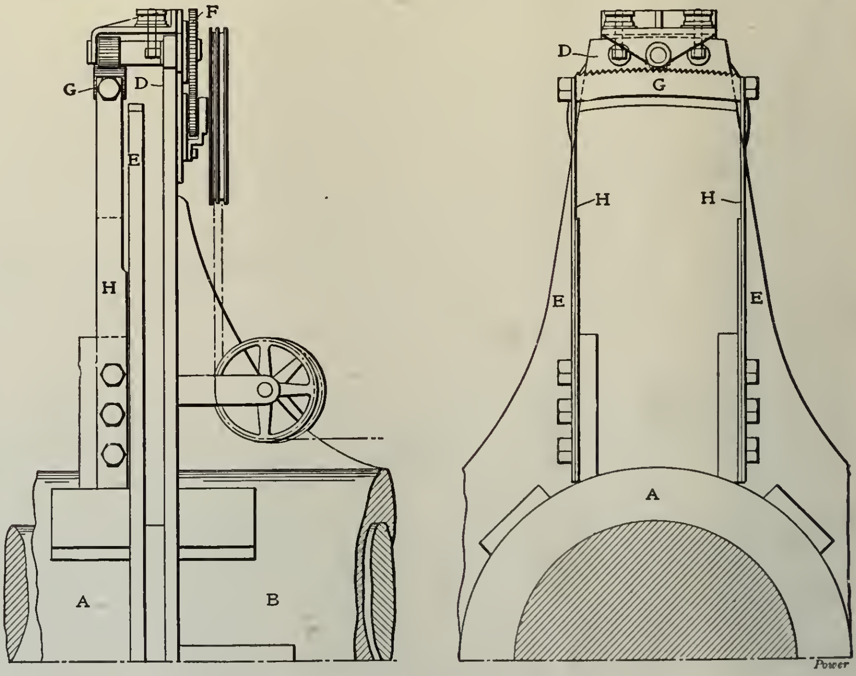


Fig. 4. Denny & Edgecombe Damping Device

pencil carriage. Behind this saddle is fixed the continuous record sheet which is fed from the main shaft by a belt shown by the broken line *L*. There is a second pencil to trace the base line of the diagram.

Evidently the saddle and entire mechanism controlling it rotate constantly with countershaft *A*, but the pencil and drum mechanism are fixed. The edge of the saddle disk *S*, which does revolve, engages with a jaw on the pencil carriage which also does not revolve. Another feature of this meter is the spring devices for absorbing the backlash of the gears.

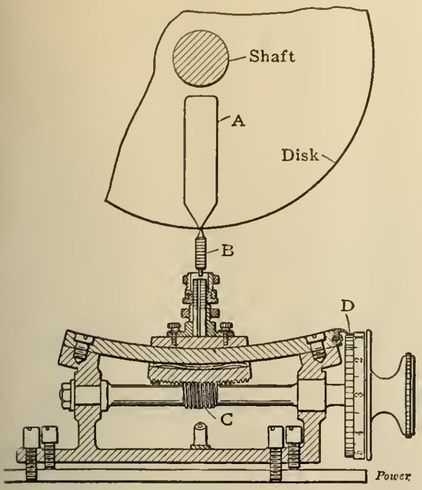


FIG. 6. DENNY & JOHNSON TORSION METER

Fig. 4 shows a damping device brought out in 1908 by Denny & Edgecombe, by which the effect of violent fluctuations of torque that might endanger the apparatus is taken up by springs. It is applicable to the countershaft type of torsion meter already described, but as here shown is adapted to a pattern by which is measured the relative movement between adjacent ends of sleeves attached to the shaft some distance apart. The arc-shaped gear *G* actuates the multiplying gear *F*, mounted upon arm *D* at the inner end of the sleeve *B*, and, instead of being rigidly mounted, is carried by two springs *H* fixed to brackets upon arm *E* at the adjacent end of the other sleeve *A*. Under abnormal stress the springs *H* yield. When the torsion meter is used in connection with shafts not subjected to a fluctuating torque, such as those of turbines and electric motors, the gear segment may be rigidly connected to the arm *E* by means of a block fitting between the springs *H*, and secured to the arm.

THE COLLIE

In the Collie type, Fig. 5, the rotation of the shaft is transmitted by chain belts to a smaller shaft consisting of two parts, one of which screws into the other. According to the amount of torsion of the main shaft between the two pulleys, the two portions of the small shaft are screwed together, which relative position controls the movement of a needle on a dial.

ELECTRICAL TORSION METERS—DENNY & JOHNSON

The first electrical torsion meter devised by Mr. Denny was a simple affair and consisted of two insulating disks placed at some distance apart on the shaft; each of these carried a point touching a brush and, when unloaded, thus completed an electric circuit, at a certain point in the shaft revolution. The circuit passed through a telephone receiver which gave a click, but no current was produced unless both brushes touched simultaneously. When the shaft was loaded its torsion threw the points out, so that they did not touch at the same time. Then one of the stationary brushes had to be moved through an arc about the shaft a sufficient amount to restore concordance, which amount was measured on the shaft by a micrometer screw. This gave the angle of twist from which the power could be calculated.

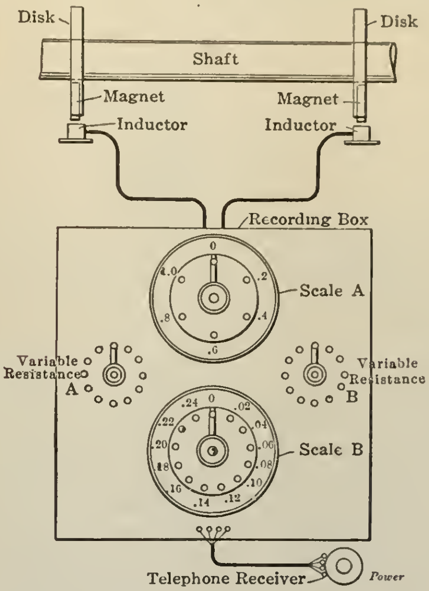


FIG. 7. DENNY & JOHNSON'S IMPROVED TORSION METER

This apparatus worked fairly well for slowly revolving shafts, but for high speeds a better form of contact became necessary. This problem was solved by Mr. Johnson and is shown in Fig. 6. The points on the shaft were replaced by chisel-shaped permanent magnets *A* and the stationary brushes by corresponding chisel-shaped electromagnets *B*. The electromagnets were connected with a differentially wound telephone receiver and at the instant the permanent magnets passed over them, currents were generated in the electromagnets and sent through the telephone. If these currents were simultaneous, being equal and opposite, the telephone gave no sound. If, by reason of the twisting of the shaft, they were produced at different times, they became audible in the receiver. The position of one of the electromagnets could then be slightly changed by the gear *C*, until the telephone recorded no sound. The amount of shifting denoted the degree of torsion and was measured by the micrometer *D*.

While this arrangement worked very well, it required the operator to be in the noisy tunnel of the ship. To overcome this, Mr. Johnson designed another arrangement shown in Fig. 7. The permanent magnets were retained but the chisel-shaped electromagnets were replaced by two soft-iron sector cores, or inductors; one contains six insulated coils spaced 0.2 inch apart, and the other has thirteen insulated coils spaced 0.02 inch apart. Below these is the recording box, which contains two disks fitted with contact studs and movable contact arms. The upper one, marked scale *A*, is connected with the inductor nearest the propeller and has six contacts to correspond with the six coils on the inductor, that are spaced 0.2 inch apart. The other one, marked scale *B*, has 13 contact studs, connected to the 13 coils spaced 0.02 inch apart on the other inductor, these contact studs being devices for switching any desired coil into the telephone circuit. On the recording box there are also two side coils, marked respectively resistance *A* and resistance *B*. These are for throwing resistance into series with the differential windings of the telephone receiver, which is connected to the two inductor circuits, and their circuits must be accurately balanced before absolute silence can be obtained in the receiver.

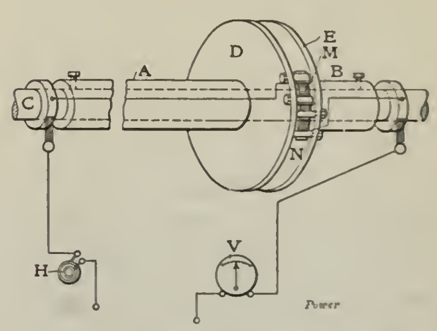


FIG. 8. NEW TYPE OF DENNY & JOHNSON METER

To take a reading after the instrument has been set, and the resistance of the two telephone circuits has been adjusted by means of the variable resistances, it is necessary only to turn the arm on scale *B* around to the various studs until there is silence in the telephone, when the amount of torsion is immediately read on the scale. If no such position be found, it means that the shaft is being twisted more than is covered by the scale; the arm on scale *A* is then turned to the first contact, and the arm on *B* is again swept round the circuits. If silence be still not obtained, the arm on *A* is turned to the second contact, and so on, the combined range of the scales being 1.24 inches, which is more than sufficient to measure the maximum twist usually obtained. The object of this is to throw in those coils that will be passing under the centers of the two permanent magnets simultaneously. The change from one to another coil of the coarsely wound electromagnet serves for coarse adjustments while the subsequent fine adjustments are made with the finely wound magnet. From torsional experiments on the shaft, made previous to the meter being fitted in the ship, or from a predetermined formula, the factor by which this reading is to be multiplied is obtained, and the power is easily calculated.

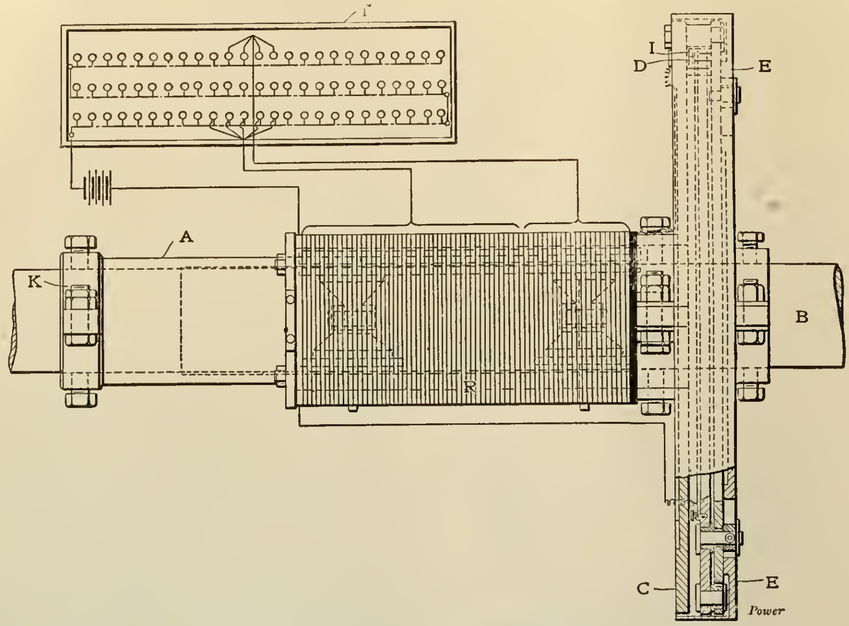


Fig. 9. Rangdale Torsion Meter

A patent has been granted this year for another Denny & Johnson electrical torsion meter, quite different from the foregoing. Fig. 8 shows two sleeves *A* and *B*, attached at their outer ends to the shaft *C*, the torsion in which is to be measured. At the near ends of these sleeves are disks *D* and *E*, upon which are electromagnets *M* and *N*, with their pole faces adjacent. One end of the winding of the magnet *M* is connected with an alternator *H*, the opposite pole of which is grounded. The other end of the winding of the magnet *M* is grounded to a stud on disk *E*. One end of the winding of the magnet *N* is connected with a voltmeter *V*, the other terminal of which is grounded. The other end of the winding of the magnet *N* is grounded to the stud on the disk *E*. The alternator *H* is driven from the shaft *C* and supplies current to the electromagnet *M*. This current gives rise to a magnetic field, which in turn induces a current in the windings of electromagnet *N*. The current supplied by the alternator varies with the speed of the shaft, and the induced current in the windings of *N* depends upon the strength of this current and the air gap between the electromagnets *M* and *N*. This air gap varies with the torque to which the shaft is subjected. By suitably calibrating the voltmeter *V* the torque or horsepower can be read directly.

The Rangdale

A torsion meter whose indications appear through a series of electric lamps has been patented by George Stuart Rangdale and the Palmers Shipbuilding and Iron Company of England.

Referring to Fig. 9, sleeve *A* is fixed on the shaft *B* at *K*, and at the other end is carried a flange *C*, turning opposite to a wheel *E* which is fixed upon the shaft. Between the flange and the wheel is a set of levers by which is obtained a movement multiplying their relative angular displacements due to the twisting of the shaft.

This amplified motion is employed to move the brush *I* over an arc-shaped commutator *D*, placed between the wheel and the flange. The strips of this commutator are connected with a series of insulated collector rings *R* situated around the sleeve; and each of these is provided with a brush which places it in circuit with a lamp on the table *T* The current is furnished by a battery. The contacts in commutator *D* are arranged in order with the lamps, so that the latter are lighted in regular succession according to the angle through which the shaft is twisted. The arrangement is double-acting, indicating movements of the vessel both ahead and astern. The commutator brush is sufficiently broad so that each lamp is extinguished only after the next one has been lighted.

The Gardner

In the Gardner device there are mounted upon the shaft two similar collectors, with brushes connecting them to the circuit of a galvanometer. When the shaft is not under torque, the position of the brush on the segments of one collector corresponds to that of the brush on the insulated sections of the other collector and no current passes. When the shaft is twisted, a circuit is completed, and increases directly with the torque.

Optical Torsion Meters—The Bevis-Gibson

Reference has already been made to the use of a ray of light for indicating purposes in torsion meters. A recent application of this principle is found in the Bevis-Gibson flash-light torsion meter, a device capable of high precision. At a distance apart on the shaft are mounted two disks, each pierced near its periphery by a small radial slot, both slots being in the same plane when no torque is exerted on the shaft. Behind one slot is placed a light and in front of the other an eye piece, called a "torque finder." When the shaft is transmitting no power, the two slots, the light and the eye piece are all in line and, at a certain point of each revolution, the observer perceives through the eye piece a flash of light. If the shaft is turning at a speed of 100 revolutions per minute or more, this flash will appear to be continuous. If, however, a load be applied, the torsion of the shaft will throw the slots out of line so that there is complete darkness all the time. If, now, we move the eye piece a little to one side, we can bring it again in line with the slots and lamp, looking obliquely, and can restore the flash.

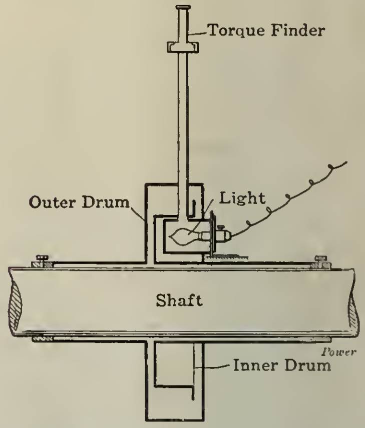


Fig. 10. Type of Meter Used When Only Short Length of Shaft is Available

The angle through which the eye piece is shifted is measured by a micrometer and the torsion determined accordingly. One advantage of this type is that the disks can be fitted a greater distance apart and closer readings thus obtained.

For reciprocating engines a simple modification of the flash-light torsion meter enables the operator to take several readings in one revolution of the shaft. The disks are perforated with 12 slots arranged in the form of a spiral—and spaced 30 degrees apart. The lamp and the torque finder are moved radially from the shaft so as to bring them into line with each corresponding pair of slots in the disk.

Cases sometimes occur, especially in modern warships, where a long length of shafting is not available for torsion-meter purposes, and recourse may then be had to a special form of apparatus. This is shown in Fig. 10. The perforated disks are here replaced by concentric drums around the shaft, which are slotted in their circumferences and inside the smaller of which the lamp is placed. The beam of light therefore is directed toward the torque finder in an approximately radial direction instead of one nearly parallel to the axis. It is evident that the points on the shaft to which the flanges of these respective drums are attached limit the length whose twist is measured.

The extreme sensitiveness of the apparatus is almost incredible. The angle of the flash proceeding radially from the shaft can be measured to 0.001 of a degree, so that although only 3 feet of shafting may be available, the result is as good as if a 30-foot length had been used with an axial-ray apparatus.

THE HOPKINSON-THRING METER

At a recent meeting of the British Institute of Naval Architects, Professor Hopkinson described the Hopkinson-Thring type of torsion meter as manufactured by Siemens Brothers. In principle it depends upon the reflection of light from a pivoted mirror somewhat similar to the manner of taking galvanometer readings. Referring to Fig. 11, a sleeve is clamped on the shaft in the plane *A A*, and under it a collar is clamped in the plane *B B*. The operation depends upon the mutual rotary displacements of the sleeve and collar where they overlap. By an appropriate leverage system, shown in detail at *E*, a mirror pivoted in a plane of the shaft axis is caused to rotate out of this plane, about an axis radial to the shaft, when the twisting of the shaft causes a change in the relative positions of the sleeve and collar. Assuming the shaft to be running under no load and a lamp to be placed at *J* behind a scale in a line perpendicular to the center of the mirror, its ray will be reflected back to the same point for an instant each time the shaft turns around. When a load is applied, the twisting of the mirror throws the ray to a point *K*. at one side of *J* on the scale, its angular reflection bearing a definite relation to the torsion of the shaft. In practice, usually two or more mirrors are employed, they being placed on opposite sides of the sleeve. The rays of light reflected on the scale are, of course, mere flashes, but at high speeds they appear continuous. In certain ways it is possible to distinguish the reflections of the different mirrors. As their readings may be expected to disagree slightly, a mean of them is taken.

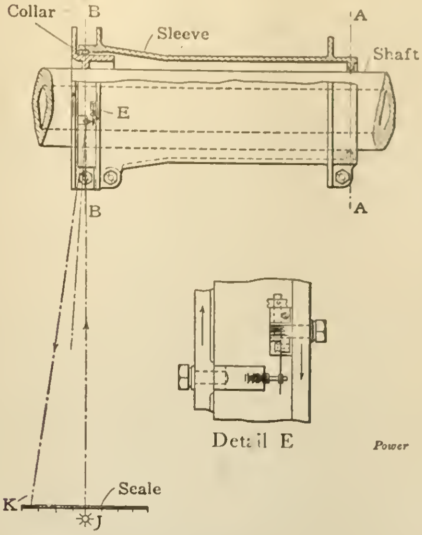


FIG. 11. HOPKINSON-THRING TYPE OF TORSION METER

Increasing the Power of an Engine

The work of an engine is expressed by the formula

$$P \times L \times A \times N = Work$$

in which

$P =$ Unbalanced pressure per square inch on the piston.

$L =$ Length of the stroke of the piston in feet.

$A =$ Area of the piston.

$N =$ Number of strokes per minute.

In other words, the work performed in a minute is the product of the pressure acting and the space through which it acts in that time. The product of *P* and *A* is the pressure and the product of *L* and *N* is the space.

If numerical values be given to the letters in the left-hand member of the equation, a definite number will appear at the right.

If any of the factors be increased in value the work will be increased in the same proportion and a decrease in work will result from a decrease in the value of any one or all of these factors.

What has just been said was prompted by a correspondent who says that this question was put to him during a recent examination for an engineer's license:

Suppose you were running an engine at full capacity, the limit of flywheel speed had been reached, you did not wish to compound, did not wish to bore out the cylinder, could not put on two eccentrics, could not raise the steam pressure, could not make the cutoff later nor run condensing, what would you do in order to carry more load?

If, as stated in the question, none of the factors that go to make up the power of the engine can be increased, no more load can be carried.

But there is an implication in the question that some change may be made that will enable the engine to carry a heavier load and suggestions may be sought for in the question itself.

First, limit for flywheel speed is reached.

The speed of the engine may be safely increased by putting a smaller flywheel on the shaft, increasing the factor *N* in the equation and thus increasing the power. Compounding, boring to a larger diameter and an extra eccentric being forbidden, they are passed and the question of steam pressure taken up.

Steam pressure at the boiler may not be increased, but the drop in pressure between the engine and the cylinder may possibly be reduced by suitable changes in size and arrangement of piping, and the installation of a steam drum or reservoir close to the cylinder. This, though not increasing the steam pressure at the boiler, may permit a higher pressure to be realized in the cylinder.

It may be found that there is more inside lap of valve than is necessary and that a reduction will produce an earlier opening and a later closure of the exhaust port, thus reducing the back pressure on the piston.

Doing either of these things would be tantamount to an increase of the factor *P* in the equation and a consequent increase in power would follow.

And, lastly, "cannot run condensing."

Condensers are installed with steam engines for the sole purpose of reducing the back pressure. In the case stated a condenser is tabooed, but an examination of the exhaust passage and piping may show that the back pressure on the piston may be reduced, which is as beneficial as a raise in the initial pressure, and produces results equivalent to an increase in the factor *P* in the equation.

It would appear from an examination of the question that with an ordinary case where especial attention had not been given steam- and exhaust-pipe sizes, and to valve design and setting, the factors *P* and *N* in the horsepower formula might both be increased with consequent increase in power and still be within the restrictions stated in the question.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Primer of Electricity

Resistance Loss in a Dynamo Armature

It was explained in one of the early Primer lessons that passing a current through any conductor requires power to be spent, and the loss of power is manifested by heating of the conductor. Thus, if 10 amperes are passed through a conductor which has a resistance of 25 ohms, it requires a pressure of

$$10 \times 25 = 250$$

volts to force the current through; this is called "drop," meaning loss of pressure, and the power spent in forcing the current through is equal to the "drop" multiplied by the current, thus:

Drop in volts × *Amperes* = *Watts*

lost in overcoming the resistance. Since the drop is equal to

Resistance × *Current,*

the power lost by resistance is equal to

Ohms × *Amperes* × *Amperes;*

or

Ohms × *Amperes*^{2} = *Watts* lost.

From the foregoing it follows that when current passes through the armature winding of a dynamo or motor, there results a loss of power proportional to the resistance of the winding and the square of the current passing through it. The armature windings of all direct-current machines have at least two parallel paths through them, as explained in the Primer lesson of January 25, and many windings have more than two. For this reason, the resistance of the winding as a whole is much less than it would be if the wire in the winding were arranged in a single path; that is, all in series. This is a question of the joint resistance of parallel circuits, which was explained in detail in the lesson of October 19, last year

Suppose, for example, that an armature winding contained 80 coils of No. 13 wire, each coil containing 10 turns and each turn being 15 inches long. Each coil would contain

$$150 \div 12 = 12\tfrac{1}{2}$$

feet of wire and the whole winding would contain

$$12\tfrac{1}{2} \times 80 = 1000$$

feet. The resistance of the wire, if it were all in one piece, would be about 2.4 ohms when warmed up. But if the winding were divided into two parallel paths, each path would contain one-half of the wire and the resistance of each half would therefore be 1.2 ohms; the two halves in parallel would have a resistance of

$$1.2 \div 2 = 0.6 \text{ ohm.}$$

Now, if the full-load current of that armature were 30 amperes, the drop in it would be

$$0.6 \times 30 = 18 \text{ volts,}$$

and the loss of power due to the resistance of the winding would be

$$18 \times 30 = 540 \text{ watts.}$$

Ignoring the drop and figuring the loss from the resistance and current directly:

$$0.6 \times 30^2 = 0.6 \times 900 = 540 \text{ watts.}$$

This can be proved by considering the paths in the winding separately. If the total armature current were 30 amperes and there were two parallel paths, each path would carry 15 amperes. As just shown, the resistance of each path would be 1.2 ohms; consequently the watts lost in each path would be

$$1.2 \times 15^2 = 1.2 \times 225 = 270$$

watts, and the loss in the two paths would be twice this, or 540 watts, as obtained by the first calculation.

Now suppose this same armature winding were arranged to have four parallel paths. The total resistance of the wire being 2.4 ohms, the resistance of each path would be one-fourth of that, or 0.6 ohm, and the joint resistance of the four paths in parallel would be one-fourth of the resistance of each one separately, or 0.15 ohm. The watts lost in this winding would be equal to 0.15 × the square of the total current.

From what has been said it will be clear that

The resistance of any armature winding is equal to the total series resistance of all the wire in it divided by the square of the number of parallel paths through the winding.

Thus, with two paths, the total series resistance is divided by four; with four paths, by sixteen; with six paths, by thirty-six, and so on. (An armature winding having more than one path through it cannot have an odd number of paths.)

Table 5 is very useful for estimating the resistance of armature windings with-

TABLE 5. RESISTANCE OF ARMATURE WINDINGS WHEN WARM.

Effective resistance of the winding per foot of wire in it.

The resistances given in the upper half of the table are 1,000,000 times as great as the actual resistances, in order to avoid the use of a large number of decimal places. Consequently, final results obtained from this part of the table must be divided by 1,000,000.

Wire Gage Sizes.	2 Paths.	4 Paths.	6 Paths.	8 Paths.	10 Paths.	12 Paths.	16 Paths.
4	74	18.5	8.2	4.62	3.00	2.05	1.15
5	93	23.4	10.4	5.84	3.74	2.60	1.45
6	117	29.4	13.0	7.35	4.70	3.25	1.85
7	148	37.0	16.5	9.28	5.94	4.15	2.35
8	187	46.8	20.8	11.7	7.50	5.20	2.93
9	236	59.0	26.2	14.7	9.5	6.55	3.68
10	298	74.4	33.1	18.6	11.9	8.25	4.65
11	375	93.8	41.7	23.4	15.0	10.4	5.85
12	473	118	52.6	29.5	18.9	13.1	7.4
13	597	149	66.3	37.3	23.9	16.5	9.3
14	752	188	83.6	47.0	30.1	20.9	11.7
15	949	237	105	59.3	37.9	26.3	14.8
16	1,190	299	133	74.7	47.8	33.2	18.7
17	1,500	377	167	94.3	60.3	41.9	23.5
18	1,900	475	211	119	76	52.8	29.7
The resistances in the lower half are 10,000 times the actual resistances.							
19	24.0	6.0	2.66	1.50			
20	30.2	7.56	3.36	1.89			
21	38.1	9.53	4.23	2.38			
22	48.1	12.0	5.35	3.01			
23	60.6	15.0	6.74	3.79			
24	76.5	19.1	8.5	4.78			
25	96.4	24.1	10.7	6.02			
26	121	30.4	13.5	7.60			
27	153	38.3	17.0	9.58			
28	193	48.3	21.4	12.08			
29	243	60.9	27.1	15.2			
30	307	76.8	34.1	19.2			
31	387	96.9	43.0	24.2			
32	488	122	54.3	30.5			
33	616	154	68.5	38.5			
34	777	194	86.4	48.6			
35	980	245	108.5	61.2			
36	1,236	309	137	77.2			

out figuring the total resistance of the wire and then reducing it to the effective resistance for the number of paths through the winding. In order to avoid using a large number of decimal places, the resistance values in the upper half of the table have been made 1,000,000 times the actual effective resistance of the winding per foot of wire in it, and the values in the lower half are 10,000 times the actual effective resistances. To use the table, it is necessary only to multiply the total number of feet of wire by the number in that column of the table under the number of paths in the winding; then divide by 1,000,000 if the number is in the upper half or by 10,000 if it is in the lower half of the table. For example, suppose a winding contains 850 feet of No. 10 wire and has eight paths through it. Starting at the number 10 in the column of wire sizes, follow that line into the table until the 8-path column is reached; here will be found the number 18.6. Multiplying this by the number of feet gives

$$850 \times 18.6 = 15{,}810,$$

and dividing this by 1,000,000 gives 0.01581, or practically 0.016 ohm, as the resistance of the winding.

Again, suppose a four-path armature contains 600 feet of No. 16 wire. In the upper half of the table, in line with No. 16 and in the 4-path column is the number 299. Multiplying this by 600 gives 179,400, and dividing this by 1,000,000 gives 0.1794 or practically 0.18 ohm, as the resistance of the winding.

Once more, if a bipolar winding (two paths) contains 250 feet of No. 23 wire, what will be the resistance when warm? In the No. 23 line and the 2-path column is the number 60.6, and as this is in the lower section of the table it must be divided by 10,000 giving 0.00606 for the constant. Multiplying this by 250 gives 1.515 ohms as the resistance of the winding.

As the values in the table are based on the hot resistance of the wire, and the temperature of the actual winding may be appreciably different from that for which these figures are calculated, it is not worth while to split hairs in the final results; three figures are sufficient, and if the third figure is as small as 2, it may as well be dropped. In the last example therefore, it would be amply accurate to say that the resistance would be substantially 1.5 ohms.

It must not be forgotten that the result of each calculation is the resistance of the armature winding as a whole, that is, the resistance that would be obtained if it were measured between the positive and negative brushes of the armature and the resistance of the brushes themselves deducted. The loss in the winding is ascertained in watts by multiplying the resistance by the square of the total current, as in the foregoing examples. Thus, if the total current of the eight-path armature wound with No. 10 wire were 150 amperes, the loss in the winding would be

$$150^2 \times 0.016 = 360$$

watts. If the current in the four-path winding of No. 16 wire were 20 amperes, the loss in the winding would be

$$20^2 \times 0.18 = 72$$

watts. And if the total current of the two-path armature wound with No. 23 wire were 2.5 amperes, the loss would be

$$2.5^2 \times 1.5 = 9.375,$$

or practically 9.4 watts.

The Electric Elevator

By Wm. Baxter, Jr.

THE ELEKTRON SYSTEM

(*Concluded*)

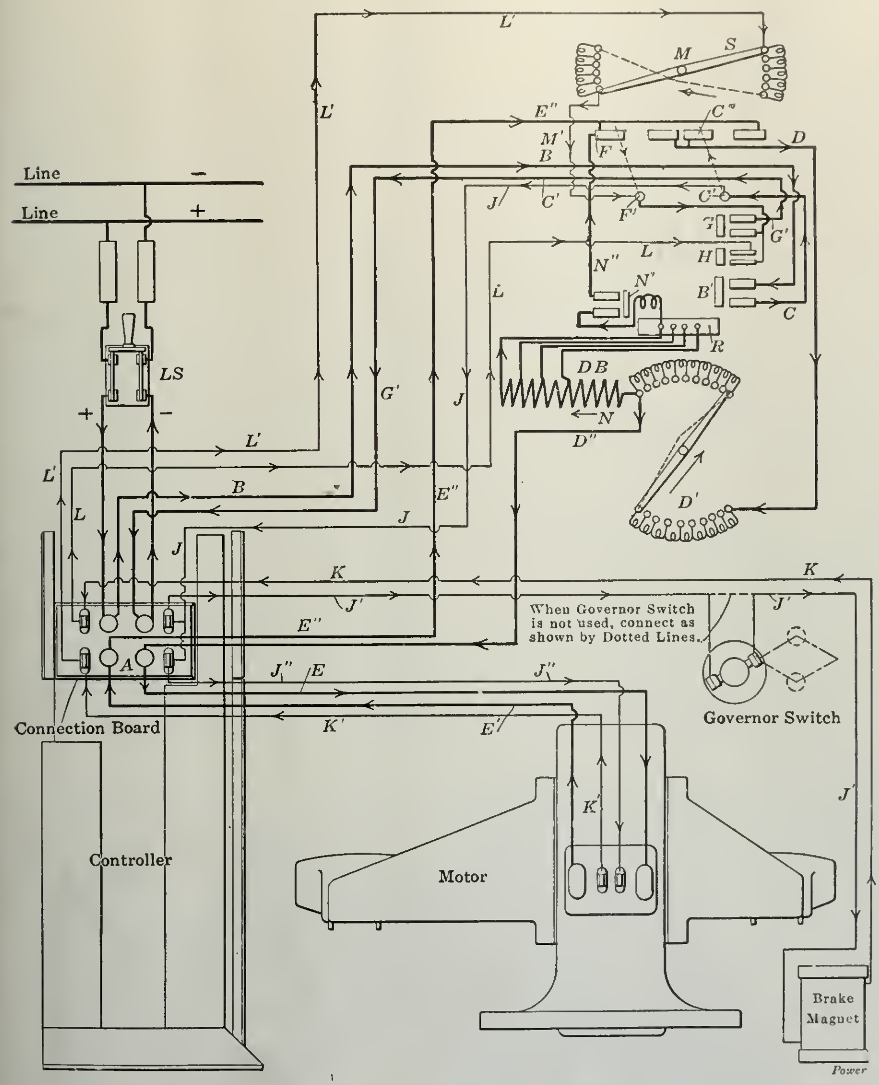


FIG. 123. COMPLETE WIRING DIAGRAM OF ELEKTRON SYSTEM

The complete wiring diagram of the Elektron controller is shown in Fig. 123. This diagram is very complete; it shows not only all the wire connections between the motor, the controller and the line, but also the connections with the connection-board binding posts on the controller box. Such a diagram serves admirably to enable one to trace out the internal connections of the controller starting from the binding posts and running to the several internal points, but the introduction of the controller connection board complicates the appearance of the diagram; for this reason the simplified diagram, Fig. 124, is presented, by the aid of which Fig. 123 can readily be analyzed. The only difference between these two diagrams is that in the latter the connections between the controller and the motor are shown direct instead of passing through the connection board on the side of the controller box. The direction of the current through each of the wires of the system is indicated by an arrow head, so that an extended explanation will not be necessary; by starting from the + line and following the arrow heads, the path of the current can be traced. The parts marked *C''* and *F* are the stationary contacts of the reversing switch which was explained in connection with Fig. 117, and *R S* and *R S'*

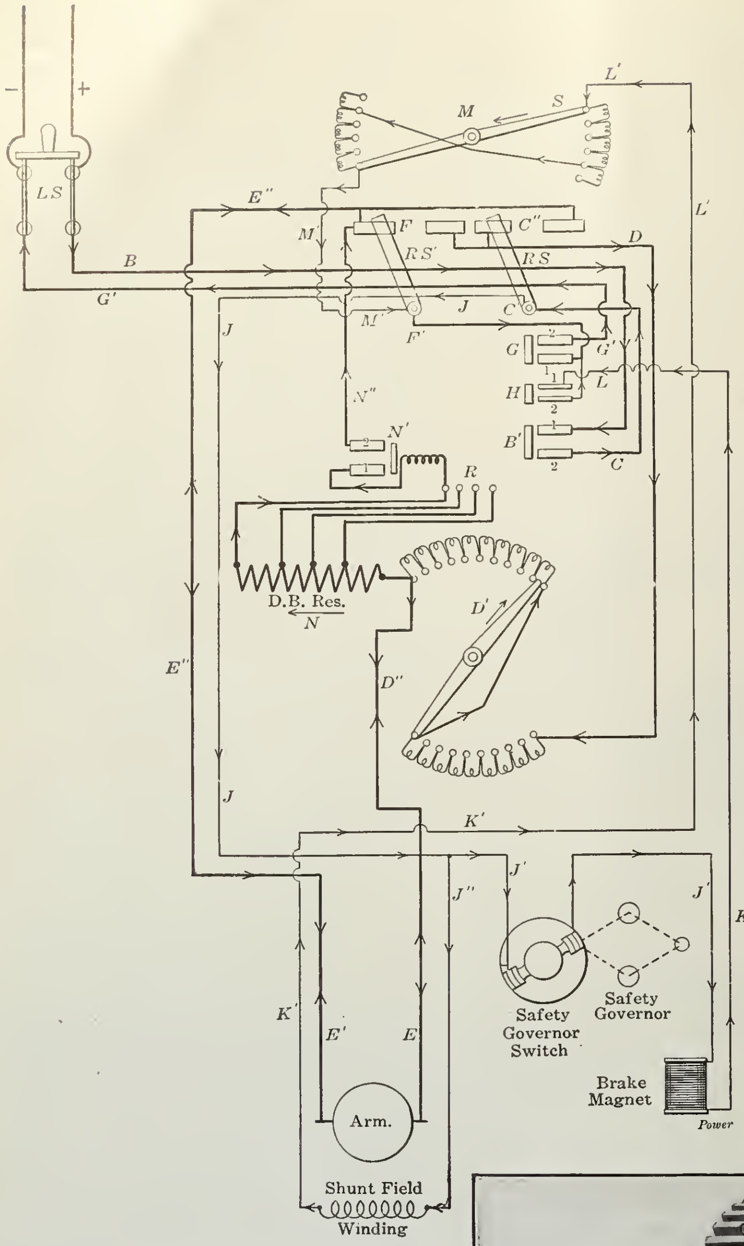


Fig. 124. Simplified Diagram

are the switch levers *A A'*. The rectangles *G*, *H* and *B'* indicate the disks of the starting switch shown in Figs. 118 and 119 and marked "Line Disks" and *J''*. The latter is marked *H* in the winding diagram, and acts to make and break the circuit of the brake magnet. The switch indicated at *N'* in the winding diagram is the dynamic brake switch shown in Figs. 118 and 119 at *G*, *G'* and *G''*.

The safety governor shown in the diagrams is a device that is commonly used with electric elevators; its object is to stop the motor if for any reason the elevator car attains a dangerously high velocity. The way in which this governor acts is indicated in the diagram. If the speed goes beyond the point for which the apparatus is adjusted, the switch controlled by the governor is opened, and in this system it acts to open the circuit of the brake magnet so that the brake may be applied to stop the motor. In some systems the governor actuates not only a switch to open the brake magnet circuit, but also another switch to open the main line connections. The arrangement shown in the diagram will accomplish the same result although in a round-about way. If the brake is applied the motor speed will be reduced and as a result the current passing through it will be increased and will melt the safety fuse, or open the circuit breaker if one is provided, and thus the line will be opened.

The Elektron elevator machine shown in Fig. 114 is a regular type of machine used for freight or passenger service, but sometimes special machines are required for special service, and then the design is more or less modified. Fig. 125 shows a heavy-duty machine that is so arranged that it can be made to lift heavy loads at a slow speed, or lighter ones at a correspondingly higher speed. To lift heavy loads the gears *B* and *C* are thrown into mesh, and gear *B'* is moved along the shaft *S* until it is out of gear with *C'*. To run at the high speed, lifting light loads, the gear *B* is drawn out of mesh and *B'* is thrown in. This machine is arranged for mechanical control, the operating sheave being shown at *F*, back of the stop motion, which is located within the casing *G*. The worm gear that is driven by the screw on the motor shaft is within the casing *A* and is mounted on the shaft *S*, so that the drum is driven indirectly through one or the other of the gears *B*, *B'*, both being made to slide along the shaft, a feather being provided to make them revolve with the shaft.

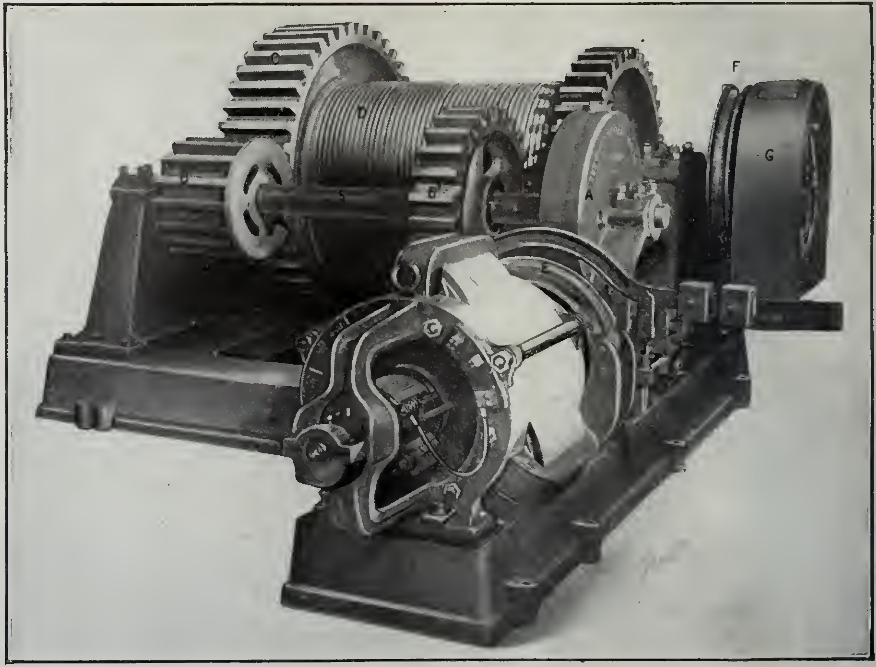


Fig. 125. Elektron Back-geared Elevator Machine

LETTERS

Kerosene Oil for Chattering Carbon Brushes

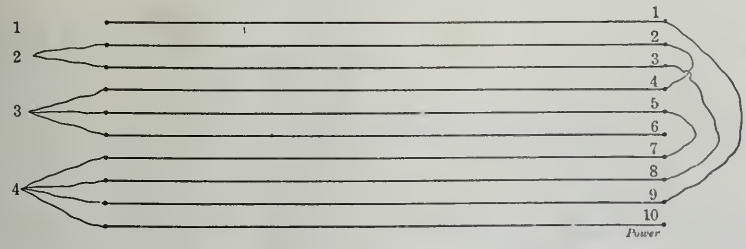
About eight or nine months ago the commutator on a multipolar direct-current machine ran dirty, with a great deal of sparking, chattering and singing of the brushes. After trying the usual remedies in regard to the setting of the brushes, sandpapering the commutator, etc., without much improvement, I was advised to soak the brushes in kerosene oil, but first to try squirting kerosene on them while running. This latter experiment had a good effect and did not cause short-circuiting, as I was afraid it would. The next day I soaked two sets of the brushes for one-half an hour in kerosene, with decided benefit; the day after, two more sets were soaked, and the following day the remaining two sets. As a result, the singing and chattering disappeared, there was a minimum of sparking, and a fine brown gloss appeared. I soaked the brushes again, about two months later, and ever since then the commutator has run smoothly, there is no noise and no sparking. Of course, after soaking the brushes in oil they should be wiped dry before using them.

Brooklyn, N. Y. H. A. HUGHES.

Identifying the Conductors in a Cable

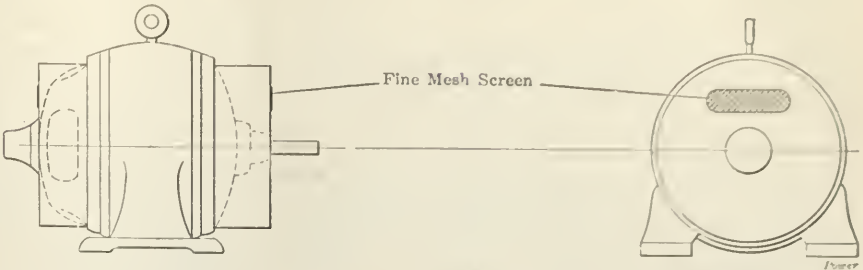
Mr. Ryan's article on locating faults in circuits, in the issue of May 24, suggested to me that perhaps a method for identifying conductors in a cable already laid, without assistance and with the least possible amount of walking, might be of service to some of the readers of POWER.

Take for example, a 10-conductor cable one-half a mile long. To identify all of the conductors by means of the galvanometer and battery by grounding one end of the conductor would call for a lot of walking back and forth. An easier method would be as follows: At the station end, connect the conductors in groups, each group containing one more wire than the preceding group. Now go to the distant end and with the galvanometer and battery identify the wires in the various groups.



IDENTIFYING THE CONDUCTORS IN A CABLE

Make a diagram similar to the one here shown. Number the conductors from one to ten and proceed somewhat as follows: Connect No. 1 to No. 9 in group No. 4; No. 2 to No. 4 in group 3; No. 3 to No. 8 in group 4, etc., taking care not to connect more than one conductor of any one group to conductors of any other one group. Indicate on the diagram the different connections and return to the station end where by means of the sketch and the galvanometer, after disconnecting the ends of the various groups, the connections as made at the distant end can be traced out and the station end numbered to correspond with the far end. For example, a wire in group 2 is connected to a wire in group 3. These can be identified and by reference to the sketch the wire in group 2 is seen to be No. 2 and the wire in group 3 is No. 4, and so on.



AN EXPEDIENT FOR PROTECTING MOTORS WITH OPEN FRAMES

This method can be used on a cable with any number of conductors and dispenses with the services of an assistant and the necessary means of communication between ends. I would suggest the use of a portable voltmeter, if one is available, instead of the galvanometer.

A. C. KERR.

Ft. Monroe, Va.

[This letter was printed in the July 5 number, but the diagram was accidentally omitted. Hence the repetition of the letter.—EDITOR.]

Motors in Dusty Places

Where open-type motors only are available for use in dusty places, the writer has got around the difficulty by making two galvanized-iron caps or shields fit the ends of the motor frame, each shield having an oval or oblong hole cut in it for ventilation; the hole was covered with a brass screen or bolt cloth of rather fine mesh so as to exclude the dust. The accompanying sketch illustrates this arrangement. The shield on the pulley end has to be made in two parts, unless the pulley can be taken off without too much trouble and the cap slipped on over the shaft.

On some types of motors, some difficulty is experienced in fastening this dust shield securely, owing to construction of the frame. On most Wagner single-phase machines, however, the shields can be held on by lugs under the yoke-bolt nuts.

I have used this method very effectively on wood-mill motors, as well as on direct-current machines operating machine tools where there was a liability of metal chips or borings and oil getting into the motor and giving trouble.

A. P. H. SAUL.

Buffalo, N. Y.

[The foregoing formed part of Mr. Saul's letter on page 1206 of the July 5 number, but the illustration was unfortunately omitted. Hence the repetition of this part of the letter.—EDITOR.]

Cheating Electric Meters

The article on "Cheating Electric Meters," by Mr. Howard, in a recent issue of POWER, was of special interest to me as I used to be a meter inspector. While so employed, I came across all the "devices" mentioned, as well as several others.

One of the most common schemes is to put a "jumper" across the current coils of the meter, so as to shunt the current around the meter. This is usually done with a short piece of wire and is easily detected, as the person doing the trick will sooner or later forget to remove the wire. Another method, which can only be worked on commutator meters of the Thomson direct-current type, is to stick a hat pin between the dial glass and the case and puncture the armature winding. I caught on to this trick by finding a piece of a hat pin stuck in a meter armature.

My experience has been that very few people try to beat electric meters; and those we have detected at it have been almost all men connected with electrical work of some kind.

G. ANDERSON.

Park River, N. D.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

The Four Stroke Cycle*

A cycle, in engineering, is any operation or sequence of operations that leaves conditions the same at the end that they were at the beginning. To illustrate this definition, consider the case of the water evaporated in a boiler into steam, passed through the engine to do work, discharged from the engine to a condenser, and finally discharged into the stream from which it was originally taken. When the water reaches the stream it has been through a *cycle* of transformations consisting of elevation by a pump, heating to the boiling point, evaporation into steam, expansion behind the engine piston, condensation back to fluid state in the condenser, and final cooling back to its original temperature in the stream. The cycle is not complete until the last event occurs; then the condition of that water is exactly the same as before the pump lifted it from the stream. When a surface condenser is used and the condensed water is returned to the boiler through the heating system, the illustration is more exact because then the same particles of water continue to circulate through the whole system.

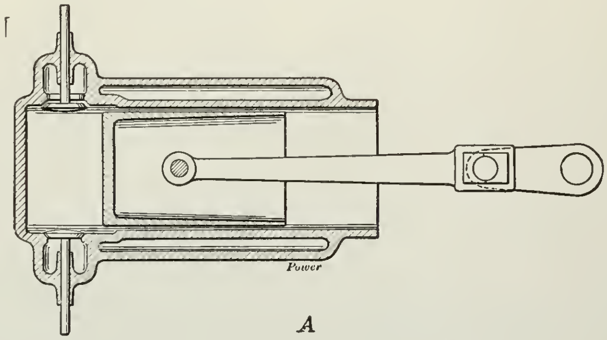


Fig. 1. Beginning of the Cycle

The engine itself affords a simpler example of cyclic operation. Consider one end of the cylinder alone; starting from the moment the admission valve starts to open, the cycle consists of steam admission, expansion and exhaust, or expulsion, which brings us back to the moment of admission again, when conditions are exactly the same as when we started. This is one cycle of steam-engine operation.

The gas-engine cycle is not nearly so simple because the engine combines both pressure generation and application, whereas the steam engine has to do nothing but apply the pressure generated outside of it. The gas-engine cycle consists of admission, compression, combustion (commonly called explosion), expansion and exhaust, or expulsion. When expulsion is finished, the engine is ready to begin taking in a fresh charge, which begins the next cycle.

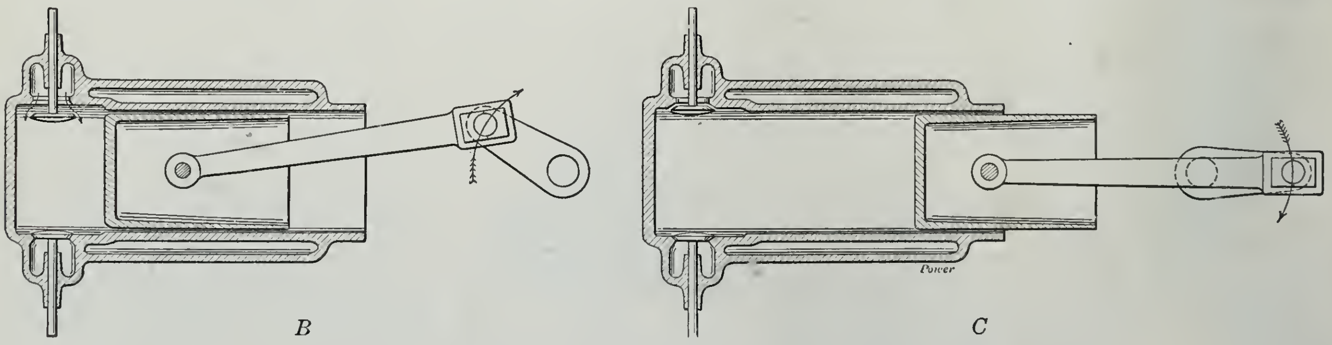


Fig. 2. The First Stroke (Suction); Drawing in the Charge

From a consideration of these facts, it should be plain that a so-called four-cycle engine does not perform four cycles per stroke or per revolution. One complete stroke is devoted to taking in the fresh charge, the next (return) stroke accomplishes the compression; at the end of this stroke, while the crank is passing the dead center, combustion occurs, producing the rise of pressure necessary to give power to the engine; then an outward stroke is devoted to expansion, during which the gases push the piston forward and deliver power to the crank shaft, and the succeeding stroke clears the burned gases out to make way for the next charge. To illustrate the cycle, the familiar diagrams of the four strokes, somewhat amplified, are presented herewith.

In Fig. 1 the crank is shown on the inner dead center and the inlet valve is just beginning to open to admit the charge of gas and air, which begins to be drawn into the cylinder by the suction of the piston a moment later, as represented at *B*, Fig. 2, where the inlet valve is shown wide open and the entrance of the charge is indicated by the two arrows.

While the piston moves from the position *A* to that of *C*, the charge is drawn in, and this constitutes the "suction" or admission stroke; the inlet valve is shown just about to seat, after which compression begins, as illustrated at *D*, Fig. 3. (A comparison of the piston and crank positions at *C* and *D* will show how deceptive it is to consider gas-engine events in terms of crank motion. Although at *D* the crank has moved about 19 degrees from the dead-center position, *C* the piston has moved about one-fiftieth of its stroke.)

In moving from the position represented at *D* to that at *E*, Fig. 3, the piston compresses the mixture in the cylinder, the valves remaining closed, as shown, and this is therefore termed the compression stroke. Just before the crank reaches the inner dead center at the end of the compression stroke, the igniter operates and the rise of pressure due to combustion begins; this continues while the crank is moving over the center, from the position at *E*, Fig. 3, to that at *F*, Fig. 4, the motion of the piston during this brief period being so small as to be negligible. This completes the third event of the cycle—combustion—and the fourth event, expansion, begins.

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*Although the cycles commonly used in gas engines have been explained before in these columns, letters received from regular readers during the past few months indicate that this elementary discussion will be of interest to a considerable number of subscribers. The four-stroke cycle here described is that in which combustion is assumed to be effected at "constant volume;" that is, while the piston is practically stationary.

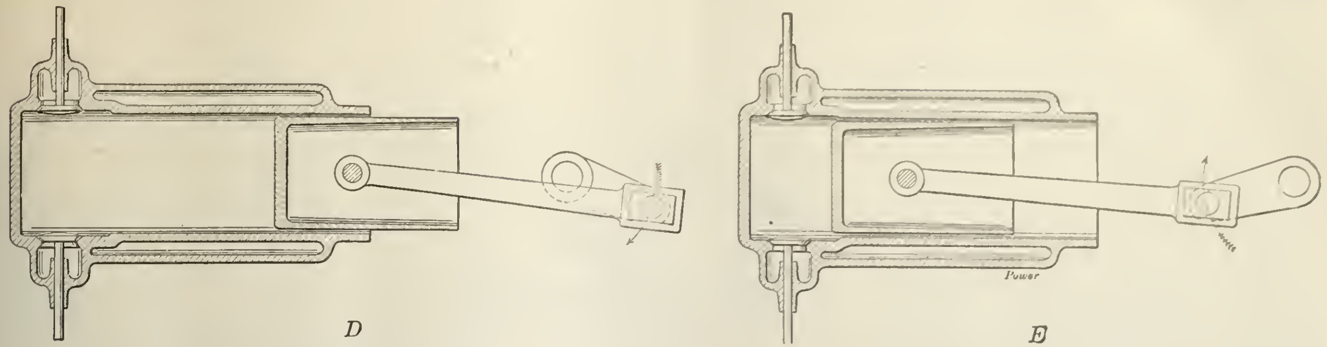


Fig. 3. The Second Stroke; Compressing the Charge

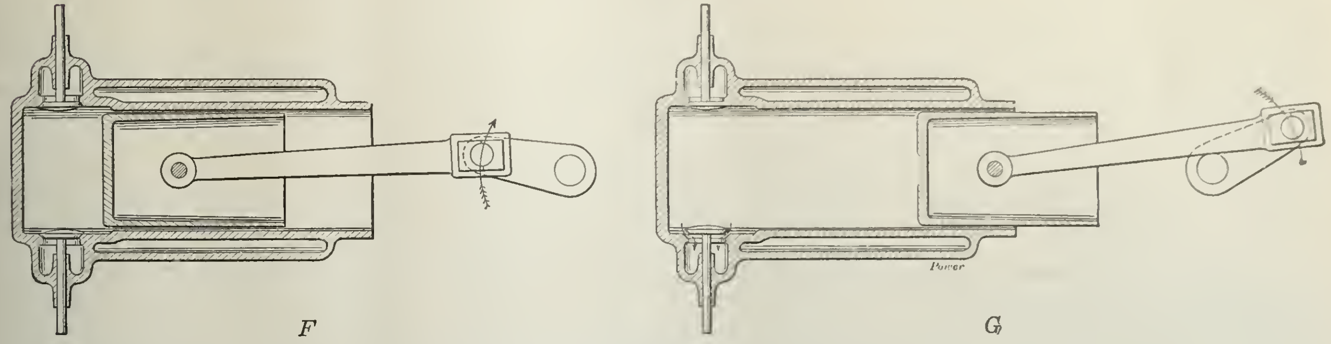


Fig. 4. The Third Stroke; Expanding the Hot Gases

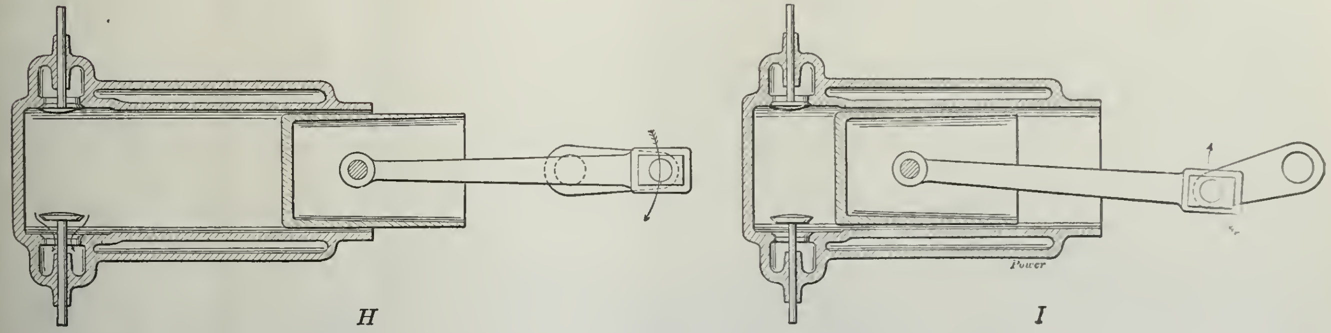


Fig. 5. The Fourth Stroke; Expelling the Dead Gases

Expansion, like all the other events except combustion, has practically a full stroke of its own. It begins as soon as the crank passes the inner dead center (*F*, Fig. 4), and continues during the outward stroke until the exhaust valve begins to open, just before the end of the stroke, as indicated at *G*, Fig. 4. While the crank is passing from this position to that of *H*, Fig. 5, the burned gases escape to the atmosphere under their own pressure, which is some 15 to 30 pounds above that of the atmosphere when the exhaust valve first cracks open. By the time the crank reaches the outer dead center (*H*, Fig. 5), the exhaust valve is wide open, and as the piston moves back on the instroke again, the burned gases are pushed out of the cylinder by the piston, which is, of course, being driven then by the flywheel if the engine has only one cylinder. The exhaust valve remains open, as indicated at *I*, Fig. 5, until the end of the expulsion stroke, which is the movement of the piston from the position *H* to that of *J*, Fig. 6, and then it closes and the inlet valve begins to open again, ready to take in the next fresh charge and begin a new cycle.

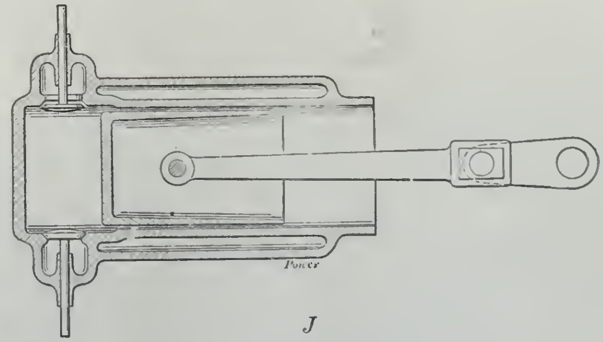


Fig. 6. End of the Cycle

Summarizing the foregoing detailed explanation, the four-stroke cycle consists of taking in a charge during one outward stroke, *A*, *B*, *C*; compressing the charge during one return stroke, *C*, *D*, *E*: combustion while the crank passes the inner center, *E* to *F*; expanding the heated gases to do work during a third stroke, *F, G, H,* and driving the burned gases out of the cylinder during a fourth stroke, *H, I, J.*

From this it should be clear that name four-stroke cycle (commonly miscalled four cycle) does not mean merely admission, compression, combustion, expansion and exhaust, but means that these five events are carried out in four strokes of the piston. The five events themselves constitute a gas-engine cycle; they can be carried out in two strokes, four strokes, six strokes or any other even number of strokes, and the number of strokes employed determines the type of the engine and, generally, the manner of performing the cycle. The statement that one sees sometimes in descriptions of new or proposed engines to the effect that the engine works on the four-stroke cycle, but completes the cycle in one revolution of the crank shaft, or some other period less than four piston strokes, is nonsensical because it contradicts itself.

The Bogart Single Acting Engine

A relatively recent addition to the list of single-acting horizontal engines is the Bogart machine, the single-cylinder form of which is illustrated herewith. The encrank construction in sizes up to 100 horsepower and of side-crank construction when built with tandem cylinders; the latter form is employed in sizes ranging from 50 to 500 horsepower.

Figs. 1 and 2 show external and sectional elevations, from the same side of the engine. As Fig. 2 indicates, the main construction differs from the more usual arrangement in that the cylinder is supported near the rear end by a pedestal instead of being overhung, and is bolted to the frame at its extreme forward end. The rear end of the cylinder is also of unusual design. The combustion chamber is reduced symmetrically in diameter, beginning at the point where the piston travel stops, and the breech end is closed by a removable valve chamber carrying both the inlet and exhaust valves and the mixing valve. This obviates any packed joint common to both the cylinder bore and the water jacket, and therefore renders leakage between the two impossible. The valve chamber is water jacketed separately.

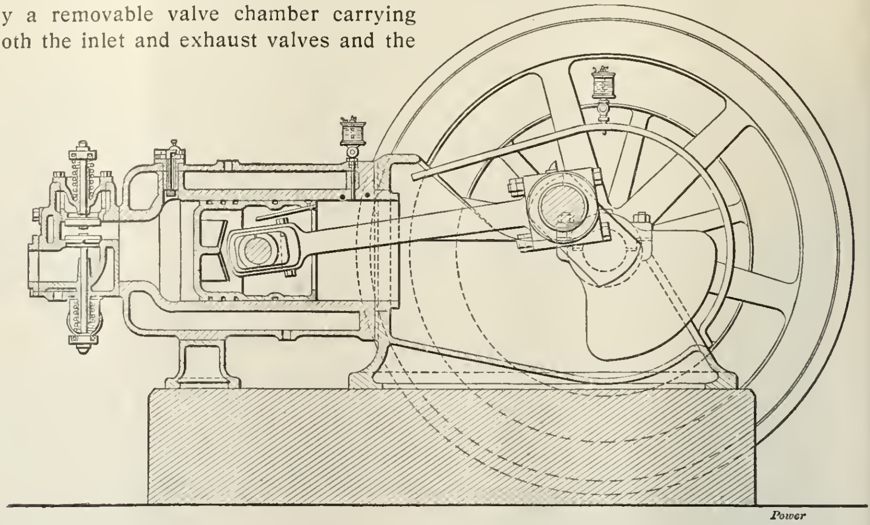


Fig. 2. Longitudinal Section of Bogart Engine

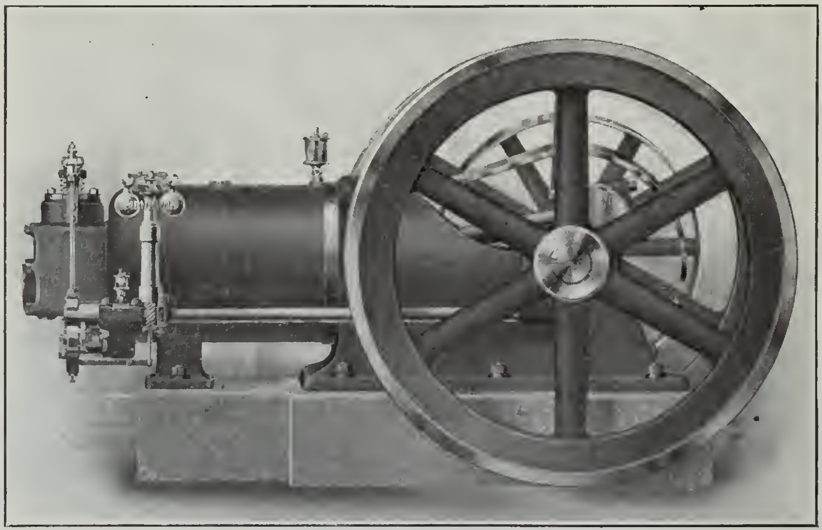


Fig. 1. Bogart Gas Engine

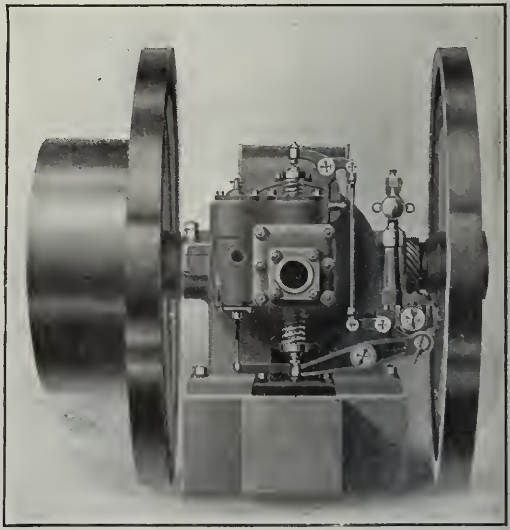


Fig. 3. Rear View of Engine

In small sizes all of the jacket water, and in larger sizes a part of it, is discharged into the exhaust pipe near the engine, and this water is taken out farther along by a separating muffler. The objects of this arrangement are to keep the exhaust piping cool near the engine and to quiet the exhaust.

The valves are of the usual poppet type, opened by cams and closed by springs. The inlet and exhaust valves are in line with each other vertically, as shown in Figs. 2 and 3, and a combined mixing and throttle valve is located in a cage on one side of the main valve chamber. Fig. 3 shows this construction and also shows the arrangement of levers, push rods and cams for operating all of the valves. The mixing valve is of the multiported cylindrical type and is moved by the governor to control the admission of both gas and air according to the load variations. Regulation, therefore, is effected by the constant-quality and variable-quantity method.

The ignition system is of the make-and-break class, with an igniter operated by an "iron-clad" electromagnet, and a separate inductance coil ("spark" coil) connected in either leg of the circuit. The igniter is located in the top of the combustion chamber near the end of the piston travel, as represented in Fig. 2. The timer is adjustable, of course, so that the igniter can be made to trip at any point in the crank travel within the practical range.

The constructions of the main frame, shaft, connecting rod and piston are so simple and so clearly illustrated by the engravings that a description of them is unnecessary.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Long Runs of Power and Pumping Machinery

I am anxious to obtain information relating to long record runs of power and pumping machinery, either of the reciprocating or turbine type, especially the former. No doubt the readers of POWER are in possession of highly interesting facts in this connection and mention of them in the columns provided for that purpose would be appreciated by the engineering fraternity in general.

EDWIN D. DREYFUS.

East Pittsburg, Penn.

Boiler Inspections

So much has been said upon this subject that it would seem there was very little more to be said. There is, however, one point in connection with boiler inspection that is not given the attention that seems to be necessary from what is known to prevail generally throughout the country. This relates to the proper preparation of the boiler for the inspector.

If steam users and boiler owners would give more attention to such matters, they would save during the year, perhaps thousands of dollars. They insure their boilers, but they do not get, in many instances, the kind of inspection they should have, for the reason that the boilers are usually not properly prepared. Owners apparently think that they cannot afford the time to shut down for an internal inspection, and as a consequence, the inspector has to do the best he can under the conditions, fully realizing that his inspection is anything but satisfactory.

When a boiler has been shut down with the idea of making an inspection, it is often needed again within twelve or fourteen hours, and as a matter of course, it is impossible between the shutting down and starting up again to either clean or inspect the boiler thoroughly. There is not time, for the reason that the boiler is too hot, as no boiler set in brickwork can be cooled down properly under twenty-four, and sometimes thirty-six, hours. It is not of much use to the owner or to the insurance company to have an inspector go into a boiler that is so hot that he is obliged to come out for breath every few seconds; under such conditions he is unable to stay in the boiler long enough to give it the necessary inspection. Besides, the boiler should be thoroughly scaled and cleaned before the inspector goes in, to enable him to get at the sheets covered by scale, as many a dangerous crack has been hidden beneath a covering of scale.

Insurance inspections are made periodically, and the date at which the inspector is due, is easily remembered if any interest is shown. When the inspector arranges for the internal inspection, the boiler should be shut down, the plates taken off as soon as possible after steam is off and the boiler allowed to cool thoroughly, after which the process of cleaning, scalding and washing out should be started, the grate bars removed, the combustion chamber thoroughly cleaned and all soot brushed from the fire surfaces with a wire brush. Then, and only then, can the inspector do justice to the owner and to his own company.

Do not close your boiler if the inspector happens to be an hour late in arriving at your plant. He has other boilers to inspect, and perhaps the boiler that he inspected at the last place was so dirty that he had been obliged to secure extra clothing to enable him to keep his appointment with you, or he may have been delayed through other causes.

This is mentioned particularly because a very bad explosion might have been prevented on a certain occasion if the engineer had not been in such a hurry to close his boiler. This inspection was arranged for 11 a.m. on a Sunday. The inspector having been delayed at another plant arrived about noon and found the boiler closed. The engineer upon being asked why he did not wait a little longer, replied that he could not wait, as he wanted to go home. The inspector naturally felt annoyed, but the best he could do was to make an appointment to inspect the boiler on the following Tuesday—two days later. On the day intervening, Monday, the boiler blew up, fortunately killing no one, but entirely demolishing the plant, with a property loss of over $23,000. From the condition of the plates which were examined after the explosion, it is certain that had the inspector gotten into the boiler on Sunday he would have condemned it.

It will pay any concern to shut down its boilers two or three times a year in order to permit the inspector to make a thorough internal inspection, as the opportunity given to clean off the scale, and otherwise clean the boiler, will amply repay in fuel saved, for the time taken. The loss in heat transmission due to scale of the thickness of 1/16 of an inch will amount to as much as 10 or 12 per cent., and we have records of more than one case where, after the boilers were thoroughly scaled and cleaned and settings slightly altered, a saving in coal of over 60 per cent. was attained.

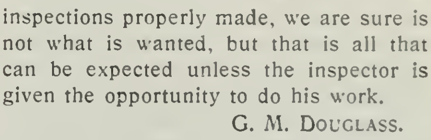
The insurance inspector visits scores of steam plants yearly, finds all kinds of apparatus and conditions, has new experiences, and is uptodate in matters pertaining to steam-plant equipment, and is in a position to advise sensibly. Inspection work is not ideal by any means; the work is hot, dirty and disagreeable. The inspectors strive to perform their duty in trying to detect weaknesses, defects and dangerous conditions if they are given the opportunity. Make an effort to have the boiler properly prepared for his inspection, and give him a reasonably cool and clean boiler to inspect. He will appreciate it and the owner will be getting not only his money's worth under the policy, but save hundreds of dollars annually besides. Boiler insurance without internal inspections properly made, we are sure is not what is wanted, but that is all that can be expected unless the inspector is given the opportunity to do his work.

G. M. DOUGLASS.

New York City.

Are the Rim Bolts in Tension or in Shear?

At a plant in this locality a new driving pulley is being installed and is causing much discussion. It is 13 feet in diameter with a 72-inch face and is built up in halves with arms as shown in the sketch. The wooden rim is to be placed after the wheel is in position and will be held to each of the arms by four 1½-inch bolts.



HALF OF PULLEY WITHOUT RIM

The controversy which has arisen among the engineers is in regard to the stress on these rim bolts. Some say that the bolts will be in shear while others say that they will be in tension. Will some reader of POWER kindly settle the question?

PATRICK MULHAM.

Adams, Mass.

[This letter is reproduced from the July 5 issue. The sketch was accidentally omitted when the letter was first published.—EDITOR.]

Vibrations in Boiler Header

Last fall the writer had an experience in a lighting plant in Oklahoma which, although nothing serious resulted from it, he sincerely hopes he will never again be called upon to pass through.

It happened before the lighting load came on in the evening. The engineer had fired up one of the boilers and had cut it in but had not yet cut out the other one when the steam header began to vibrate lengthwise. The vibrations slowly but gradually increased until the engineer and the writer, who happened to be in the plant at the time, became alarmed and set about trying to find the cause of the trouble. As the vibrations appeared to be in step with the stroke of the engine, which was a Corliss, running at 100 revolutions per minute, we decided to start up another engine which ran at 325 revolutions per minute and then shut down the Corliss.

However, this change did no good, and the vibrations kept on getting stronger until the header appeared to move about an inch back and forth and the asbestos covering began to shake loose. We then decided to start up the Corliss again and to shut down the high-speed engine before the Corliss was up to full speed, hoping to thus set up vibrations out of step with the header. This ruse succeeded and the header began to quiet down so that by bracing it with planks we were able to bring it to a standstill without interrupting the service more than three minutes.

The question as to "what caused this trouble" naturally arose, but we were unable to account for it. Boiler No. 1 had been removed a short time before and a new one installed in the place of the old one at No. 2. The header had also been turned over so that No. 2 could be connected to it from above instead of from the side, as before. Aside from these changes the installation was as it had been for several years.

Another question arises: How far is it an engineer's duty and to what extent is he justified in risking the machinery and probably his life, in an attempt to maintain uninterrupted service? The public does not know and does not seem to care what risks an engineer takes, providing the service is not interrupted, but it is quick to complain if a shutdown occurs.

G. E. MILES.

Denver, Colo.

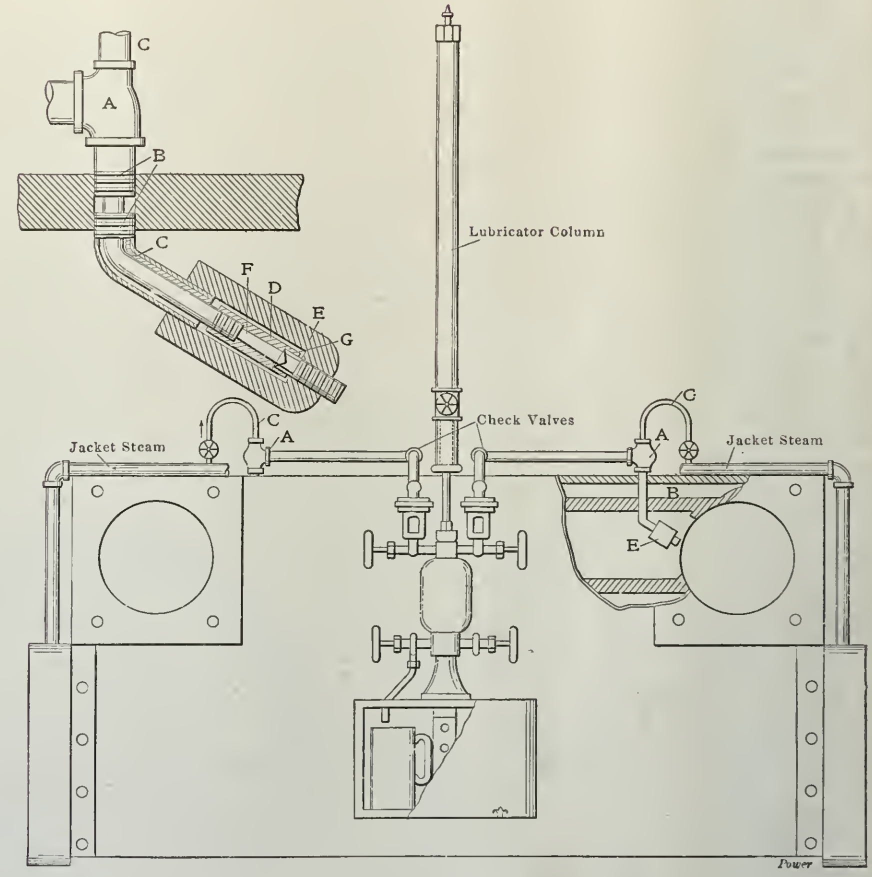
Lubricator Loop

There has recently been much discussion concerning the quantity of oil fed to steam-engine cylinders, but little has been said about the best ways of introducing it into those cylinders. The following method is a little out of the ordinary, but is taken from actual practice. The illustration shows in detail a method of atomizing the oil as it enters the low-pressure cylinder of a 20x40-inch cross-compound Corliss engine. The lubricator has a double feed, with a branch leading to a point a little to one side of the opening edge of each of the steam valves. Both feed branches, including nipple *B*, are of ⅜-inch pipe. Entering the reducing tee *A* is a ⅜-inch pipe *C*, which connects with the inflow pipe of the cylinder jackets, from which is taken the steam necessary to atomize the oil. It will be noticed by referring to the detail in the upper left-hand corner, that *C* is continued inside the nipple *B* and ends in the special expansion coupling *D*, which is inside the reducing coupling *E*, the latter capping the oil inlet inside the valve chest. The small hole *F* in coupling *D* is 1/32 inch in diameter at the upper end and increases to ⅛ inch at the lower end, its purpose being to expand the high-pressure steam, so as not to create any back pressure on the sight feeds through the small siphon holes *G*.

When the cylinder head was removed after this system had been in operation for a while, the usual unatomized oil at the bottom was lacking, and the cylinder began to rust inside of 24 hours, from which fact it was concluded that atomized oil has very little power to cling to the surfaces and penetrate the metal.

K. P. EVERETTE.

Boston, Mass.



DETAILS AND APPLICATION OF LUBRICATOR

Burning Oil Fuel

The purpose of this letter is to describe how a few changes in our oil-fuel installation cut down the oil consumption from 33 to 28 barrels a day; a saving of five barrels in 10 hours' run.

When I took charge, the steam connection to the burners was taken from a safety valve attached to a steam drum in the rear of the boilers about 14 feet from the front end. This pipe extended to the front of the boilers and down to the burners on one side, then across on the top to the other side and down again to the front burners. The plant was found to be burning too much fuel. An expert was sent for to investigate the cause but failed to improve matters. The flame frequently went out without any apparent cause, and this necessitated relighting the burners, on some occasions several times an hour. Having previously used this type of burner with satisfactory results, I was convinced that the fault was not in the burner.

After due consideration, I decided that the chief cause of the trouble was due to accumulated moisture in the steam, caused by the unnecessarily long steam-pipe connection to the burners. I therefore removed all the piping from both sides, cut and threaded a hole in the top of the front steam drum just back of the breeching and carried the steam pipe through the breeching, making connections at the front in the old way. This change proved most beneficial; the trouble with the burners disappeared, and a big saving in fuel was at once noticeable. A significant fact is that the superheating is now done with waste gases.

We had a heater in connection with the oil pump but the oil, in my opinion, was not heated to the most satisfactory degree, so I also passed the oil pipe through the breeching and attained still better results.

I had noticed in the *Report of the Government Liquid Fuel Commission* that the heating of the air prior to combustion was recommended. Here again, I made an improvement in the furnace and closed up the air space immediately back of the "target." This space was supposed to allow just sufficient air to complete combustion, but this was not necessary; in fact, it was rather a detriment. I therefore increased the air space in front, which proved both economical and satisfactory.

Careful experiment, as well as extended experience, has shown conclusively that the more burners used under a battery, the greater will be the economy in fuel. However, it is best to use as little steam as possible, just enough to properly atomize or spray the oil.

The "target" is most beneficial as it maintains the heat, and if for any cause the burner should fluctuate, the heat from the bricks in the target will prevent the flame from going out.

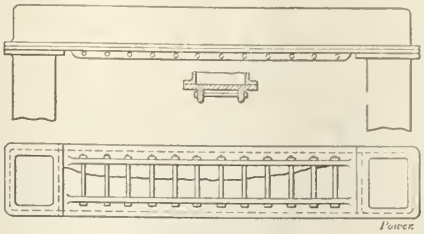
The draft has also considerable to do with the effective burning of liquid fuel; the bottom doors of furnace should be opened only just sufficient to admit the proper amount of air necessary for complete combustion. The tuyere holes and all other air holes in the boiler front should be entirely closed. The damper also should be regulated to get the best results; in ordinary cases the damper when one-third open will be found to be about right. A considerable saving is effected here, over the damper being wide open as is frequently the practice. But local conditions must determine how wide open both damper and bottom door ought to be. If a mud drum is used, be sure that it is properly covered and made air tight, as an increased fuel consumption is sure to result from air holes in the back or sides of the furnace.

T. K. McCarter.

Temple, Texas.

Repair of a Casting

A leak developed in a casting forming part of the intercooler of our air compressor. As shown in the sketch it extended nearly the whole length of the casting, a distance of about six feet, and opened up about ⅛ inch.



Crack in Casting

The conditions of service were such as to preclude shutting down the compressor for any length of time and as a result we were at a loss as to how the repair might be accomplished. It was finally decided to try "Smooth-on." The ribs were drilled for twelve ½-inch bolts, the "Smooth-on" applied to the crack, and the bolts inserted and tightened. This produced a solid joint and we have since been running, for some time, without any signs of a leak.

F. W. Bellinger.

Harve, Mont.

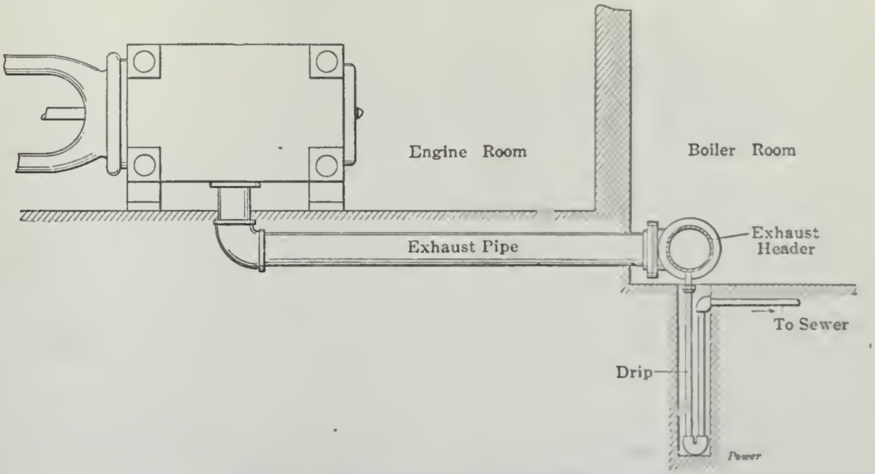
Drip for Exhaust Pipe

The accompanying sketch shows a drip that I used successfully in cases where there was but little back pressure on the engine. It consists of a loop of 1- or 2-inch pipe, depending upon the size of the exhaust header, which is sunk into a hole in the floor as shown. The depth depends upon the pressure in the exhaust pipe, the column of water serving to balance the steam pressure and prevent the escape of any steam. The drip is open at all times and allows the excess water to overflow into the sewer.

Where the pressure in the exhaust line is not over one pound, a loop about 2½ feet deep has been found sufficient. It is an easy matter, however, to figure out the length of loop required for any given back pressure.

R. L. Mossman.

Tampa, Fla.



Drip Connected to Exhaust Pipe

Runaway Engines

I have noted many serious accidents occurring through the bursting of a flywheel, which has generally been caused by the engine running away.

It has occurred to me that all of these accidents could have been prevented if a stop valve had been placed in the main steam line leading to the engine. Of course, this valve must be placed inside the engine room, and within easy reach of the engineer. Then, if the throttle valve should stick or the governor break, causing the engine to travel at a dangerous speed, the engineer could always regain control of his engine by closing the stop valve.

To prove that a stop valve so placed has been the means of preventing an accident due to a runaway engine, I will relate the experience of a friend of mine who was in charge of a cross-compound Corliss engine. One day, on account of the steam pressure being low, he had occasion to open the throttle valve wide, to keep the engine running at normal speed. Shortly after this, the steam pressure began to rise rather rapidly and the speed of the engine increased. My friend ran to the throttle valve but to his dismay found that the wheel would not move. By this time the engine was traveling at a dangerous speed. Displaying presence of mind, he rushed up some steps leading to an elevated platform, from which he closed the stop valve. There is no doubt but that the accessibility of this valve, together with the engineer's efforts, were the means of saving a serious accident.

The engine had to be kept running day and night until the end of the week, so for the rest of the week the regulating had to be done by the stop valve. After the engine was shut down, the throttle valve was opened for examination, and a large nut was found jammed in it, but where the nut came from has never been solved.

C. Trube.

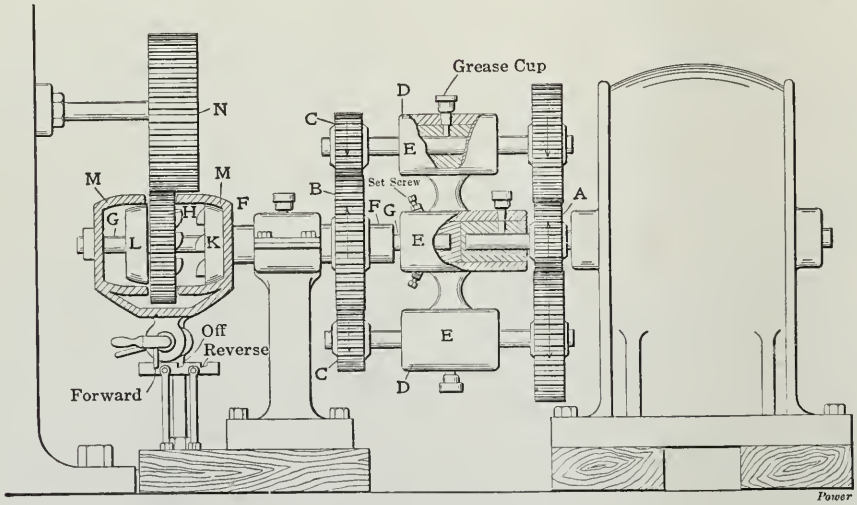
McKeesport, Penn.

A Complicated Drive

The illustration shows a rather curious form of drive between a motor and a large machine, and is an example of remarkable ingenuity for complicating a simple drive.

The problem was to reduce the speed of the motor to produce a forward and a reverse drive, and to stop the machine without stopping the motor.

If it be supposed that the motor gear *A* turns in the direction of the arrow, then gear *B* will turn as indicated; but if a brake be applied to gear *B* to stop it from revolving, then gears *C* will tend to run around it, and the two arms *D* will be revolved in the opposite direction to that of gears *B*. The bearing boxes are represented by *E*. The gear *B* is keyed to the quill shaft *F* and the two arms *D* are keyed to the solid shaft *G*. The direction of motion of these two shafts gives the forward and reverse drives to the machine. Gear *H*, on each side of which heavy lugs are cast, is free to move on the shaft. The castings *K* and *L*, the former keyed to shaft *F* and the latter to shaft *G*, have also heavy lugs cast on their inner faces to engage with those on the gear whenever it is pushed by the shifter *M* toward them. This causes the machine gear *N* to revolve in either direction desired. *K* and *L* are stopped revolving by friction between their beveled faces and those on the shifter *M*. This system of transmission has one good point about it, in that there is no shock to the driving lugs, for the machine will not reverse until the lugs are engaged and brake applied to the other drive.



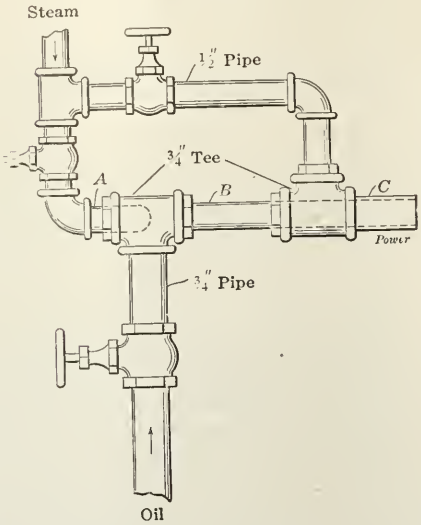
Arrangement of Gears

S. Tremont.

Cambridge, Mass.

Homemade Oil Burner

The accompanying illustration shows a homemade oil burner which the writer has employed successfully for some time.



Homemade Oil Burner

There are two of these in operation, one on a 75-horsepower boiler and the other on a 50-horsepower boiler, and they furnish all the steam that is required.

The burner is made entirely of pipe fittings. The ½-inch nipple *A* is simply forged down on the end to a 3/16-inch hole and a long thread is cut on it, in order to bring the end of the jet about ½ inch from the junction of the tee and thus form a siphon to draw the oil in. The nozzle *C* is a piece of ¾-inch pipe about 4 inches long, cut off straight at the end with all burs removed. The inner piece *B* is of ½-inch pipe cut with a long thread and ending flush with *C*, the threaded portion within *C* is filed or turned down, leaving a clearance of 1/32 inch all around through which the steam is sprayed.

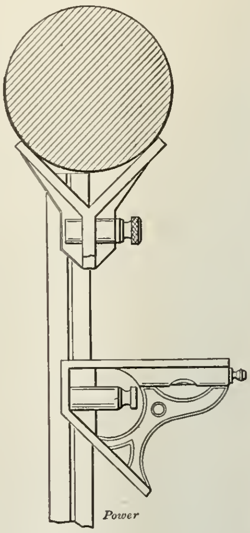
The burner is regulated by the valves in a manner similar to any other burner.

H. S. Armstrong.

Jeanerette, La.

Laying Out a Key Seat

The accompanying sketch shows a convenient method of laying out the key seat on a shaft, by the use of a combination square. The illustration is self-explanatory and needs no further comment.



Laying Out a Key Seat

F. M. Sterling.

Middle Branch, O.

Proper Way to Drain a Boiler

I should like to hear from some of the readers of Power as to the proper method of draining a header of a high-pressure boiler.

I have seen three different methods used. First, a 1½-inch pipe leading from the bottom of the header to a trap and discharging into the heater. Second, a 1½-inch pipe draining the bottom of the header, with a pitch of 6 inches in 10 feet, and additional drains in the back head of the boiler above the water line. Third, the pipe is located similar to that in the second case, except that instead of entering the header it drains through the blowoff pipe, being supplied by a swing-check valve and a stop valve.

Is the latter method practical? It is my opinion that it is not. I know of a case in which two boilers are run in the morning and at 10 o'clock one is banked until 3 in the afternoon. The header valve is left open all the time and while at times the boiler will fill up with water in two hours, yet at other times it will fill at the rate of only one inch in five hours. This is because the valve becomes obstructed with foreign matter.

H. M. Walker.

Chicago, Ill.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Trouble With Governor Bearings

An article by G. E. Mills in a recent issue brings to my memory an experience along the same lines with a high-speed engine and sticky governor. I was employed at the time as a fireman in a sawmill and as we were running overtime the engineer made it a practice to go home to supper at 5 o'clock and leave me in charge. There was a small engine belted to a direct-current dynamo to furnish lights to the mill and when light was wanted, the saw operator let it be known by sounding the whistle.

One night the engineer, as usual, had gone home to supper and the operator whistled for lights. I went into the engine room and started the dynamo engine, first getting it warmed up and running up to speed with no load. I then stepped over to the switchboard, raised the voltage to normal and started to throw in the switches. As soon as the load began to come on the voltage dropped, but by throwing the switches in and out a few times, I managed to get light.

I mentioned this to the engineer when he returned and he laughed at the idea, but nevertheless he took precautions and had the night watchman smear the governor bearings with coal oil. The next day he dismantled the engine and cleaned up the rollers in the governor bearings and no more trouble was experienced.

F. S. Punches.

Williamsport, Penn.

Increasing the Capacity of Boilers

With reference to the article in Power of May 24 in relation to increasing the capacity of steam boilers, I presume, after a careful reading of the article, that the data were taken from tests made, and call attention to the fact that one may secure under test conditions results which it would be imprudent if not impracticable to secure with stationary boilers operating under common conditions, for the following reasons: In 95 per cent. of such boilers we have scale to contend with and with an evaporation of, say, 4 pounds of water per square foot of heating surface the solids set free are to be considered. If the evaporation of a given boiler is doubled, the amount of solids is doubled also. Hence, the boiler must be cleaned twice as often one will say. But with double the evaporation per square foot of heating surface would not it be absolutely necessary to clean four times as often, having in view the increased danger from solids on the fire-swept surface on account of the greater amount of heat generated? Granted that solvents are used, would one care to operate a boiler under such forced conditions, knowing that the contained water was a solution and held matters in suspension, its capacity for absorbing heat diminished, and the danger of burning the sheets or tubes clearly increased? That we may do this in tests running a few hours is, of course, admitted, but the practicability of continuous operation with average feed water I seriously question.

That we should not have scale in steam boilers is quite true, but owners are seldom directly interested in ideal conditions and boilers must be run for power purposes, to produce results in the way of manufacturing. The owner is interested mainly in output and to a lesser degree in keeping down expenses in the power plant.

The problem of scale, therefore, does not warrant an increase in the present rate of evaporation per square foot of heating surface, which runs from 3 to 4 pounds. Indeed, even with this rate the owner has difficulty in keeping his boilers in constant operation and objects viciously enough when the repair bills due to scale are presented. True, we have certain localities, notably Massachusetts, where nature has granted water free from objectionable matter, but in the main the scale problem is a factor to be considered. Even where the solids are in small proportion the human equation enters and cleaning is neglected until overheated metal calls attention to the neglect. In my opinion, before increasing the rate of evaporation, the scale question must be disposed of.

In considering the evaporation per square foot of heating surface it might be said that the term is misleading. Consider a boiler with horizontal fire tubes. Assume a fire-tube boiler, 72 inches in diameter by 18 feet in length and having 70 tubes, 4 inches in diameter. We have apparently 1319 square feet of heating surface. Assume the rate of evaporation to be four pounds per square foot of this surface. Now it is conceded by all authorities that the actual heating surface, that is, the surface used, is one-half this amount for, beyond doubt, the lower half of a horizontal fire tube is of little value in absorbing heat. Hence, the real rate of evaporation is eight to one in a tube of this class and probably from five to six to one with a horizontal or semi-horizontal water-tube boiler. This should be borne in mind in calculating the evaporation with every type of boiler using tubes. Along this line and with a horizontal-tubular boiler it would be interesting to know the amount of evaporation per square foot of shell plate over the fire and then to remember that in doubling or tripling this rate we must increase proportionally the evaporation on the shell plates and that in these plates rests our security from explosions.

With regard to reducing the size of the tubes in order to absorb the heat generated by a doubling of the draft, it is evident that with boilers of the horizontal-tubular type we must at once look after the matter of soot regardless of the statement that soot will not occur due to the increased velocity of the gases. As the temperatures in the tubes drop notably farthest from the furnace, soot will accumulate. Western coals, commonly used, are notorious for soot and certainly when we arrange to double the amount of coal burned per square foot of grate then we have double the soot to contend with. Hence, with smaller tubes the result would be decreased evaporation or extra expense in removing soot. Possibly the experimenters will point out how to remove this evil with ordinary furnaces. Certainly we would appreciate their advices on both the soot and the scale ends of the subject.

In considering the doubling of the rate of evaporation in water-tube boilers, besides bearing in mind the foregoing, we must remember that the water and globules of steam are confined by the wall of the tube, the escape usually being at a given end. That part of the tube directly over the fire has the greatest work to do. Just how much water is evaporated in such a tube per foot under given conditions we would greatly like to know. If we assume a four-to-one ratio for the entire heating surface, what would be the actual rate at this point, say with a Babcock & Wilcox, a Stirling, or Heine type? Tests made of the Hohenstein boilers by the Navy Department showed an evaporation of 16 pounds of water per square foot of heating surface with oil fuel and with 275 pounds, gage, steam pressure. Such tests are, of course, valuable for naval purposes. A warship might for a few hours in its life need such high evaporation with feed water free from scale due to surface condensers, but, even so, before stationary engineers consider the adoption of such high rates may we not await the naval engineers' solution of their troubles with tube failures in times of peace when boilers are run at perhaps ordinary rates of evaporation?

Granting the advisability of increasing the capacity of a boiler, doubling, tripling, or quadrupling the same as the gentlemen suggest, and assuming that they will look after the scale problem and the ash deposits, it would appear that the vertical fire-tube boiler would be the most promising one for results. All of the heating surface is available in this type. We could dispense with the water leg for stationary purposes, and with a reverberatory-brick furnace, fitted with grate surface to suit, we could use small tubes of any required length. It should be noted that in this type of boiler the globules of steam are not confined as in other types hence the safety under forced draft. We could even apply spiral baffles projecting part way down the tube for forcing the impingement of the gases against the sides of the tube. The heat absorbed by such metal baffles would be transmitted to the water. The shell, not exposed to the gases, would not be subjected to the severe expansion and contraction as in an externally fired boiler and our factor of safety would run up to 6 or better in the beginning, and remain there.

Further, in this type we have a superheater giving excellent results at minimum first cost, with practically no expense for upkeep. With a given fuel and a fixed draft, one could easily determine the proper length of a given size of tube to get a certain amount of superheat. With 200 pounds gage pressure, 387 degrees, one could get superheat within reasonable limits by raising or lowering the water level. With other types of boilers the superheater is a separate affair frequently located where it must be carefully looked after and where repairs are expensive. With the vertical, fire-tube boiler, we can get 50 to 90 degrees of superheat at little first cost and with waste heat.

Then again, with this type of boiler, the major troubles with brick settings are eliminated, for from a reverberatory furnace the products of combustion flow directly into the tubes similar to the delivery of gases into the chimney of a puddling furnace. With a short low-walled chamber surrounding only the furnace and the bottom of the boiler we would save in first cost of brickwork, in repairs to same and in avoiding heat losses due to the cracking of the walls which is common with boilers incased in long and high walls. The matter of insulating the shell to prevent heat losses is a simple one, comparing its temperature, 387 degrees, against even the lowest temperatures in the furnace.

Again, with this type, floor space is saved as against the horizontal boiler. That head room is needed is true, but in most instances this is not an objectionable feature. In the above I have pointed out what seems to be valid objections to increasing the capacity of steam boilers of the stationary type under existing conditions. Under tests we may get certain results and these tests are certainly of value in blazing the trail for designers. On the other hand, if the gentlemen engaged in the fascinating occupation of solving boiler troubles, at the expense of our genial relative Uncle Sam, will kindly devote their attention to the scale problem and solve this satisfactorily, we, as engineers, boilermakers and inspectors, will more readily listen to the argument in favor of increasing the boiler capacity by means of forced draft.

T. T. PARKER.

New York City.

Vacuum Trouble

I read with interest Mr. Reynolds' account of his "Vacuum Trouble" in the May 31 issue. I have recently had trouble in this line myself.

The pumping equipment in a certain plant consisted of a horizontal duplex triple-expansion condensing pumping engine having a surface condenser and an independent air pump. This pumping engine was working against a head of 60 pounds and a suction lift of 15 feet. The machinery was new, the plant having been running only about two months. At the time in question, the air pump was carrying a vacuum of 27 to 27½ inches. Suddenly I noticed that the vacuum started to drop and a peculiar knocking began in the pump. I was unable to locate the sound and thought at first that it was in one of the steam ends and that one of the piston heads had come loose, but upon investigating I found that the three heads on both sides were firmly in place. I started the pump again but the knocking still continued although the vacuum held up. I thought then that something might be under the water valves but upon stopping the pump and opening the water ends, I found all of the valves in good shape and seating nicely. Upon starting up again, the vacuum went up to 27 inches; suddenly, the knocking started and the vacuum began to drop, going back to 15 inches. Thinking possibly that a tube was loose in the condenser, I took the head off the condenser and after cleaning the tubes, tested it thoroughly by placing a blind gasket on the intake end (the suction line had no foot valve) and bypassing the water from the discharge line into the condenser. Everything was snug. Then I tried the bypasses from the discharge decks of the water cylinders to the suction decks, but found everything tight and the valves holding in good shape.

As a last resort, I went over the suction line and found a number of minute holes in the two cast elbows in the line. I painted these with a light preparation of metal cement and found when I started up that the knocking was gone and that the pump held the vacuum nicely.

I do not know why this trouble did not show when the pump was first started unless it was because these many small holes were filled with a fine powdered dust or sand from the casting and this had not had the time to work out.

M. BAILIERE.

Norwalk, O.

Fake Diagrams

Mr. Westerfield's letter in the issue of May 10, in reply to my questions in the issue of March 15, is a very poor, slip-shod explanation of the defects shown by his Fig. 1 in the issue of January 18.

The shortness of the diagrams at the point of release he calculates to have explained, but neglected to blame the engraver for the very sharp point of cutoff.

Mr. Westerfield states that I should be able to see that the back-pressure line has been omitted. Just what connection the back-pressure line would have had with the point of release, or the length of the diagram, requires explanation. My eyesight is good enough for me to see that the back-pressure line is very clearly shown, instead of being omitted; and that the atmospheric line is omitted.

On the same page, in the issue of January 18, Mr. Westerfield gives us another account of his experience.

His statements in this letter do not hold water, and the letter was answered by readers, pointing out the variations in the statements, but no explanation in reply was made by Mr. Westerfield.

In the December 21, 1909, number, he also gave us some of his experience as a "Trouble Man." The diagrams which he showed with this letter were very strange; in fact, I believe such a condition as he described did not exist, and I said so in my letter, replying to his article.

Mr. Hughes also disputed the correctness of Mr. Westerfield's statements. These diagrams in Fig 1 show a very good steam line and an excellent point of cutoff. Anyone who has had much experience with indicators and valve-gear adjusting knows that it is almost an impossibility to get so sharp a point of cutoff.

The diagrams in Fig. 2 are good average diagrams; notice the difference in the point of cutoff shown in Fig. 2 compared with that in Fig. 1.

Perhaps Mr. Westerfield *has* had a very extensive indicator experience, but I know and so should he that such a sharp point of cutoff is an impossibility from the same engine as that from which the diagrams in Fig. 2 were taken. I am not satisfied with his explanation and still say that the diagrams do not look genuine.

Mr. Kirlin also disputed these diagrams, but Mr. Westerfield failed to defend himself in that case.

Here are three letters from Mr. Westerfield, and all of them have been disputed by different readers.

The only defense Mr. Westerfield makes is that in reply to my letter on page 496 in the issue of March 15, and in this he only explains half of what I disputed; the half which he does explain he does so by blaming the engraver.

A poor excuse is worse than none.

Harry W. Benton.

Cleveland, O.

Central Station vs. Isolated Plant

I read with much interest Mr. Naylor's letter on the above subject in the May 3 issue. To a man in Mr. Naylor's position, familiar with the situation in all its details and thus able to judge and weigh everything for or against the central-station service, the proposition no doubt looks quite different than it does to many readers of Power. We know what would be the probable cost of a kilowatt delivered to the busbars in a well managed noncondensing plant of the size required to supply the service as stated, but with no details in regard to the heating and ventilating requirements in this particular case, it is merely guesswork for an outsider to try to size up the situation. Where, say, 2000 boiler horsepower is necessary seven months in the year for heating purposes alone, it would seem that the coal- and ash-handling proposition would not be much different whether the boilers were operated at high pressure or not; and as 2000 boiler horsepower would go a long way toward supplying power for light, it would be interesting if Mr. Naylor would give us an idea of what it means to supply heat and ventilation for a mammoth establishment such as he has in charge.

As to the question, "will the isolated plant be able to withstand the onslaught of the central station?" it is the writer's opinion that most any properly laid out and well managed plant will for several reasons. It is true, of course, that the central station has rapidly reduced the cost of producing current through favorable location, ability to run condensing, low cost of handling coal and ashes and minimum of attendance per unit of output. But at the same time, the central station has now reached a point where further reductions in cost of producing current mean such small fractions of a cent that the isolated plant in charge of the National Association of Stationary Engineers man is producing a reduction in cost which is rapidly catching up with the figures established by the central station. In addition, in the isolated plant, good use can be made of the exhaust steam where condensing facilities are lacking. Besides this, the isolated plant does not have to contend with the expense of transmitting current to long distances, which must be quite an item with the central station in a large city like Chicago.

The central station in Chicago has increased its output at a tremendous rate, in fact, judging from contemplated extensions, it seems that the demand is steadily in excess of its capacity. But it is equally true that most of the modern office buildings, hotels and factories erected in the last year or two in Chicago and those now under construction have their own private power plants.

Whether this is because the building owners find it more economical or because the central station is not equipped to handle the additional load would be interesting to know and I trust we will hear more from readers of Power who are in position to give reliable figures in regard to this question.

R. Cederblom.

Chicago, Ill.

Oil as Fuel

I read with much interest the letter in the May 24 issue under the title "Oil as a Cause of Boiler Failure," and would like to make a few remarks concerning the subject. While not wishing to say that the failure of the flue in the Lancashire boiler in question was not caused by the presence of grease or oil, I do say that my experience with the use of oil in boilers, extending over a period of 12 years in the oilfields, has been that oil in boilers is *not* usually fatal.

The boilers used in the oilfields are of the portable locomotive type, and in the past 20 years thousands of these have been in use in northwestern Ohio and Indiana. Four-fifths of these boilers have had from one to two pails of crude oil (petroleum) pumped into them every 24 hours and in not one instance, to my knowledge, has the oil been the cause of any bad results; in fact, it is well known that the use of crude oil in the boilers effects a saving in fuel, and this is the main reason for its use.

During my entire experience I have known of only one boiler explosion in the oilfields, and in this the crown sheet gave in, killing one man and throwing the boiler some 50 or 60 feet. I am unable to say whether oil was used in this boiler or not, but have reason to believe that it was.

If any engineer knows, for a certainty, of any boiler failure caused *directly* by the presence of oil in the boiler, I would be pleased to hear of his experience.

D. A. Dickinson.

Lima, O.

Perpetual Motion

In the May 3 issue, Mr. Kirlin describes a so called perpetual-motion device. In subsequent issues several writers tried to prove that it will not work. Nearly all of them assume that it is in motion and then hunt around for "moments" and such truck to counteract an assumed force acting upward, losing sight of the fact that with the construction shown there can be no unbalanced upward force to begin with. An upward force acting on the exposed plungers does not act to turn the machine, but acts on the air within, and since air pressure acts equally in all directions it would force out all plungers alike, hence, the forces on all of the plungers are equal. So why look for a force to balance a force that does not exist? Again, the plunger falling into its cylinder on one side instead of not doing any available work, as one writer says, would on the contrary force out a plunger on the opposite side, and give as available work the buoyancy due to the transferred displacement. But the force acting on the second plunger would react on the first and force it back to its initial position. And there we are, where we started from, hence the falling plunger, so called, does not fall. The problem with perpetual-motion machines is not how to keep 'em going, but how in the world to stop 'em. Since, if they ever got started, the uniform force acting would increase the speed to such an extent that there would be no machine left.

Recognizing this, I have made what I term an accelerating motor. This device depends for its operation upon accelerated motion.

I am a first-class inventor but a poor draftsman, so will refer to Mr. Perras' sketch (June 14, page 1086) to illustrate my machine. He says that the weight of the floats descending on one side is helped by their buoyancy on the other side, or *vice versa.* If that is so, why not increase the weight to the maximum; that is, to the point where their weight is equal to their buoyancy? See the fallacy? That is the trouble with all perpetual-motion cranks, they want heavier weights, longer levers, bigger things and less friction, whereas, I use as little weight as possible and utilize the friction to hold the speed constant, as will be seen when we come to that part of it. In my machine, I use two tanks, the one in which the floats (Mr. Perras' sketch) ascend is filled with mercury, and in the other, the one in which they descend, is a variable air or gas pressure, occasionally equaling or exceeding the pressure caused by the mercury.

The floats are rubber spheres, hollow and filled with a light gas. Hydrogen or nitrous oxide is preferable. If we submerge in water a toy rubber balloon such as the children play with and let go of it, the balloon will rise to the top with an increasing velocity and will jump out of the water so that it will rise higher than the point from which it started, if we consider that it was first at rest on the surface.

When I brought my experimental machine to the attention of the boys at the lodge, all that I could hear for a while was, "It is against all principles; conservated energy; a pendulum will rise no higher than the point from which it fell;" and much more "dope" of the same order. When I showed them that the rubber ball would rise higher than the point from which it fell, and explained the philosophy of the thing, they did not have so much to say. You see it works like this:

The uniform force acting on the ball accelerates its motion; the accelerated motion raises the ball higher than the surface of the water, and this distance multiplied by the time and by the weight of the ball equals the surplus energy, which is plain enough if you can see it, although one of the boy's said that the friction of the balloon going through the water heated the air in the balloon, hence expanded it and gave an increased displacement, which goes to show that he does not understand the subject. By mechanics, the kinetic energy of a moving body is proportional to the weight times the square of the velocity. So it will be seen that weight does not have as much influence as velocity.

Again, "I proved by mathematics" that the final speed of the descending ball or balloon and the distance it rose above the surface was proportional to the square of the hight of the column of water or other liquid, but as the process is somewhat "complex" and hard for those not versed in higher mathematics to understand. "I will omit the proof." It can be seen that the acceleration can be made any amount desired by varying the hight of the column of fluid, and this together with the vacuum tank gives a means of regulating the speed to a hair.

If we concede the fact that the rubber floats can enter and leave a vacuum without gain or loss of work, it is true that, depending upon the degree of vacuum or the density of the medium within the tank, the direction in which the floats rotate will depend on whether the medium is lighter or heavier than the floats. Now, the mercury accelerates the floats upward and the vacuum adds to its effect to produce rotation. Then, acceleration gets in its fine work, and when the speed is brought up to the normal point, we can utilize the friction to expand the gas in the tank. This will create more pressure to retard the velocity of the floats and offset the acceleration under constant load. If the load changes, the regulator on the liquid tank will vary the hight of the liquid according to the requirements. In order to stop the motor we pump up the air pressure in the tank to equal the mercury pressure and there you are. With a little more pressure in the tank, the motor reverses. I believe that this is the first time that anyone has ever heard of one of these "something-for-nothing" machines reversing.

All that you have to do to start this motor is to let out gas from the tank until the required speed has been reached, when natural forces come into play and do the rest.

On account of the high price of rubber, although gas is cheap enough, I have not built a working model yet, but expect to do so when I can sell enough stock. You see, my experimental machine is so small that the friction interferes with its continuous operation, but I expect to overcome this in a large machine, because with more and bigger balloons there will be more gas, hence less weight and less friction.

Albert Ihlenfield.

Cleveland, O.

In the June 14 issue of Powdr there is a description of a perpetual-motion machine by F. M. Perras. He states that the machine will not work because of friction between the floats and the walls of the opening in the bottom of the vessel. The fact is that if the friction were not too great the machine would revolve in the opposite direction and the floats would be expelled at the bottom as long as the head of water was maintained in the vessel. The reason for this is that the force tending to expel the floats at the bottom is equal to a column of water having a diameter equal to the diameter of a float and a hight equal to the head of water in the vessel, plus the weight of the floats, etc., on that side.

The force tending to raise the floats in the water is only equal to the amount of water which they displace plus the weight of the floats, etc., on the opposite side. The machine is, therefore, a water-wheel of very poor design.

Thomas Henry.

West Toronto, Canada.

[A similar discussion was received from H. F. Liedtke, Philadelphia, Penn.]

It is said that William Crookes was very much surprised when he constructed his famous radiometer to find that the vanes rotated in a direction contrary to that in which he had expected them to rotate.

If F. M. Perras will construct his machine, illustrated in the June 14 number of Power, I venture to predict that if not too tightly fitted, it will run in spite of friction, but in the other direction. The buoyant effect of the water will not equal the downward pressure on the bottom ball, because the buoyant effect is equal to the weight of the water displaced by the balls. The downward pressure is equal to the weight of a column of water in cross-section equal to the cross-section of the balls at their center and in hight equal to the distance of the bottom ball below the surface of the water.

In other words, the downward pressure is the pressure on a round plug at the bottom of the tank, with diameter equal to the diameter of the balls, less the weight of the water displaced by the balls.

So the balls on the tank side would descend instead of rise and as they descended they would allow the tank to empty itself. Hence, power would be required to run the machine which is a kind of water motor instead of a perpetual-motion machine.

John A. Cook.

Bham, Ala.

Novel Feed Water Heater

In the June 7 issue, Mr. Schreiber criticizes my description of a feed-water heater, which was published in the issue of April 19. He has evidently forgotten that I have described an apparatus which has been in successful operation for a number of years, and that I have not merely made a suggestion for a new type.

I should like Mr. Schreiber to give one reason why a duplex pump would not work under the given conditions.

In regard to the oil in the exhaust, it is readily seen that only a small part of the exhaust steam is condensed and returned to the tank. Moreover, the feed water is taken from the lower part of the tank, which is always more than half full, and as oil floats on water but little oil finds its way to the boilers.

R. S. Seese.

Urbana, Ill.

Boiler Explosions

Peter Van Brock, I believe, is mistaken when he states that the chief cause of boiler explosion is low water.

I think that the great majority of boiler explosions are caused by other conditions than low water, such as scale and mud, bad management and, in particular, overpressure.

I was in one boiler explosion once. This one was caused by scale. I do not care to be in any more explosions. It is four years since the accident and I have been working around boilers every day since.

In the plant where the explosion occurred I had nothing to say about the cleaning or management of the boilers; but where I am now employed I have entire charge of the boiler management and certainly am careful that the boilers are kept clean and in good operating order.

John Berkley.

Homer, Ill.

Boiler Explosion in Kentucky

By O. Monnett

A decrepit boiler, an over-weighted safety valve and an ignorant fireman form the combination which was responsible for the recent explosion at Bowling Green, Kentucky. The circumstances surrounding this disaster point to the urgent need of proper laws for supervision and inspection.

If anything can be added to the literature of boiler explosions which will aid in showing just what operating conditions are in this country, and point a moral which, if observed, will result in benefit not only to the owners and operators themselves but also to the public in general, it is the duty of the investigator along this line to make the facts known.

Although in the technical magazine is the place where naturally such reports will be looked for, still if the inside facts could be laid before the public in general through the medium of the daily press, it would sooner arouse public interest and a general realization of the extreme carelessness with which steam boilers are operated in all parts of the country.

There is no doubt but what we are generations behind European practice in this regard. As a matter of fact we are the laughing stock of the world, were it a laughing matter, when it comes to the supervision of steam boilers. Last winter some of our greatest savants got together and seriously discussed the question whether cast iron was suitable for superheated steam. It was definitely settled in England thirty years ago that cast iron was absolutely unsuitable for this service. Yet the question is just beginning to be serious in this country. This shows how we are lagging behind in some regards, and in the matter of boiler inspection and supervision our methods are probably in worse shape than in any other branch of engineering.

Knowing that there had been a boiler explosion some time previous in a sawmill at Bowling Green, Ky., and happening to be in the neighborhood, it was decided to sojourn in that city over Sunday and make an effort to pick up something that would add to the store of knowledge of this subject and perhaps lend its influence toward the ultimate correction of some of the operating abuses practised in the rural boiler plant.

It was found that the boiler was an ancient one, containing forty-six 3-inch tubes and constructed of innumerable small sheets. The longitudinal seams were all double riveted, but the ruptures occurred seemingly without much regard for the seams, tearing through the middle of the sheets in all directions. The boiler had been in charge of a young inexperienced fireman, and it is asserted that the lever-and-weight safety valve was tied down or heavily overweighted. The story goes that a visitor to the plant came up to the boiler and noticed that the gage indicated 165 pounds pressure. Upon remonstrating with the young fireman, and calling his attention to the danger of blowing up the mill, he replied, "I don't care. Let her go." Upon hearing this the visitor rushed to the owner, who was working nearby, telling him of the danger. He had hardly delivered his message when the explosion occurred, killing one workman and injuring another so seriously that he died later. Whether or not the safety valve was really tied down, or whether an overpressure was on the boiler at the time it is impossible to vouch for. In view of the absolutely disreputable condition of the metal as found it would hardly seem necessary to assume much overpressure in order to account for the explosion. Moreover, if the steam gage was of the same vintage as the boiler, no reliance could be placed upon it as a pressure recorder, and it seems impossible to believe that the boiler could have stood the above mentioned pressure long enough to register on the gage.

What Was Left After the Explosion

Be that as it may, the boiler was blown completely to pieces, the failures as above pointed out being principally in the sheets. When a boiler is built of good material the seams are always the weakest points, there being no such thing as a 100 per cent. joint in boiler work at present. But as time goes on the plates are subjected to external and internal corrosion, pitting, grooving, breathing due to varying steam pressure, and other actions which tend to deteriorate the metal, so it frequently occurs that the joints become the strongest part of the boiler. Whenever the failure occurs in the plates, leaving the majority of the seams intact, it indicates that enormous deterioration has taken place, and is always seen where very old boilers explode. The front head was blown away from the rest of the boiler, the plates tearing off half way between the end of the sheet and the first curvilinear seam, indicating that this seam, although single riveted and originally of only 56 per cent. efficiency, was much stronger than the sheet itself. The boiler was liberally patched on the bottom but in no case, as far as could be observed, had the patches let go. A patch on any boiler most certainly weakens the structure, and when in a boiler explosion it is found that the patches are intact, it further indicates the bad condition of the original metal of the boiler. It was practically impossible to find a spot on the sheets where the approximate thickness of the original metal could be determined. On the smoke-box extension the metal had wasted away to about $\frac{1}{8}$ inch in thickness. Laminations were apparent in the fractures at nearly all points.

It is understood that several suits have grown out of the accident for the recovery of damages from the company, but the latter holds that the accident was entirely the fault of the inexperienced fireman.

Whose fault it really was that such conditions should be allowed to exist is not so difficult to determine. A combination of decrepit boiler, unskilled fireman and an overweighted safety valve is certainly one that is hard to beat, and is something that can be laid directly at the door of the public itself. The inevitable result of such a combination is seen in the accompanying illustration.

There is no law in Kentucky requiring an owner to have his boiler inspected. There is no law requiring an owner to hire a competent man to operate his boiler. There is nothing to prevent a man who is totally ignorant of boiler operation from accepting a job as fireman in a boiler plant; in fact, he is the man who will most likely get the job, as he will work cheaper than anybody else. Until laws covering these subjects are passed, not only in Kentucky but elsewhere, we will have repetitions of these criminal conditions and it will be nobody's but the public's fault.

Accounts of boiler explosions have appeared and will continue to appear in the technical press for a long time to come and will be read by a class of men the great majority of whom appreciate the importance of the question, but until the public mind is aroused to the deadly possibilities of the old boiler and the careless operator, nothing much will be accomplished. If it could become a matter of common knowledge that steam boilers, stuck away in obscure places, are as dangerous as a dynamite factory, then some politician who might know no more about a steam boiler than about a soup kettle would be glad to take up the question as a political issue and we would get something done. Politicians will not take up this question until the public is educated to the importance of voting for it, however, and thus it will, no doubt, be many years before we see satisfactory legislation along the lines so much to be desired.

In investigating boiler explosions, particularly in the rural districts, one is impressed with the firm hold which the low-water theory has upon steam users. Low water is the cause invariably assigned for a boiler explosion by this class of users. There is an impression that a boiler will be perfectly safe forever if there is plenty of water kept in it when steaming. Second-hand boilers are in great demand among such people. The probable age of a boiler as effecting its safety does not seem to enter into their calculations.

Before the wreck of the Chewtown (Penn.) boiler explosion had been cleared away the management was busy planning where it could get some more "good" second-hand boilers to replace those damaged in the explosion. Similarly at Bowling Green another "good" second-hand boiler was scraped up from some place and the work of the mill went merrily on. It is needless to say that the firemen who attend these "new" second-hand boilers will have special instructions about keeping the water level correct at all costs, with the result that if they let go they will at least release enough energy to mar seriously the surrounding landscape.

As a matter of fact, a boiler begins to deteriorate from the time it is built. The forces of oxidation attack the metal unceasingly. Any boiler, however perfectly built or managed, will explode *if kept in service long enough,* merely from the deterioration of the plates. It remains for expert inspection and supervision to determine how far this deterioration has gone and be governed accordingly, before the boiler has an opportunity to demonstrate what an enormous amount of energy is stored within it.

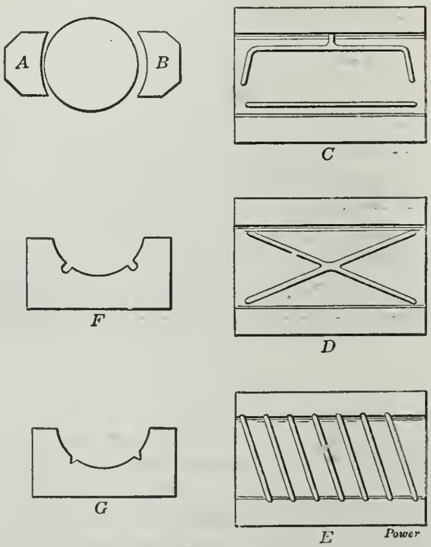
Oil Grooves in Bearings

By C. R. McGahey

The failure of main bearings to run properly after having been newly babbitted is invariably due to improper workmanship. Simply lining up and pouring the bearing does not mean that it will be a finished piece of work, nor can we depend upon the bearing to run cool; but we can save ourselves a great deal of work in the way of hand scraping by properly arranging the journal for the reception of the babbitt. Many bearings made of babbitt will give much better service from the start without the aid of hand scrapers, especially when the scraping is done with no object in view other than simply going over the surface, as we see it done.

The fact that the journal will dissipate heat in proportion to the amount of surface in direct contact, would point to the desirability of a perfect contact, yet this contact must have a certain relief so as to allow the journal to run cool. For instance, the box *A* would tend to run cool while *B* would run hot. All bearings expand when running and if we make no provision for this, we will have trouble. The form shown at *A* could expand and still have ample clearance as the babbitted portion has a slightly larger curvature than that of the shaft. It is the oil entering between the two surfaces that reduces the friction, therefore a bearing that is made to allow a free entrance of oil will run cooler than one not so made.

The next important point is the oil grooves. I have noted with interest the remarks of some about the uselessness of oil grooves, but will say from practical experience that I have taken bearings with no oil grooves which were giving some trouble and by simply putting some good grooves in the surface, no further trouble was experienced. I am a strong advocate of good, clean, deep oil grooves. Sketches *C*, *D* and *E* illustrate three types of oil grooves but the style of groove will depend upon the bearing to be lubricated. The one shown at *C* is a good form for large bearings while that shown at *D* is a common type. Either will work well, if large enough and having the proper depth, so that it will hold a sufficient amount of oil. The cutting of the groove is shown at *F* and *G*. That shown at *F* is a common form which is cut with a small chisel; it is seldom cut deep, but will hold the lubricant. That shown at *G* makes a groove which is easily cleaned, but the grooves should be deep enough to carry plenty of oil.



Types of Oil Grooves and Proper Radius for Bearings

One of the better forms of oil grooves is the spiral groove. This is shown at *E* and can be formed by simply wrapping a coil around the bearing to be poured. Place a good piece of manila paper around the journal and wrap a piece of cord around it. This will take up the space closed in by contraction and the cord will form the oil groove. This is especially good on small bearings and saves time and scraping. The original surface of the metal without being scraped is much better as a bearing surface than the one presented after scraping; however, on large bearings it is hard to get a surface that will not need scraping and fitting after being poured.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T McKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for June, 1910, *was* 139,000.

July 5................ 34,500
July 12................ 34,500
July 19................ 34,500

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

The Education of the Operative Engineer

Power-plant engineering now offers opportunities for the application of learning of a high and specialized sort. Men with this sort of learning acquired at technical schools are taking executive and supervisory positions and making mere mechanicians and underlings out of the men with less educational attainments. The technical man frequently makes mistakes for lack of the practical man's training; the practical man is often not broad enough and big enough and well educated enough to be the responsible head of the department. If these two men could be rolled into one he would be an ideal man for the job.

In the contest for supremacy the technically-educated man has all the advantage in position, in connections, in prestige and usually in presentability, ability to make a favorable impression and to present his case; and it is incomparably easier for him with his knowledge of principles to acquire what the ideal man needs of practical skill than it is for the practical man to acquire what the ideal man needs of education.

If the big positions are to be filled by those who have come up from the ranks these must be given the means for acquiring the education which will round them out to the requirements of the situation.

Many a man has done this on his own hook. If he had not, many a big job would be looking for a man today. He could have done it easier and more effectively if there had been a course laid out for him to follow, regular terms and grades as in an established school, recognition when one grade had been completed and he passed into another, and the incentive ahead of him all the time of ultimate recognition as a master in his art, such as the technical student has in his diploma.

The difficulty which the National Association of Stationary Engineers has encountered in its educational work is the diversified condition of its membership. You cannot teach algebra to a man who does not know his arithmetic, and while all of them can enjoy and get something out of lectures, etc., it is impracticable through delicacy about admitting elementary deficiencies, difficulty and expense of procuring instructors in various grades, to make the association into a systematic organized school with a regular course of study and promotions.

Suppose a young man who is about to start as an engineer could enter such an institute or guild as is contemplated in the Institute of Operating Engineers, as a junior apprentice, could be told what it was necessary for him to study and what experience it was necessary for him to have to pass to the grade of senior apprentice, could be advised what books to study if he must study alone, what courses to select if he wishes to study by correspondence, or told what classes had been organized or arranged for either among Institute members of his own grade or in public or endowed institutions of learning in his neighborhood. This course would be within the reach of his present attainments, would be such that a man of ordinary ability and a limited education could pick it up and follow it. One part of it would lead naturally into another, and all of it be applicable to the practical work which he was following or of that general educational character necessary to make a man as well as an engineer of him. He would naturally settle down to getting through that line of work in the two years allotted to it. He would have a definite program to follow and a definite end in view, and if at the end of the two years he had qualified himself as far as both practical experience and study were concerned he could pass to the grade of senior apprentice and would have won his first success in his progress toward his object of being a master operating engineer.

If he does not stick to his job and make the most of its advantages in learning the practical end of the business, i.e., if he is unsatisfactory as an apprentice workman, if he does not follow the line of study laid out, he will remain a junior apprentice. His status in the Institute and in the vocation will depend upon his success in following the prescribed course and there would be every incentive for him to do so. It would be like a school or college, and a man would naturally go on or get out, not remain perpetually in one class.

With the growth of the Institute he would be likely to be apprenticed to one of its master members who would give to him the interested attention which he in turn would give to his apprentices when he became a master himself.

The Use of Superheated Steam

The use of superheated steam in engines and turbines has been such that its behavior can now be predicted with more or less certainty. At the present writing, no engineer can successfully plead ignorance as an argument for or against its use. It is now pretty clearly understood that the economy resulting from the use of superheat is due to the prevention of condensation both in steam pipe and cylinder, rather than to any great increase of efficiency in the steam itself. Nothing succeeds like success—that is, nothing is so good a preventive of cylinder condensation as steam which is initially dry or slightly superheated. The exact amount of superheat necessary to prevent condensation varies with circumstances but it is probably from 5 to 10 degrees of superheat for each one per cent. of moisture present at cutoff.

Current practice favors the use of an inclosed superheater connected to the boiler, rather than the independent type. Where a superheater is placed in the passageway of hot gases without interfering with the natural heating surface of the boiler, there will be no loss of efficiency in the latter; in fact, there will be sometimes a slight gain.

Experiments both on stationary and locomotive engines show a possible economy in steam consumption of from 15 to 20 per cent. and a net heat saving of from 10 to 15 per cent. with a comparatively low superheat of from 100 to 150 degrees.

The engine economy is most favorably affected in those engines which are particularly subject to initial condensation and wet steam on account of low speed, poor proportions, etc. Locomotive engines are particulalry susceptible to improvement in this respect because of the large amount of condensation which is ordinarily present when saturated steam is used.

Experiments made thus far show a continual improvement in the economy of locomotives as the amount of superheat is increased up to 180 degrees. The maximum would probably be higher for locomotives than for stationary engines, due to the causes mentioned. With a superheat of from 50 to 150 degrees at the throttle, little trouble will be experienced from the increase in temperature. Additional care must be used in lubricating the valves and pistons of the engine and a high grade of cylinder oil must be used.

Troubles will be experienced with the pumps attached to the steam line unless special attention is paid to the packing and the lubrication.

When superheat of a greater degree is used, there will be trouble, and plenty of it, both in the boiler and engine room, unless the piping and machines have been especially designed for high temperatures. Cast-iron fittings become more or less unreliable at temperatures of over 500 degrees Fahrenheit and some grades of iron are brittle and dangerous at these high temperatures. Ordinary gaskets leak and blow out. The packings of the pumps rapidly deterioriate and there is trouble with the lubrication on both engine and auxiliaries; serious cutting of slide valves frequently results, and metallic packings become necessary on the piston rod. If there is much fluctuation of temperature, great trouble is experienced with the piping on account of uneven expansion and fittings are sprung and sometimes broken.

Our advice to anyone who contemplates using high degrees of superheat in an old plant is most emphatically "Don't," because the disadvantages will more than offset the gain in economy. Like most panaceas, superheat must be used with discretion and is not a cure for all the evils in the calendar. It is doubtful if temperatures of over 600 degrees are advisable in any plant but when a new installation is made with piping, fittings and gaskets designed for high pressures and temperatures, the result is much more satisfactory than when an attempt is made to graft new ideas onto old equipment. Remember that a moderate superheat, 50 to 100 degrees, can be employed without serious difficulty and will pay a better return in proportion to the trouble than higher degrees, and that the high temperatures are only desirable when the whole equipment is designed especially for their use.

The Conservation of Water Powers

The people have awakened to the fact that this is an age of power, that electrical distribution and the multiplying uses of electricity have increased the value of every water power many fold, and that if they sit quietly by and see them appropriated as the coal and oil have been appropriated they will pay in perpetuity a baron's toll for any of that power which they, however indirectly, consume. They have therefore demanded that the unrestrained relinquishment of the rights of the people in their inheritance be stopped, and that power sites be developed by the people for the benefit of the people, or by groups of the people incorporated for the purpose under such restrictions that the corporate group may impose upon the rest of the people such a charge as will render a fair profit upon the money which they have actually invested, and no more. It makes no difference whether the sites are developed by the Federal Government, the State government or by private capital so long as the charge put upon their use is sufficient only to pay the interest and sinking-fund charges and actual cost of maintenance and operation, plus the fair profit to private capital if private rather than Governmental enterprise and means are invoked. Neither government would any more than get its money back, but the State where the water powers are located would profit in cheap power for their inhabitants and by the influx of industries to take advantage of that cheap power.

In view of this simple and self-evident condition one wonders through whose spectacles the Honorable William E. Borah, United States senator from Idaho, was looking when he conceived the following utterance, delivered when the subject was under debate in the senate:

"These power sites are our wealth. We have the means to control them and to dedicate them to the use of the people. They are a part of the State's heritage. It is a violation of every principle of the Constitution to withhold them from our use. If you tax them for the Federal Treasury will not our people have to pay the tax? Do the people of the older States pay any such tax? If Illinois or Massachusetts receives thousands or even millions of dollars as a revenue for their water power and Idaho receives nothing, but, on the other hand, pays thousands or millions into the Federal Treasury, is there equality among the States? The old States make these power sites a source of incalculable revenue out of which to pay the expenses of the State and lessen the general taxes. But we must not only pay our taxes, but, in addition, pay this revenue to the general Government."

The water powers of Massachusetts are owned by private interests. If the State gets "millions of dollars" from them in revenue, the millions of dollars come out of the ultimate consumers of the products of the water powers and not out of the corporations which own them. If the unappropriated and undeveloped water powers of Idaho and the rest of the country are developed by the interests unrestrained by their retention in them of their rights and the power to fix rates by the people, the people will pay the cost of their surrender, as they pay now in every coal bill and in the price of every article into the cost of which coal-made power enters the surrender of their inherent rights in the country's mines.

The comet is a "has been" just now, and won't be heard of again for 75 years; so is the slovenly, lazy man of the engine room, but he will never be heard of.

It has been estimated that the strength of the average horse is that of five men. Few engineers can back up the assertion that they feel as strong as a horse.

Don't block the lever of a safety valve against the roof to prevent the escape of steam. It isn't worth it.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Steam-engine Efficiency

A steam plant shows a boiler efficiency of 62 per cent. using 1100 pounds of coal of 10,000 B.t.u. value per hour. If the engine indicates 250 horsepower, what is the efficiency of the engine?

F. A. H.

The heat contained in the coal is 10,000 times 1100 or 11,000,000 B.t.u. At 62 per cent. efficiency of the boiler plant, this is 6,820,000 B.t.u. delivered by the engine. A horsepower is equivalent to 33,000 foot-pounds of work every minute or 1,980,000 per hour.

A heat unit is equivalent to 778 foot-pounds. Dividing 1,980,000 by 778, gives 2545 heat units per hour as the equivalent of one horsepower. Multiplying this by 250, the horsepower of the engine, gives 636,500 thermal units as the equivalent of the energy furnished each hour by the engine. Dividing by 6,820,000, the heat furnished to the engine, gives 0.0933 or 9.33 per cent. as the efficiency of the engine.

Piston Speed of Single-acting Engine

What is meant by single-acting engine, and how is the piston speed found?

J. H. R.

In a single-acting engine the steam acts on one side of the piston only as in the Brotherhood and Westinghouse types. The piston speed in feet per minute is found by multiplying the stroke in feet by the number of revolutions per minute, or by multiplying the stroke in inches by the number of revolutions per minute and dividing the product by 12.

Valve Setting

How will I set the valves on a Greene, a Corliss and a Brown engine?

J. C. D.

In a general way the valves of all engines are set by the same method, and general knowledge of the fundamental principles will enable you to set the valves of any engine. Adjust or make the eccentric rods and valve stems the right length. Then with the engine on the center turn the eccentric in the direction that the engine is to run until you have the proper lead at the right end of the cylinder.

Direct Side of Pump

Which is the direct side of a duplex pump?

F. E. D.

It is that side on which the rocker moves the valve in the same direction that the piston travels.

Wavy Lines in Diagram

What makes the wavy lines in a high-speed indicator diagram?

J. D. D.

A variety of causes contribute, among which are too little tension in the drum spring, too light a spring on the indicator piston, too much travel for the drum, too heavy take up and hook on the drum cord making it "whip." If smooth lines with curves which are even and probably truthful are wanted in high-speed engine diagrams, they should be short, and stiff springs should be used. For 300 revolutions per minute the diagram should not be over 2½ inches long nor more than 1¼ inches high.

Horsepower of Belting

I am taking power from a shaft driven by a six-inch belt running over a 30-inch pulley, making 165 revolutions a minute, and am paying for 15 horsepower. This seems to be too much and I would like to know if it is possible for a six-inch belt to transmit so much power?

J. F. L.

At the speed the belt is running, nine horsepower is a fair load and 15 horsepower is an overload of about 80 per cent. It is possible to transmit a much greater horsepower than the accepted rules allow, because they are not calculated for the ultimate strength of the belt but for what it will carry with safety for a number of years.

Frequency of Lighting Current

At how low a frequency can incandescent lamps be supplied with alternating current without causing inconvenience to the users?

L. J. G.

Investigations made several years ago with carbon-filament lamps indicated that 30 cycles was the lowest frequency possible without discomfort to the lamp user. With tungsten lamps it is probable that a considerably higher frequency would be required to avoid discomfort.

National Association of Stationary Engineers

How old is the National Association of Stationary Engineers, and what are its aims?

E. F. W.

It was organized in 1882. Its aims are:

The education of its members in the art and science of steam engineering.

To protect the interests of competent engineers in their vocation.

To enroll all competent engineers.

To impart information beneficial to the calling.

To assist members to obtain employment and to procure by law greater safety in the operation of steam plants.

Long Suction Pipe

What precautions should be taken in laying out a long suction pipe for a pump?

V. H. N.

All joints should be air tight and the pipe should have a gradual rise without depression at any part as any unevenness in the grade will be liable to form air pockets which will prevent the satisfactory working of the pump.

The Heat Unit

What is a B.t.u.?

R. B.

"B.t.u." are the initials of the words British thermal unit. One British thermal unit is the quantity of heat necessary to raise the temperature of water one degree Fahrenheit, from the temperature of maximum density—about 39 degrees Fahrenheit.

Terminal Pressure

How is the terminal pressure found in an indicator diagram?

A. C. W.

If the expansion line is prolonged, guided by the eye, to the end of the diagram as though the exhaust valve had not opened, its distance from the atmospheric line measured in the scale of the indicator spring will be the terminal pressure.

Position of Knife-blade Switches

How should a knife-blade switch be set on a wall or panel with regard to the opening and closing of the blades?

R. B.

The switch must be mounted so that the blades cannot close by their own weight. The best position is with the handle pointing straight upward when the switch is closed.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

The Ajax Four Valve Engine

This engine is built by the Hewes & Phillips Iron Works, Newark, N. J., which firm has entered the field as builders of Corliss, four-valve, automatic, high-speed engines, fitted with inertia shaft governors. The "Ajax" engine is made in simple, tandem-compound, cross-compound, condensing and noncondensing types, in sizes ranging from 100 to 600 horsepower. In Figs. 1 and 2 are shown a front and a back view of the engine.

The builders have recently purchased the shop rights to manufacture and use the Robb-Armstrong-Sweet inertia shaft governor, and all of their engines of the automatic, high-speed, four-valve type are equipped with this device.

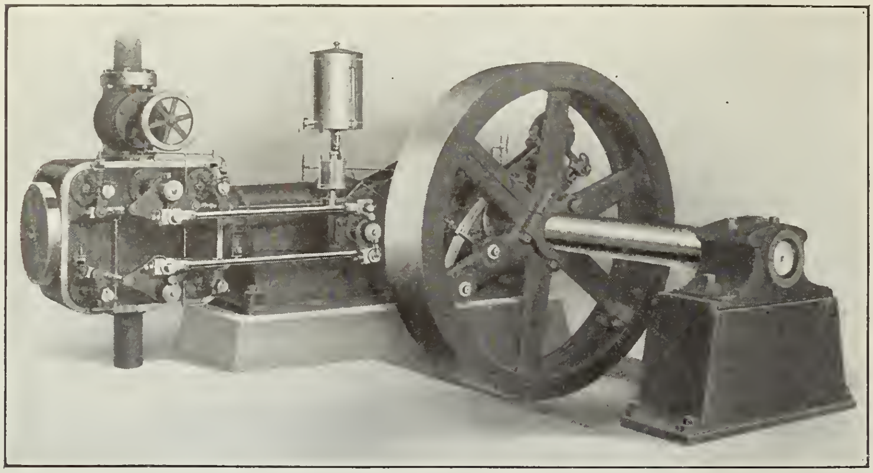


Fig. 1. Valve Gear of Ajax Four-valve Engine

Designed for 125 pounds working pressure, the cylinder is made with the exhaust chamber separated by an air space from the cylinder barrel. The two steam and two exhaust valves have straight passages through them, as shown in the cross-sectional view, Fig. 3. The valves have an equal disposition of metal on each side of the vertical center line and the outer members of the valve are tied at the center with a sufficient number of bridges to meet the requirements of the work. The valves rest on their natural seats, steam pressure, gravity and vacuum acting along the same vertical lines. Attached to one side of the cylinder are two steam- and two exhaust-valve brackets operated from independent swing plates attached to separate eccentrics. The exhaust eccentric is fixed in its throw, but can be varied as to the angular position with reference to the crank. The steam eccentric is of variable throw and comprises a part of the governor.

The arms attached to the valve stems are connected to two swing plates which produce a long dwell when the valves are open and a corresponding condition when they are shut, but at the moment of opening and the moment of closing, the speeds of the valves are most rapid. Neither the steam nor the exhaust valves are ever entirely at rest except at their moment of reversal. This condition prevents any tendency for the surfaces to seize. Fig. 4 shows diagrams of the valve movement. The steam valves are practically balanced when they are moving toward the admission or closing points, producing a minimum of strain on the governor and valve connections.

The frame and guides are of a new pattern of the heavy-duty tangye type in which ample provision is made against the discharge of oil either by splashing or creeping down the surfaces of the casting. A glance at the illustrations shows the general arrangement of the parts and the particular disposition of the material. In general the bed is of rectangular box form, having flowing base lines which have a grooved form extending entirely around the lower edge of the casting, giving complete security against oil reaching the foundation.

The main pillow block is made in four pieces, the top, bottom and quarter boxes being lined with babbitt metal hammered into place and fitted to the shaft by boring and scraping. The quarter boxes are backed by heavy steel plates planed true. These plates are adjusted by set screws. The lower section of the bearing is made hollow and provision is made for water circulation through it, permitting control of bearing temperatures under unusual circumstances.

Gun-metal cast iron is used in the crosshead, which has two adjustable shoes fitted to the guide. When adjustment is required the hexagon nut at the side of the crosshead is loosened, then the nut at the rear of the shoe is unlocked and a slight turn of the steel spool adjacent to it, moves the shoe forward or backward, tightening or slacking the running fit of the crosshead.

The steel crank shaft is made with a

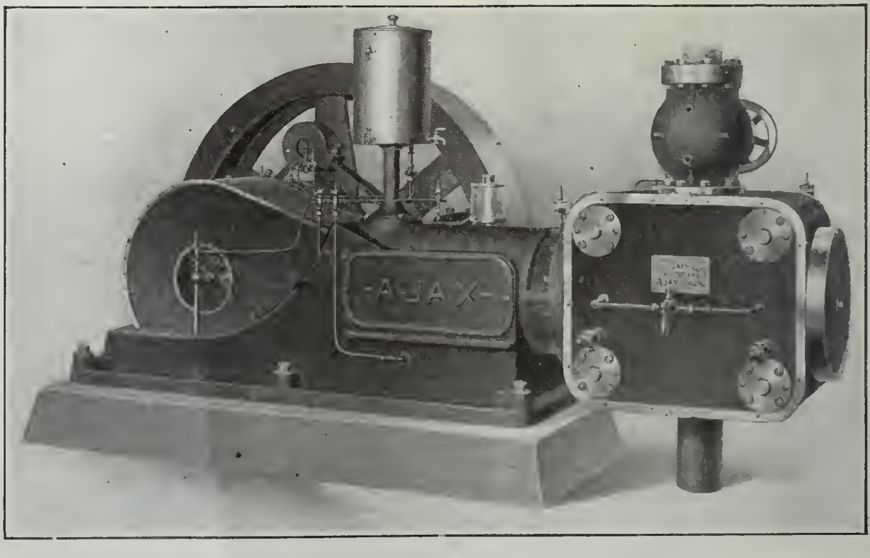


Fig. 2. Crank Side of Ajax Four-valve Engine

swell in the center where the flywheel and generator are located. Long tapers or fillets are used wherever the shaft changes its diameter.

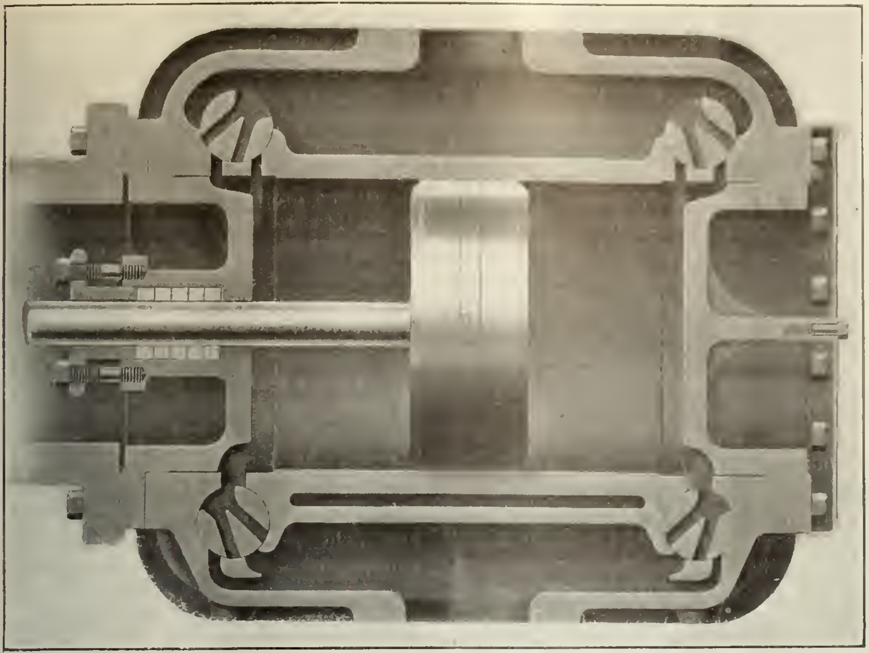


Fig. 3. Sectional View of Cylinder

A crank of the counterbalanced type, heavily proportioned is pressed onto the shaft and keyed in place. The crank pin is forced into the crank and the end riveted over.

The flywheel is heavily proportioned and where furnished for alternating-current generators, it is made of sufficient weight to secure proper regulation when generators are run in parallel. The wheel is designed with one set of oval arms and is so proportioned that the strains arising from centrifugal force will be far below the elastic limit of the material employed.

A steel connecting rod, made with solid ends, is provided with wedge adjustment for taking up wear. The wedges are so disposed as to keep the length of the rod constant. The ends of the rod are fitted with bronze boxes bored and scraped to fit the pin. Straight and curved arrows indicate the direction for adjustment.

A bull-ring type of piston is used and is so arranged that it can be adjusted with set screws. One center packing ring in sections is provided and it is cut in four or more parts, having lap joints provided at the under side of the joints with brass clips set out by spiral springs. The follower is scraped to the spider and the rings are accurately fitted into the bull ring. The bull ring has a bearing the entire width of the piston.

All of the important bearings of the engine are oiled by a system of piping. The bearings are provided with sight-feed adjustment. The oiling system consists of a tank superimposed on the engine bed and supplied by a mechanically operated oil pump, attached to a filter tank suitably located in the sub-base under the tangye frame. Means for separating sediment from the oil are provided in the tank. The pump has a capacity far in excess of the engine requirements. The overflow from the tank on the engine bed discharges automatically into the filter tank.

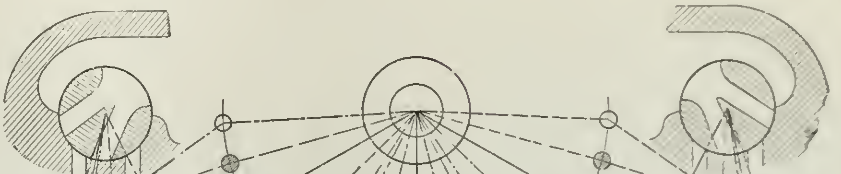


Fig. 4. Diagrams of Valve Movement

Hauck Kerosene Torch

The accompanying illustration shows the general appearance of this torch. It is simple in design, and strongly built. It is adaptable for various heating operations, light brazing and tinning. It produces an intense clear flame.

Hauck Kerosene Torch

The burner is designed to use kerosene oil, and the torch is so constructed that the tank will remain cool, while the burner is being operated. Only one pound pressure is required on the torch.

This torch is made by the Hauck Manufacturing Company, 140 Cedar street, New York City.

Anti-Smoke Convention, Minneapolis

The International Association for the Prevention of Smoke convened at Minneapolis, June 29 and 30 and July 1, for the consideration of matters of interest to those responsible for the supression of the smoke nuisance in our cities. Papers on the program for this, the fifth annual session, included "The Harrington Automatic Stoker," by Joseph Harrington, chief engineer of the Green Engineering Company, of Chicago. Mr. Harrington's paper took up the subject of the installation of stokers of less than 200 horsepower, in which the disproportionate cost of the stoker as compared to the rest of the plant ordinarily offers a serious objection. Attention was called to the fact that the average small boiler is set too low for regular stoker installation and generally there is not available the room for ashpits and satisfactory combustion chambers. In view of the large field for a stoker which could be easily applied to small boilers already installed the question had been taken up with the idea of producing a stoker with low grate surface, small grate area, simple and cheap, which could be used under the above conditions. The paper described the design of stoker which had been brought out to meet these conditions. The stoker consists essentially of a self-contained structural steel frame, practically independent of the brickwork, and provided with movable grate bars inclined at an angle, upon which the coal is pushed from a dial plate, by means of the driving mechanism. The stoker also has an arrangement whereby the ash is continually removed to the front of the boiler as fast as formed. A complete description of the device will be given later.

C. F. Hodges, of the B. F. Sturtevant Company, Hyde Park, Mass., presented an illustrated paper on "Forced and Induced Draft and its Relation to Combustion." Typical installations were shown and the peculiarities of the two types were brought out in the lantern slides.

"Taylor Stokers and Steam Boiler Efficiency," was the subject of a paper by R. S. Riley, of the American Ship Windlass Company, Providence, R. I. Mr. Riley explained that smokelessness and efficiency did not always go together, and on the other hand, it is quite possible to make considerable smoke and yet realize satisfactory efficiency. From the point of view of the operatives, he said that efficiency was the main object, although if smokelessness could also be obtained it was very desirable. Excess air, it was pointed out, was the great enemy of efficiency and care in keeping this item down to the necessary requirements was given as one of the principal considerations when both efficiency and smokelessness were to be realized. Mr. Riley's paper contained considerable data on boiler and furnace efficiency as obtained with Taylor stokers in practice and contained a detailed description of the stoker itself, which has been fully treated in POWER AND THE ENGINEER.

S. S. Howell, of the Under-Feed Stoker Company of America, Chicago, read a paper on "Furnace Draft and its Relation to Smoke Prevention." The paper emphasized the fact that a strong draft is necessary in getting the air through the fuel, overcoming the resistance of the boiler and breeching, and getting the gases up the stack. With bituminous coal distilling and breaking up into its constituents it is essential that the air permeate the finely divided particles, for the purpose of perfect combustion. The evolution of gas, it was shown, was very rapid, and therefore the air should be supplied with considerable velocity; the resulting forcible impingement of the air and fuel being depended on to accomplish the necessary mixing, this being the critical point in the process. Proceeding with the paper, Mr. Howell took up in detail the calculations for stack, breeching, combustion areas, grate surface and all other items entering into this phase of steam-plant design. The result of the analysis showed that, using the ordinary formula, stacks are frequently built too small in diameter, are lacking in hight, and that sufficient attention is not always given to details of the design of the gas passages to get the best results from the standpoint of economy and smoke.

One of the most interesting papers was that by Prof. R. H. Fernald, engineer-in-charge United States Geological Survey, Case School of Applied Science, Cleveland, O. The paper was entitled "Gas Producers as Related to the Smoke Problem." It pointed out that with the present rate of using coal the visible supply would be exhausted by the middle of the next century, and that therefore it was necessary to begin to use the lower grades of coal, lignites, etc., wherever possible. Experiments at St. Louis and Norfolk were cited to show that 2½ times more power could be developed with producers than with steam plants of ordinary design, using the same amount of coal, and that frequently the low-grade varieties gave as good results in gas engines as the more expensive coals.

It was shown that the use of producers meant the elimination of the smoke nuisance and that central plants for the distribution of gas for power purposes were perfectly feasible as one solution of this problem. The different types of producers on the market were illustrated and a large amount of data on the coal supply of different states and their characteristics when used in producers was given, together with records of test runs by the Government, which have been issued from time to time in bulletin form.

Health Commissioner Hall, of Minneapolis, in speaking of the administration problems in regard to smoke suppression, gave his views as to the detriment to health and property from this source, and hoped for the time when electrification or centralization of power plants outside of the city limits, would solve this difficulty.

One of the sessions of the convention was devoted to railroad smoke, representatives of the Illinois Central, the Chicago Northwestern and the Chicago, Milwaukee & St. Paul railroads being present. It developed that the Illinois Central might be characterized as a "Steam Jet" road, especial attention being paid to this method of smoke suppression. The steam jets are introduced in the firebox about twenty inches above the grates and work in conjunction with a blower in the stack which is automatically turned on when the throttle is closed.

On the other hand, the Milwaukee relies mostly on the brick-arch system of reducing the smoke, every engine being equipped with this device. The Northwestern, it was brought out, relies on a combination of the brick arch and steam jet, inclining to the latter method. It developed that according to the opinion of the smoke inspectors present, from 30 to 50 per cent. of the total smoke in cities was caused by the railroads, the percentage varying from the lower figure in the winter, to the larger one in the summer time. It was unanimously decided that electrification was the ultimate solution to this problem.

The next convention will be held at Newark, N. J., the time being the last week in June, 1911. Officers for the ensuing year were elected as follows: Paul P. Bird, of Chicago, president; Daniel Maloney, of Newark, vice-president, and R. C. Harris, City Hall, Toronto, Can., secretary-treasurer, the latter being reelected.

The hospitality of the city was extended to the visitors at all times and special arrangements were made for entertaining the ladies. Of the social features, an informal dinner at the Commercial Club, and an excursion to Lake Minnetonka, were the leaders, both being most enjoyable occasions.

There should be a society for the prevention of cruelty to machines as well as to animals. Did you ever see a brutal operator pound a nice bright finished part of his machine?

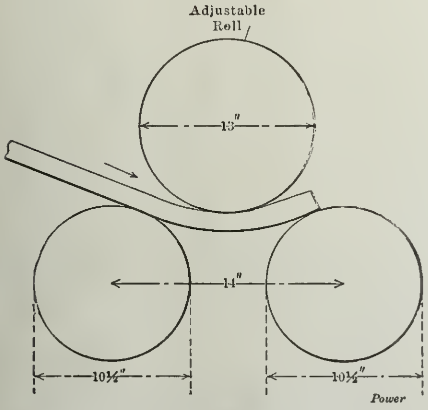
Probable Cause of Recent Explosions

By T. T. Parker

A significant fact connected with a number of recent boiler explosions is that in each case the boiler was constructed with one sheet on the bottom, extending from head to head. The defects of this type of construction are pointed out and its discontinuance recommended.

Reviewing the Canton boiler explosion as described in the May 31 issue of Power, we may accept as conclusive that the boiler failed by the shearing of the rivets at both longitudinal seams, which were of lap-joint and double-riveted construction. The shell was built in two sheets each extending from head to head, the joints or seams being on each side below the lugs and, of course, below the upper row of tubes so that each longitudinal seam was inaccessible for inspection either externally or internally.

The feature of special note is that this is the third explosion within a few months of boilers built with one sheet on the bottom extending from head to head. The boiler that exploded at the Robinson clay-product plant, Midvale, O., in December, was built in the same manner and the boiler that exploded about the same time at Shelton, Conn., was built with one sheet on the bottom. In 1902 a boiler built in this manner exploded at the Swift plant in Chicago, killing 13 men and injuring several others. Also in 1898 or 1899, another similar boiler exploded at Willy's mill, Appleton, and resulted in heavy property damage and the killing of one man. The initial fracture of the latter took place at the long seam, while the Swift boiler failed on top in the solid plate from head to head. Probably other explosions of boilers built in this manner have occurred that the writer has no knowledge of. It is the purpose of this discussion to point out the defects in this type of construction and the bearing such matter would have upon boilers of similar construction now in operation, with the hope of pointing out reasons that may explain such explosions with a view of possibly preventing future explosions. The publicity given to boiler explosions, together with the vast number of intelligent engineers and superintendents who read and assimilate the articles in Power, prompt me to place my views before them, knowing that if I am in error as respects the conclusions, the mistakes will be pointed out. If, otherwise, perhaps, an explosion will be averted.



Sheet Passing through Rolls

In the construction of a cylinder the plates composing the shell must be cold rolled into circular shape. Consider the ordinary boiler-shop rolls used for this purpose. The roll shown in the sketch is a common type; the upper roll being the adjustable one. The flat boiler plate enters at one side and is passed along under the upper roll until it strikes the second lower roll. It is evident that the plate is not curved from a point perpendicularly below the center of the upper roll to where the plate touches the second lower roll. It is also evident that the edge of the plate directly opposite to that first entering the rolls will have a flat uncurved section and the width of each section will vary with the diameters of the rolls. Large-diameter rolls are needed for heavy plates, and for long plates. With plates rolled to form one-half the circumference and 16 or 18 feet long, the diameters of the rolls must be increased to prevent springing in the middle; the rolls being supported at the ends only. Even then, with sheets of this length, we find considerable springing, as may be expected with cold rolling. With hot rolling difficulty is also experienced, the plates increasing in width as is evidenced by the mills requiring larger percentage as regards thickness.

To curve the flat section at each end of the plate where the horizontal seams are located is a most important object. In many instances the sledge is used for this purpose and the metal mauled into a rough irregular shape. S. F. Jeter described in Power, December 21, 1909, the results of this found on the seams of a boiler wherein one plate formed the full circle, and showed with illustrations the actual deformation along the line of the horizontal seam. But with a sheet forming half the circumference and allowing for the spring of the rolls, it would appear reasonable to expect the plate to be more out of a true circle than would a full-course sheet. Consider then the effect of one sheet on the bottom as compared with a plate forming a full circle but only 6 or 8 feet long. True, some shops use special tools to force the plate evenly into a circular shape. Also, other shops use a section of very heavy plate as a liner over the lower rolls, placing the sheet on this liner to finish the operation. But this is impracticable with sheets 16 feet long and rolled into half circles.

With a boiler built either in full-ring course or in half sheets extending from head to head, the importance of having the plate truly curved to the proper radius cannot be underestimated. If this is not done the continual flexure or bending action, will eventually result in failure at the joint regardless of whether the seam be a butt, strap or lap joint; for the wider the joint then the stiffer it will be. Indeed, we might expect less trouble with a single-riveted lap joint as it would to some extent act as a hinge. Not that I advocate such a joint, but it is mentioned in order to discuss the effects of this flat section.

If we design a boiler 16 feet long to be built in three courses the breaking joints are on opposite sides and we secure the benefit of the girth seams in stiffening the shell, with the weak sections—the horizontal seams—away from each other. Then we have but 16 feet of longitudinal seam whereas with the "one sheet on bottom" boiler we have 32 feet of weak section without any circular seams to support it. Consider the hoops in a barrel as an illustration of the principle applied to a boiler. In the three-course construction the weak sections of the seam are not continuous. In this form of boiler the long seams are placed above the tubes in the steam space where they can be examined both internally and externally.

I believe that beyond question, boilers having one sheet on the bottom are structurally weaker, regardless of rules for determining the safe working pressure, than boilers built in courses. Details may vary with different explosions of boilers built in this manner. In the Canton explosion the rivets sheared, the plate being 3/8 inch and the rivet holes 13/16 inch with a pitch of 3 inches. Other details of the joint were not given. Assuming the usual flexure, it may be that the plate was better or stronger than the shearing resistance offered by the rivets, and the latter received the effects of the bending action.

Looking further into boiler construction, assume a 1/2-inch plate to be rolled to 72 inches diameter. The outer circumference would be 226.19 inches, the inner 223.05 inches; that is, one side would be 3.14 inches longer than the other. Stresses are thus set up that cannot be determined to a certainty. Some little is known of the molecular grouping of steel and of the cohesion of the molecules, wherein rests the strength of the plate. When the molecules begin to slip apart and loosen their hold upon one another we say that the limit of elasticity has been reached and we avoid loads that will produce this result. In a 72-inch diameter boiler with ½-inch plates shall we say that the metal at the inner side of the plate has become denser, or what would appear to be more correct, that the cohesion of the moleculcs at the outer side of the plate has become lessened? It will be noted that this bending action is on a line parallel to the axis of the cylinder and in the path of the horizontal seam. It is usually contended that the factor of safety takes care of these stresses, but we may define this factor of safety as one representing a "factor of ignorance" of something of which we have no correct knowledge.

Looking further into the subject, let us consider the question of punched rivet holes as regards injuring the net section of the plate between the rivet holes and as to whether the joint should be a lapped or a double-butt strap. Specifications calling for the holes to be drilled, or if punched to be reamed out ¼ inch, are ridiculed as old fashioned. English practice as well as United States marine rules recognize the injurious effects of flat punching. W. S. Hutton is authority for the statement that through punching the loss in tenacity is 8, 18 and 26 per cent. for plates ¼, ⅜ and ½ inch, respectively. We need accurate data on the subject but doubtless we shall find that the holes had best be reamed ¼ inch full when assembled.

If one concedes even part of what has been said in this discussion, then the question follows: "What should be done regarding boilers now in operation and constructed with one sheet on the bottom?" May I suggest the value of the hydrostatic test, even granting this may set up strains tending to weaken the boiler? A test applied under the eye of a competent inspector and with the brickwork removed, in order that both seams can be examined, appears to be the best way; and the engineer should afford the necessary facilities for such a test. Also, he may easily provide wooden curves properly supported and struck from a radius, say, 2 inches longer than that of the boiler, and use these to note the proximity to a true circle of the boiler shell both under and without pressure.

The factor of safety for boilers of this type should be greater than for those built in three courses, regardless of rules. This factor should be increased 20 per cent. when the "one sheet on bottom" boiler is 10 years old. In addition I am so convinced of the structural weakness of boilers of this type that I believe no more boilers should be built on these lines and that it is the duty of persons operating or owning such boilers to replace them with new ones designed on better lines.

Doubtless many will say the case has not been proved beyond the point of a doubt. Even so, the fact remains, singular, sinister, and not otherwise explainable, of three disastrous explosions of boilers within a few months built in this manner together with others herein mentioned. Evidence of so strong a nature surely demands consideration.

The annual meetings of the several associations devoted to steam engineering in its various branches will soon take place. In view of the alarming number of boiler explosions would it not be well for them to devote some attention to a consideration of what may be done to assist in preventing further explosions of this kind?

Massachusetts N. A. S. E. Convention

The usual large attendance of delegates and guests assembled at Lowell for the fifteenth annual convention of the Massachusetts State Association of the National Association of Stationary Engineers. The meeting covered a period of three days.

The exhibition of steam and mechanical appliances was opened on Thursday morning, July 7, in Runel's hall, in which the meetings of the convention were also held.

On Friday morning at half past ten o'clock, the convention was formally opened. It was well attended. Past National President Theodore N. Kelsey, chairman of the local committee, called the meeting to order, and after a brief address introduced Mayor John F. Meehan, who welcomed the convention to Lowell. Edward H. Kearney, State deputy, responded fittingly.

Harvey B. Greene, president of the Lowell board of trade, was then introduced and told of the educational advantages of the city. Past State President Fred L. Johnson spoke of the importance of membership in the National Association of Stationary Engineers.

National President William J. Reynolds told of the advantages of State associations, and paid a tribute to Massachusetts as having the best organization.

State President P. E. Tirrell then took the chair, and after appointing the necessary committees, an adjournment was taken until the afternoon at two o'clock.

At the afternoon session considerable important business was transacted. The reading of the reports showed the State association to be in a healthy condition. The election of officers resulted as follows: George L. Finch, president; James H. Sumner, vice-president; Ole B. Petersen, secretary; Walter H. Damon, treasurer; Albert Smith, State deputy. The officers were installed by National President Reynolds. It was voted to hold the next annual meeting at Worcester in July, 1911.

At the close of the session a vote of thanks was given to the Lowell local committee and P. E. Tirrell, the retiring president, was presented a past president's jewel.

One of the main features of entertainment was a smoker at Lincoln hall on Friday evening, at which there were over seven hundred present. The following enjoyable numbers were given: Paragon quartet in vocal selections; Frank J. Corbett, Consolidated Valve Company, songs; John Paine, songs and stories; Frank McCartin, comic songs; Jack Armour, of POWER, monologue and recitals; William Murray, Jenkins Brothers, popular ditties; James E. Donnelly, Scotch songs. The entertainment closed with the singing of "America."

On Wednesday morning the delegates listened to an interesting lecture on "Smoke Prevention," by John S. Schumaker, of Boston. After the lecture, the delegates and visitors boarded a special train for Canobie lake, a beautiful New Hampshire summer resort. The feature of the day was a baseball game between the engineers and supplymen, which was won by the engineers, the score being 8 to 5. There were fully three hundred at the outing. Refreshments were constantly on tap, winding up with an appetizing dinner at three o'clock, after which the return trip to Lowell was made.

During the day suitably inscribed signet rings were presented to Theodore N. Kelsey and Joseph Collins in appreciation of their earnest work in the successful outcome of the convention.

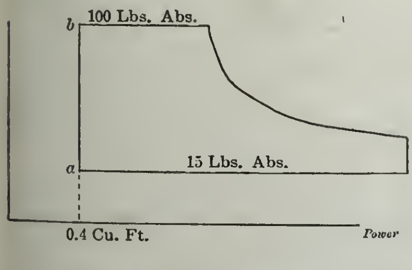
The exhibition was held in a large high ceiling hall, affording ample facilities for the various exhibitors. The booths were erected and decorated by the local association, and everything that could be done for the comfort and convenience of the supplymen was done as only the men in charge of a Massachusetts State convention can do it. Among those who had booths in the exhibit hall were: Albany Lubricating Compound Company; American Oil Company; American Steam Gauge and Valve Manufacturing Company; Ashcroft Manufacturing Company; Ashton Valve Company; Boston Steam Specialty Company; Bundy Department, American Radiator Company; Burke Engineering Company; Cancos Manufacturing Company; Challenge Tube Cleaner Company; A. W. Chesterton & Company; Charles A. Claflin Company; Cling-Surface Company; C. B. Coburn Company; Consolidated Safety Valve Company; Crandall Packing Company; Crosby Steam Gage and Valve Company; Paul L. Crowe; M. T. Davidson Company; Dearborn Drug and Chemical Works; Dyna-Como Company; Eagle Oil and Supply Company; Economy Lubricating Company; Enterprise Rubber Company; Evans Mill Supply Company; Federal Metallic Packing Company; France Packing Company; I. A. Fraser; Gardner Grate Company; Garlock Packing Company; Greene, Tweed & Co.; Hancock Inspirator Company; Hart Packing Company; Hayden & Derby Manufacturing Company; Home Rubber Company; Jenkins Brothers; H. W. Johns-Manville Company; Keystone Lubricating Company; George W. Knowlton Rubber Company; Lagonda Manufacturing Company; Lake Erie Boiler Compound Company; Liberty Manufacturing Company; Lloyd Manufacturing Company; Lunkenheimer Company; McLeod & Henry Company; Mason Regulator Company; Mechanical Rubber Company; Monarch Valve and Manufacturing Company; *New England Engineer;* Patterson Lubricating Company; Peerless Rubber Manufacturing Company; Philadelphia Grease Company; POWER; *Practical Engineer;* Quaker City Rubber Company; Revere Rubber Company; Walter G. Ruggles Company; C. E. Squires Company; Strong, Carlisle & Hammond Company; J. H. Williams & Co.; William R. Winn; Herbert W. Yeomans.

A Steam Problem

Try this problem from Professor Spangler's "Notes on Thermodynamics." There is a development in the solution which may surprise you.

We have given:

Initial pressure, 100 pounds absolute; exhaust pressure, 15 pounds absolute;



quality of entering steam, dry saturated; quality of exhaust steam, 0.9; volume of clearance, 0.4 cubic foot. Find the weight of the steam at *b* and its condition, assuming that there is no heat transfer to the cylinder walls.

Boiler Explosion in New Hampshire

Shortly before six o'clock on the afternoon of July 5, a horizontal-tubular boiler of single-riveted lap-seam construction in the Lakeport Steam Laundry, Laconia, N. H., exploded, wrecking the building, killing one person and injuring seven others.

More specific information will be given in a later issue.

PERSONAL

On June 30, James S. Glew resigned his position as general foreman and master mechanic for the Wagner Electric Company, St. Louis, Mo. He will make an extended trip through the West.

His shopmates presented to him a token of their esteem in the form of a watch-charm which bears the emblems of the various Masonic orders of which he is a member.

SOCIETY NOTES

The United Standard Engineers' Mutual Benefit Association will hold its annual outing this year at Queens Avenue hotel and park, Twentieth street and Queens avenue, Flushing, L. I., on Sunday, August 14, 1910.

The eighty-ninth meeting of the National Association of Cotton Manufacturers will be held on September 21 and 22, at the New Mathewson house, Narragansett Pier, R. I.

The several papers, which were either offered too late to find a place on the program at the last meeting, or have been recently offered, are to be presented at this meeting. The secretary desires additional papers from the members.

Further detailed information will be given about September 1, by which time all of the preliminary arrangements will have been completed.

The fourth annual outing and field day of the Engineers' Blue Club, of Jersey City, N. J., was held on Sunday, July 10, at Midland park, Grant City, Staten Island.

At ten o'clock in the morning breakfast was served, after which outdoor sports of all kinds were indulged in, the winner of each race was presented with a valuable prize. The big event of the day was the baseball game between the engineers and the New York "Bunch." The contest ended in a victory for the "Bunch" by the score of 19 to 7. M. V. Ryerson and Herbert Self were the umpires.

Dinner was served at five o'clock, and after the several prizes had been distributed, including three past-president emblems, the company boarded the cars for home, everyone voting the day a very pleasant one.

The committee of arrangements were Martin J. Hickey, William H. Cronley, John H. Foote, Frank A. La Pointe and John Callahan.

NEW PUBLICATIONS

ELECTRIC POWER PLANTS. By T. E. Murray. Published by the author. Cloth; 337 pages, 6x9 inches; 152 illustrations.

This is a series of power-plant descriptions, the subjects being those stations which have been designed by Mr. Murray. They comprise the two "Waterside" stations of the New York Edison Company; the Central (Third Avenue) and Williamsburg (Kent Avenue) stations of the Brooklyn Rapid Transit Company; the Gold street station of the Brooklyn Edison Company; the stations of the Citizens' Light and Power Company, Rochester, N. Y., and the Utica (N. Y.) Gas and Electric Company; the station (hydraulic) of the Chattanooga (Tenn.) Power Company; the power house of the Helderberg Cement Company, and five representative substations in New York City.

The descriptions are very comprehensive and fully illustrated, so that the book is an excellent presentation of the latest engineering practice in power-station design and construction. It is not on sale; copies have been distributed chiefly amongst Mr. Murray's numerous friends and acquaintances in engineering circles.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

WIND MOTOR. Thomas H. E. Folger, Corral, Idaho. 961,766.

COMPOUND STEAM ENGINE. Herbert H. Pilcher, Mallins, Tenn. 961,795.

ROTARY ENGINE. Andrew A. Edwards and Adna Olson, Stanwood, Wash. 961,849.

CARBONIC ACID MOTOR. Ludwig Horst, Altona, Germany. 961,859.

EXPLOSION ENGINE. Pierre Alexandre Darracq, Suresnes, France, assignor to Societe A. Darracq & Cie. (1905), Limited, Suresnes, France. 961,938.

EXPLOSIVE DRIVEN ENGINE. Albert A. Jahnke, Oakland, Cal. 961,966.

ROTARY EXPLOSIVE ENGINE Elmer A. Thomas and William V. Ritter, St. Louis, Mo. 962,064.

TURBINE. Birger Ljungstrom, Stockholm, Sweden. 962,088.

INTERNAL COMBUSTION ENGINE. Geo. John Altham, Fall River, Mass. 962,110.

INTERNAL COMBUSTION ENGINE. Frederick E. Dayes, Brooklyn, N. Y. 962,131.

COMPOUND REVERSIBLE ROTARY ENGINE. Hermand Eriksen, Philadelphia, Penn. 962,204.

AIR-COOLED ENGINE. William J. Miller, Springfield, Ohio, assignor, by mesne assignments, to the Kelly Motor Truck Company, Springfield, Ohio, a Corporation of Ohio. 962,233.

MOTOR. Albert F. Rockwell, Bristol, Conn., assignor to the New Departure Manufacturing Company, Bristol, Conn., a Corporation of Connecticut. 962,254.

WATER TURBINE. Thomas B. Lee, Charlotte, N. C. 962,364.

HYDRAULIC WATER MOTOR. Lemon I. Osborn, Portland, Ore. 962,382.

BOILERS, FURNACES AND BURNERS

CRUDE-OIL BURNER. Rufus C. Reed, Jacksonville, Tex., assignor of one-half to W. A. Day, M. W. Totty, and R. D. Atkinson, Jacksonville, Tex. 961,797.

COMBINED GAS- AND SMOKE-CONSUMING ATTACHMENT. Frank Shmonsky, Ridgway, Penn. 961,810.

OIL BURNER. Henry R. Green, Spokane, Wash. 962,212.

OIL BURNER. William C. Blackmond, Tomball, and Guy Everett Stoy, Teague, Tex. 962,301.

STOKER MECHANISM. Paul L. Crowe, Jersey City, N. J. 962,322.

WATER-TUBE BOILER. Signono C. Munoz, Montclair, N. J., assignor to the Babcock & Wilcox Company, New York, N. Y., a Corporation of New Jersey. 962,454.

BURNER FOR LIQUID FUEL. Frank S. McKibben, Memphis, Neb. 962,457.

CONSTRUCTION OF MOTIVE FLUID GENERATORS. Frank S. McKibben, Memphis, Neb. 962,457.

CONSTRUCTION OF MOTIVE FLUID GENERATORS. Frank S. McKibben, Memphis, Neb. 962,458.

GASEOUS FUEL BURNER. Henry Schneider and Clarence Schneider, Lima, Ohio. 962,472.

VAPORIZER FOR GAS PRODUCERS. Baxter M. Aslakson, Salem, Ohio. 962,610.

CONSTANT SUPPLY DEVICE FOR VAPORIZING FUEL GAS MACHINES. John F. Barker, Jr., Boston, Mass., assignor to Gilbert & Barker Manufacturing Company, Springfield, Mass., a Corporation. 962,611.

OIL BURNER. Roland C. Casad, Covina, Cal. 962,695.

GRATE BAR. Anderson Wilderspin, Grand Saline, Tex., assignor of one-half to David C. Earnest, Dallas, Tex. 962,805.

BURNER. George J. McPherson, Salt Lake City, Utah, assignor, by direct and mesne assignments, to International Oil Gas Producer Company, a Corporation of Utah. 962,971.

HYDROCARBON BURNER. Ira L. Owens, Effingham, Kan. 963,010.

POWER PLANT AUXILIARIES AND APPLIANCES

CONDENSER. George M. Harden, Philadelphia, Penn., assignor to the Philadelphia Pipe Bending Company, a Corporation of New Jersey. 962,427.

PROCESS OF REMOVING SCALE FROM PIPE. Herluf A. F. Petersen, Chicago, Ill., assignor to J. H. W. Petersen, Chicago, Ill. 962,462.

PUMP. James E. Naughtin, Chicago, Ill. 962,518.

RELIEF DEVICE. Rudolph Conrader, Erie, Penn. 962,551.

BELT. Jackson W. Foster and Albert A. Stoddard, Jacksonville, Fla. 962,563.

PLUNGER PACKING FOR ELEVATORS. James C. Phelps, Springfield, Mass. 962,585.

PISTON RING. Justen Stoddard, Stillwater, Minn. 962,599.

SPARK PLUG. Gregor Walzel, New York, N. Y. 962,604.

CARBURETER. Harry A. Miller, Los Angeles, Cal. 962,649.

STARTING APPARATUS FOR GASOLENE ENGINES. Vaughan Morrill, Tacoma, Wash. 962,653.

VALVE. James G. Nolen, Chicago, Ill., assignor, by mesne assignments, to Automatic Fire Protection Company, a Corporation of Maine. 962,660.

WATER-FEED REGULATOR. Walter H. Bice, Maple Lodge, Ontario, Canada, assignor of one-half to Alfred Westman, and one-sixth to William R. Colby, London, Canada. 962,735.

PUMP GOVERNOR. Walter H. Bice, Maple Lodge, Ontario, Canada, assignor of one-half to Alfred Westman and one-sixth to William R. Colby, London, Canada. 962,736.

AUTOMATIC PRESSURE-REDUCTION VALVE. Walter H. Bice, Maple Lodge, Ontario, Canada, assignor of one-third to Alfred Westman and one-sixth to William R. Colby, London, Ontario, Canada. 962,737.

VALVE. Edward A. Lacy and Fred C. Williams, Galeton, Penn. 962,766.

FLEXIBLE CONDUIT. George A. Lutz, Plainfield, N. J., assignor to American Circular Loom Company, Portland, Me., a Corporation of Maine. 962,772.

HOISTING MECHANISM. George W. Murray, Chicago, Ill. 962,779.

STEAM CONDENSER. Preston H. Sellers, East St. Louis, Ill. 962,789.

PACKING RING. Curtis E. Drown, Tucson, Ariz. 962,820.

LUBRICATING APPARATUS FOR STEAM ENGINES. Edward Ryan, Clinton, Iowa. 962,859.

CARBURETER. Stephen P. Sanders, Cupertino, Cal. 962,860.

STEAM PUMP. John March, Dillon, Mont. 962,906.

WATER-FEED REGULATOR. Walter H. Bice, Maple Lodge, Ontario, Canada, assignor of one-sixth to William R. Colby, and one-half to Alfred Westman, London, Canada. 963,019.

GOVERNOR. Frank L. Nichols, Stamford, Conn., assignor to Nichols Quadruple-Traction Vehicle & Power Company, Stamford, Conn., a Corporation of New York. 962,163.

JOURNAL-BOX BRASS. John M. Rohlfing, St. Louis, Mo., assignor of one-half to Robert E. Frame, St. Louis, Mo. 962,166.

CENTRIFUGAL-OILING BEARING. Frederick A. Warren, Canon City, Colo. 962,182.

CRANK CONNECTION FOR MULTICYLINDER ENGINES. Frank A. Edmunds and Niels G. Warming, St. Paul, Minn. 962,200.

VALVE-CONTROL MECHANISM FOR EXPLOSION ENGINES. Norman T. Harrington, Lansing, Mich. 962,214.

MECHANISM FOR FEEDING FUEL. Albert F. Rockwell, Bristol, Conn., assignor to the New Departure Manufacturing Company, Bristol, Conn., a Corporation of Connecticut. 962,248.

COOLING MEANS FOR MOTORS. Albert F. Rockwell, Bristol, Conn., assignor to the New Departure Manufacturing Company, Bristol, Conn., a Corporation of Connecticut. 962,249.

PUMP. Albert F. Rockwell, Bristol, Conn., assignor to the New Departure Manufacturing Company, Bristol, Conn., a Corporation of Connecticut. 962,251.

SPARKING APPARATUS. Albert F. Rockwell, Bristol, Conn., assignor to the New Departure Manufacturing Company, Bristol, Conn., a Corporation of Connecticut. 962,255.

VALVE. Thomas Pittinger Long, Barberton, Ohio, assignor of one-half to Willard M. Huffman, Barberton, Ohio. 962,368.

VALVE GEAR FOR FLUID-PRESSURE ENGINES. James Thompson Marshall, Leeds, England. 962,374.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTROMAGNET. Campbell Scott, Yonkers, N. Y., assignor to Otis Elevator Company, Jersey City, N. J., a Corporation of New Jersey. 961,805.

RECHARGEABLE ELECTRIC FUSE. Agostine J. Orelli, Longbeach, Cal. 961,891.

CIRCUIT INTERRUPTER FOR ELECTRIC SPARKING DEVICES. James E. Seeley, Los Angeles, Cal., assignor to High Frequency Ignition Coil Company, Los Angeles, Cal., a Corporation of California. 961,902.

ELECTRIC WATER HEATER. William F. Cutler, San Francisco, Cal., assignor to Cutler National Electric Heater Company, San Francisco, Cal., a Corporation of California. 961,937.

CIRCUIT CONTROLLER. Fred I. Getty, Jennings, La., assignor to Charles F. Gillmann, New York, N. Y. 961,951.

ELECTRICAL CONDENSER. Ray H. Manson, Elyria, Ohio, assignor to the Dean Electric Company, Elyria, Ohio, a Corporation of Ohio. 961,978.

RECEIVER FOR ELECTRIC SIGNALING. Reginald A. Fessenden, Washington, D. C., assignor to the National Electric Signaling Company, a Corporation of New Jersey. 962,015.

RECEIVER FOR ELECTROMAGNETIC WAVES. Reginald A. Fessenden, Washington, D. C. 962,016.

ALTERNATE-CURRENT COMMUTATOR MOTOR. Rudolf Richter, Charlottenburg, Germany, assignor to Siemens-Schuckertwerke G. M. B. H., Berlin, Germany, a German Corporation. 962,101.

ELECTRICITY METER. Dugald C. Jackson, Madison, Wis., assignor to Dugald C. Jackson and William B. Jackson, Madison, Wis., a Copartnership. 962,222.

ELECTRIC TIME SWITCH. Carey E. Bunker, Oregon, Mo. 962,307.

ALTERNATING-CURRENT MOTOR. Valere Alfred Fynn, London, England. 962,335.

POWER PLANT TOOLS

WRENCH. Robert H. Peters, Buffalo, N. Y. 961,894.

WRENCH. William B. Brown, Pittsburg, Penn. 962,304.

PIPE WRENCH. Alexander Kondas and Paul Bajusz, Woodriver, Ill. 962,359.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; -sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power
and The Engineer

Vol. 32 NEW YORK, JULY 26, 1910 No. 30

THE all absorbing topic is wages. At least, so it would appear from the amount of space which it gets in the magazines. And, doubtlessly, so it is with a large class of men.

In a certain respect this is but natural. Wages mean power. Not only the necessities of life, but almost anything in the world, except contentment, can be obtained for money.

According to a man's tastes, wages mean various things.

To one man better wages mean greater ability to "crowd on a better front"—wear bigger, brighter studs; smoke cigars with more gorgeous bands, and show in other ways how much money he can spend without half trying.

Another man desires more wages so that he can play stronger at the fascinating game of chasing dollars into a bank account.

More wages, to a third man, means a feeling of greater security against privation.

And so we could go on through the whole category of desire and show that each of the items therein is a motive of the struggle for larger remuneration.

Some of the reasons why men desire more wages are laudable; many are not.

We contend that peace of mind is of far greater importance than the gratification of many of the desires which actuate the strife for higher wages.

Please do not conclude from this that we are not for higher wages. We are—*strong*.

We believe, however, that much dissatisfaction is simply the result of not realizing when we are well off.

After all, *position* is everything.

One of our subscribers in the West, wrote: "I am a fireman on a tugboat. Now, there is more honor in being a fireman and taking orders from an intelligent, experienced engineer than there is in being an engineer and taking orders from an educated ass."

This is good, sound philosophy. Not everyone can be the chief engineer. Not everyone wishes to be, as the quotation plainly testifies. Although the chief gets many more "ducats" than does our subscriber, the latter does not appear to be envious in the least.

If your position "fits" you, "stick." If you like the work, can do it satisfactorily and if the "man higher up" is a "white man," hold fast to your job even if your wages are not quite as high as those paid to Ernest Browbeaten by Hector Loudmouth, at the next plant above. Ernest probably earns, by his meek submissiveness, several times what he really gets.

It is unsatisfactory business to sacrifice self respect and dignity for a few extra dollars. Life is too short. We believe that if you improve your worth to your employer and become deserving of a better position and better wages, you will find that your wages will improve in proportion. Opportunity comes to everyone. Be ready when your turn comes.

When you have a real grievance, *kick*. But don't, above all, be a common "sore head."

Laboratories at University of Toronto

By Prof. R. W. Angus

The steam, gas and hydraulic laboratories at the University of Toronto, which were completed during the past year show the result of much forethought on the part of those responsible for their design. They are laid out with a view to meeting the peculiar needs of instruction in these departments and the equipment is fairly complete.

The laboratories described in the present article were built and ready for students in October, 1909. They occupy an entire building which is divided into two distinct parts by a wall running the full length of it, the part to the north of this wall being used for hydraulic work and for offices and study rooms, while the part to the south contains the heat engines and other similar apparatus, and in a separate room the boilers. The building has a length of over 200 feet and a width of about 100 feet, the hydraulic section being three stories high, while the remainder of the building is but one story.

At the present time the board of governors of the university have not deemed it advisable to complete the building, so that the front has not yet been erected, although the laboratory section is now complete. The total cost of the part erected to date exceeds $85,000 exclusive of equipment, for which $22,000 has been spent. As a considerable amount of apparatus was moved from the old laboratories, however, the actual value of the equipment would exceed the above mentioned amount.

Steam- and Gas-engine Laboratory

Beginning with the south side of the building devoted to heat engines, one enters a large room 156 feet long by 60 feet wide, lighted throughout from the roof, there being no windows in any of the walls. This room is divided on one side into eight smaller rooms, each 15x20 feet which are used for special apparatus, to be described later. The main part of this large division which is 40 feet wide and the full length of the building, contains all of the engines, and is served throughout by a three-ton traveling crane.

This arrangement has proved admirable, the lighting is as nearly perfect as possible, the ventilation has been found good, and the arrangement of the engines is convenient, while the smaller rooms give students a chance to work quietly on smaller experiments.

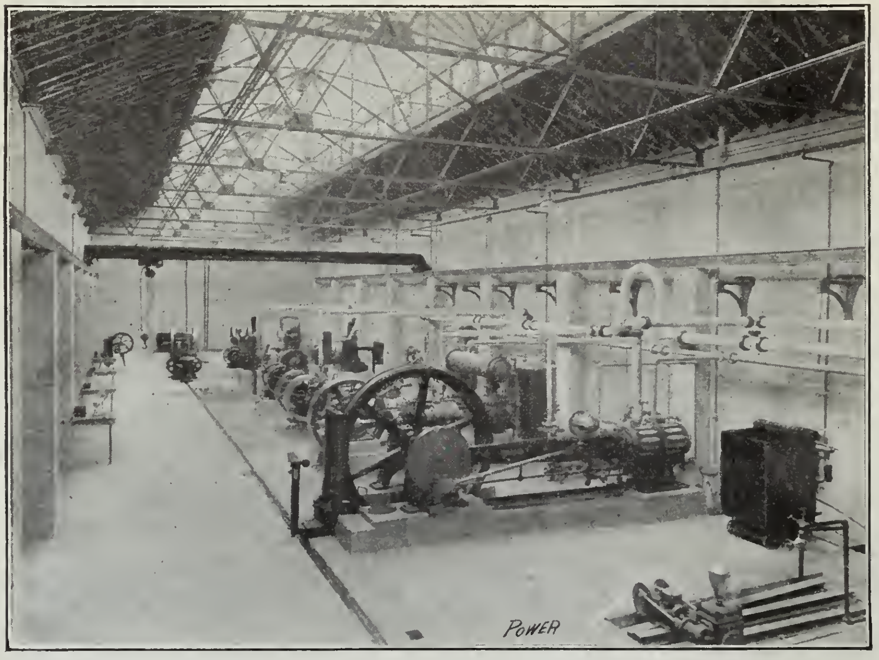


Fig. 1. Steam- and Gas-engine Laboratory, Looking West

Fig. 1 gives a general view of the main part of this room, in which the gas engines are placed across one end and the steam engines along the entire side. Fig. 2 is a photograph of the same room taken from the other end and showing the row of small rooms. There are four gas engines, one of 10 horsepower built by Fielding & Platt, and arranged for burning city or suction gas, or liquid fuel. Adjacent to this is a 22-horsepower gas engine built by the National Gas Engine Company, and like the other has specially heavy wheels, both engines having arrangements for variable compression. The third is an old Otto machine of historic interest. A suction gas producer for use with these engines has been placed in one of the small rooms to avoid annoyance from gas and dirt. A Fairbanks marine gasolene engine completes the list of internal-combustion machines, but mention may be made of test floors which have been arranged for any machines of this type which may be loaned to the laboratories for purposes of investigation. An Ericsson hot-air engine also forms part of this equipment.

Of the steam engines there are two Leonard engines, one 6x8 inches, the other 7x8 inches, both of which are arranged especially for valve setting and indicator practice; an 18-horsepower McEwen; a 12x10-inch Leonard-Ball tandem-compound engine of 29 horsepower; a 50-horsepower Brown engine, jacketed and especially arranged for research work; a 75-horsepower Williams engine, and a 15-horsepower De Laval turbine with six nozzles, condensing and noncondensing. There is also a Rand air compressor with compound steam cylinders and two-stage air cylinders, having a capacity of 340 cubic feet of air per minute at 100 pounds pressure. The steam valves of this machine are of the Meyer type and a Corliss inlet gear is used on the low-pressure air cylinder.

A large test floor has also been arranged in connection with the steam work for the study of engines which may be loaned from time to time.

The arrangement of the steam and exhaust piping and condensers is of some interest and considerable care has been taken in devising a scheme suited to the peculiar needs of a laboratory. It is desirable to have an arrangement permitting of the use of all the engines at the same time, and in order to measure the steam consumption of an engine, a surface condenser is necessary; hence a condenser would have to be provided for each engine if the steam consumption of each were to be measured at the same time. Such an arrangement was deemed too expensive and only two condensers have been installed. The piping permits the simultaneous running of any two engines condensing, while at the same time all of the other engines may be run noncondensing.

In order to accomplish this a large trench was made in which were placed two parallel 6-inch exhaust-pipe lines, one of which passes to a Cochrane heater and thence to the atmosphere, while the other is connected to two surface condensers. The exhaust from each engine passes under one of these pipes and turns upward by means of an elbow; on top of this elbow is a second one which may, by turning through 180 degrees, be connected by a tee to either of the exhaust lines. Thus it becomes a very simple operation to run an engine which ever way desired. The arrangement has worked very successfully during the past session. Of course, proper drains have been provided throughout.

Two Wheeler surface condensers with separate air pumps have been installed in connection with the above scheme, the circulating water being handled separately.

The main steam pipe is 5 inches and runs from the boilers along the side of the building, connections to the engines being from the top so as to avoid water draining to them. The fittings are all extra heavy and two expansion bends have been inserted, one in the boiler room and the other in the main engine room, the latter bend being visible in Fig. 1.

In addition to the above steam main, a special 3½-inch pipe runs from one of the 50-horsepower boilers to the Brown engine, so that a feed-water test on the engine may be made when desired. Connections have also been made so that this boiler may deliver steam to the 5-inch main and the Brown engine may in turn draw steam from this main.

All the engines are arranged with brakes of a very simple construction, the writer believing that the simpler the apparatus, consistent with accuracy, the greater will be the value of the instruction.

Of the eight smaller rooms on the south side of the main steam laboratory, the first two are connected and are used as a small repair and instrument-making shop, and are equipped with a lathe, drill press, etc. This shop is not used for instruction to students but merely to make necessary repairs to the machines.

The next room will contain the belt- and rope-testing machines, the power put into the belt being determined by an Amsler torsion dynamometer. The fourth room contains a completely arranged three-ton York steam-driven refrigerating plant.

Of the remaining four rooms, one contains the 20-horsepower suction-gas plant, already mentioned; another contains oil- and fuel-testing apparatus of various kinds; while the third has been set apart as a study room. The remaining room is used for instruments, and is arranged and managed as nearly as possible like the tool room in a shop, each student handing in a list of the instruments required for his experiment and being held responsible for their return in proper condition at the close of the day. This method has been very effective in checking the loss or damage of apparatus.

The Hydraulic Laboratory

The hydraulic laboratory occupies the greater part of the north half of the building and is located on the basement and ground floors, each of which is 40 feet wide and 113 feet long. It was impossible to obtain such perfect lighting for this laboratory as for the engine room on account of having to take light from the side, but very good results have been obtained by the use of large windows which occupy fully three-fourths of the total wall space, and it is thus comparatively easy to work during any but the darkest days without the use of artificial light.

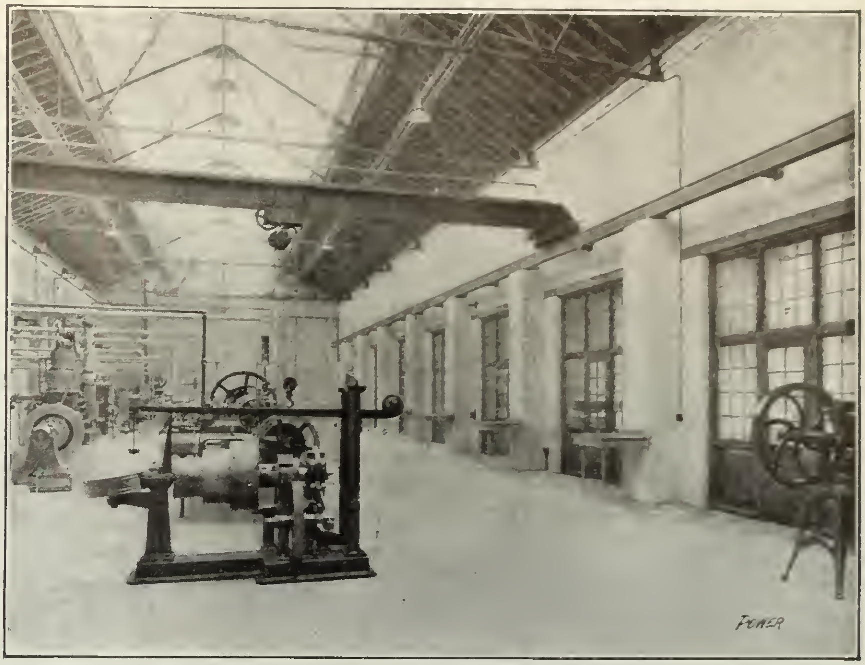


Fig. 2. Steam- and Gas-engine Laboratory Showing Small Rooms

The lower story, or basement is 18 feet high and contains the well, the pumps and the engine for driving them, the measuring tanks used for calibrating tne orifices and weirs above, several large troughs and the main part of the piping for the entire laboratory. The upper floor, which is 15 feet high, contains the orifice and weir tanks, the experimental centrifugal pump with its weir tank and motor, the various pipes for friction experiments, the meters, and also tne various types of turbines and other apparatus.

Beginning with the top floor at the west end, see Fig. 3, the first piece of apparatus is a centrifugal pump of special construction for experimental work. This pump, which is driven by a variable-speed motor, is mounted on a weir tank so that the discharge may be easily measured, the various heads under which it is operated being obtained by throttling.

Adjacent to this are two orifice tanks and a weir tank used mainly for instruction in the use of orifices and weirs. The water is discharged through 6-inch pipes into calibrating tanks below, of which there are six, two for each orifice and weir. These calibrating tanks have been designed so as to be operated entirely by the movement of a single lever. When the lever is put in one extreme position, one tank is made to fill, its outlet valve being closed, while at the same time the outlet valve of the other tank is open. For small discharges the tank fills so slowly that the upper surface of the water is quite smooth and undisturbed, and therefore the exact hight of the water in the tank, and consequently the volume of the water at the moment of dumping is easily observed; but where the discharges are large the surface of the water in the calibrating tank is so disturbed that accurate observations of this kind are impossible. To obviate this trouble the mechanism is so designed that by moving the lever to an intermediate position the outlet valve on the other tank is closed and the discharge is turned into it. The water surface in the first tank comes to rest at once and its hight is easily measured; after which the lever is pushed to the other extreme position and the outlet valve of the first tank is opened without disturbing the conditions in the second tank. The quantity of water in the tank after filling may be found by direct weighing or by reading the level in an attached gage glass.

An opening in the floor protected by a rail enables the student to observe the whole operation at one time.

In order that experiments on these

tanks and orifices may be carried on without interfering with the other work, a large reservoir provided with an overflow has been placed above them and from this the water is drawn. So long as there is enough water passing into the reservoir to cause some overflow, the experiments may be carried on at various heads and the water discharges without difficulty. Water from this reservoir also drives a hydraulic ram.

A 3-inch Venturi meter has been installed on a pipe in which other meters will also be placed. This meter is rated by sending the discharge from it through one of the orifice tanks already described, and using the coefficient determined by actual experiment.

Near the center of the laboratory an elevated platform has been erected for the two impulse turbines, one of which is a 12-inch Doble wheel with glass sides enabling the action of the water to be studied conveniently; the other is an 18-inch Pelton wheel. The water used passes over a weir and then back to the pump well; for the use of these turbines it is possible to get a discharge of 1 cubic foot per second against a maximum head of about 500 feet.

At the east end of the room is a large standpipe 32 feet high and 5½ feet diameter, which is used as a reservoir for the reaction turbines and also for any experiments requiring constant head. This standpipe has nozzles for the attachment of turbines and orifice plates and tubes of various sizes. The water is delivered to the standpipe through two pipes near the bottom, and by an arrangement of baffle plates there is no trouble caused by the surging or eddying of the water.

In front of this standpipe is a large weir tank 20 feet long by 6 feet wide, which is arranged with a sharp-crested weir 4½ feet wide, with end contractions. This plate may, however, be changed so that weirs of different sizes, with or without end contractions, may be easily inserted. The weirs used are calibrated by means of a pair of measuring tanks located in the basement, each holding approximately 240 cubic feet.

There are three reaction turbines; the smallest one has a 6-inch runner and has been set up in a steel penstock and connected to the standpipe by a 14-inch steel pipe containing two elbows. This pipe is to be used for experiments on the flow of water, the conditions being examined by a Pitot tube. The turbine has been operated under a total head of over 25 feet; the horsepower is measured by a Prony brake and the discharge is determined from the 4½-foot weir. There is also a 9-inch McCormick wheel.

The latest turbine is one purchased recently from the firm of Escher, Wyss & Co., Zurich, Switzerland. This wheel is of the horizontal Francis type and has a spiral casing and 14-inch runner; the gates being operated through a ring by a handwheel on the casing. A conical draft tube over 7 feet long is used and the wheel will deliver 10 horsepower when supplied with 6 cubic feet of water per second at a head of 20 feet.

Along the north side of this floor near the windows, arrangements are made for the testing of the friction in fire hose and pipes. A 2½-inch iron pipe 50 feet long has been set up, also a 50-foot length of fire hose in which the frictional losses are determined. The flow in the iron pipe is studied by a Pitot tube.

The basement, shown in Fig. 4, contains, in addition to the orifice and weir calibrating tanks already mentioned, the well and pumping plant; all water used in the laboratory being drained back to a large well from which it is pumped into the system and used again.

There are two sets of turbine pumps used for this purpose, one of the Gwynne type and the other of the Escher & Wyss type. Each set consists of two two-stage pumps bolted to a common bedplate and driven by a single pulley, the couplings being arranged so that either pump may be operated separately. Further, the two pumps on the one base are piped together in such a way that they may be made to deliver to the line in three ways, viz., separately, in series and in parallel. A connection is also arranged to connect the discharge from one pump to the suction from the other pump, thus placing the two sets in series for high-pressure work.

Each set of pumps will deliver 1 cubic foot of water per second, one set acting against a head of 125 feet while the other will deliver against a head of 150 feet; the discharge at lower heads being, of course, much greater.

The arrangement of the plant is very flexible, permitting the operation of the four pumps separately, thus allowing four absolutely independent experiments at one time; the operation of the two sets separately on separate experiments; the operation of the two sets in series for high pressures and moderate discharges such as are required in fire streams and impulse turbines; and the operation of the two sets in parallel, giving about 6 cubic feet per second at the heads available in reaction-turbine work in the laboratory.

The pumps are of modern design and are belt driven from a jack shaft placed on the floor to avoid vibration. The jack shaft is driven by a Belliss & Morcom engine of 130 horsepower running at 530 revolutions per minute. The exhaust from this engine is used to heat the building.

A large trough 6 feet wide, 4 feet deep and 110 feet long has been provided in the basement for the rating of current meters and tubes and other work of this nature. It is believed that this will be especially useful in view of the rapid advance in water-power development at the present time.

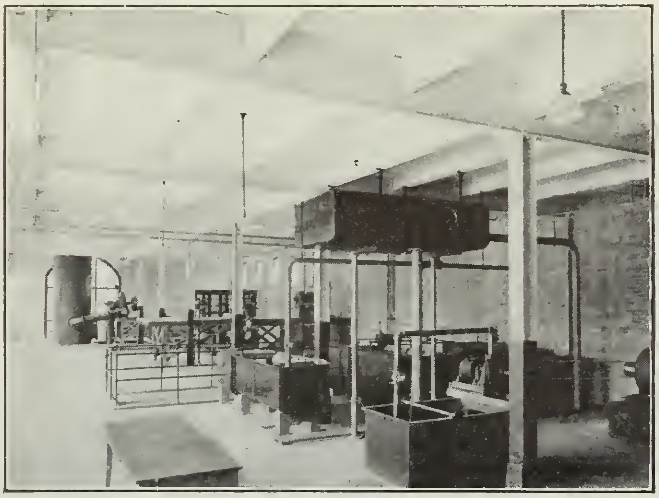


FIG. 3. HYDRAULIC LABORATORY, TOP FLOOR

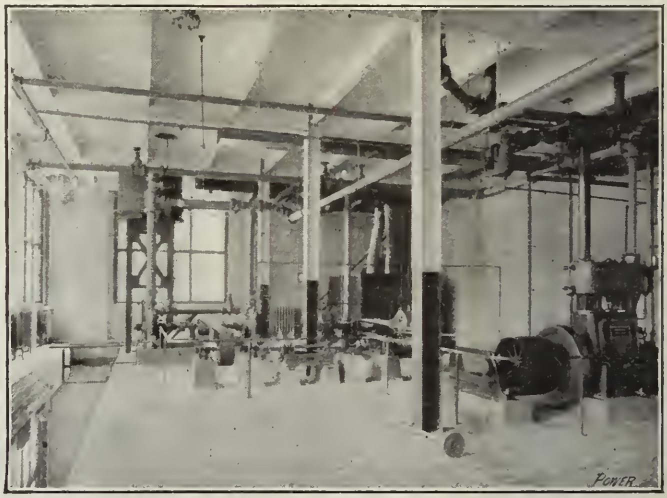


FIG. 4. HYDRAULIC LABORATORY, BASEMENT

THE BOILER ROOM

This room is 70x45 feet and contains three Babcock & Wilcox boilers, all built for 200 pounds pressure, and having separate feed pumps.

The 100-horsepower boiler is provided with a superheater, while the two 50-horsepower boilers have been set up independently, one being arranged with a special type of setting so that comparative results may be obtained.

A steel breeching conducts the products of combustion to two 36-inch by 100-foot brick stacks. An arrangement of dampers allows these stacks to be operated simultaneously at different powers and thus reliable experiments may be made on the capacity of each stack.

OFFICES AND STUDY ROOMS

At the northwest corner of the building, just west of the hydraulic laboratory, is a space occupied by the stairways, halls and other rooms. These rooms are as follows: In the basement, the students' room with shower bath, on the first floor the professors' and lecturers' rooms, while the top floor is occupied by a lecture room and a private lavatory.

The entire floor above the hydraulic laboratory is divided into students' study rooms, a lecture room, a library, a demonstrators' room and a students' lavatory. All of these rooms are comfortable and bright, and the students' rooms are provided with convenient tables and a locked drawer for each person.

ARTIFICIAL LIGHTING

The large thermodynamic laboratory is lighted by ten Nernst lamps, which are attached to the trusses and provide sufficiently good light to do any work required at night. The hydraulic laboratory is lighted by Tungsten lamps, there being ten to each floor.

The remainder of the building, with the exception of the boiler room, which also has Nernst lamps as well as incandescent lamps, is lighted entirely by incandescent lamps.

Selection of Lubricating Oils

By Charles B. Arnold

> *Owing to their superior qualities, mineral oils are now almost universally used for the lubrication of cylinders and other parts subjected to high temperatures. In the selection of these oils the flash point, specific gravity and viscosity must be carefully considered, as certain operating conditions require certain values for these factors.*

Of the many supplies used in a power plant, the lubricating oil is one of the most important, and it is essential that only oil of the proper quality be selected for the various bearings and cylinders.

Of the numerous kinds, there are animal oils, found in small animal substances such as lard and sperm; vegetable oils, such as rape oil made from a French turnip and castor oil obtained from the seed of a plant grown in the West Indies; and lastly mineral oils, from which the greater part of our lubricants are made. Formerly only animal and vegetable oils were used to lubricate the cylinders, but it was found that they decomposed under high heat and generated acids which corroded the metal of the cylinders, and further, when using condensers and returning the condensation to the boilers, these oils led to the corrosion of the boiler plates. This objection was removed by the introduction of proper mineral oil for cylinder use.

The amount of oil cannot be estimated exactly by counting the drops, as one cylinder oil may feed more at the rate of three drops a minute than another at six drops a minute. Consequently, when it is desired to ascertain the amount used in a certain time, it is necessary to measure the oil instead of counting the drops.

For use in a cylinder where the steam pressure is less than 100 pounds per square inch, the oil should show a flash test of not less than 590 degrees nor more than 630 degrees Fahrenheit, and for pressures over 100 and up to 200 pounds, the flash test should be from 640 to 660 degrees. The specific gravity of a cylinder oil showing a flash test of 600 degrees should be from 26 to 27, and for a flash test of 660 degrees, from 24 to 25.

Cylinder oil with a high flash point, when used in a cylinder carrying a low steam pressure, will not give satisfactory results, as the heat is not great enough to thoroughly atomize the oil, which passes through with the steam without thoroughly lubricating the surfaces of the cylinder. On the other hand, an oil of a low flash point should not be used with high-pressure steam, as it loses its lubricating qualities and passes out of the cylinder without doing its work.

For the cylinder the pure mineral oil is the best, and it is not necessary to compound it with animal or vegetable oil. Some engineers insist upon using tallow with their cylinder oils and if this is done, nothing but refined tallow should be used, never common lump tallow, as it contains acid which attacks the cylinder walls and leaves charred particles in the steam. Cheap grades of cylinder oil are often adulterated with wool fat, which is used to cut the gummy ingredients and give the oil a good flow in a cold test. Wool fat, however, causes a separation and a thick deposit in the bottom of the barrels, resulting in the same charred deposits in the cylinder as with lump tallow. In the report of the annual meeting of the Engine Builders' Association, the following statement was made: "The lubrication of the cylinders of engines, where superheated steam is used, is apt to cause a great deal of unnecessary anxiety to one not experienced with superheated steam." This problem is very simple, however, if it is borne in mind that the temperature inside the cylinder is never very high, even when the superheat at the entrance is considerable; and, if the cylinder is lubricated direct without mixing the oil with the steam at the entrance point, there will be no difficulty, provided a good grade of mineral oil is used.

The value of any kind of liquid for external lubrication depends to a large extent upon its viscosity. The viscosity of the oil changes directly with the temperature; as the temperature increases the oil will flow more freely and have less body. An oil of high viscosity should be used in a room where the temperature is high, as the greater the viscosity of the oil the thicker will be the layer or film between the rubbing parts.

Flat surfaces, such as the guides of an engine, are more difficult to keep lubricated than the journals. The latter in turning form an unbroken film of oil, and if there is a continuous supply this film will remain unbroken. The oil wedges itself between the journals and bearings and keeps them apart, and as long as this film is maintained, the journal and bearing do not come in contact, therefore causing no wear. If there are high places on the bearing or if grit thicker than the film of the oil gets in the bearing, heating will result, the high places heating first; the oil will then become thinner, leaving this part of the bearing where it is needed most and flowing to the cooler parts. The fact that the journal runs without heating is no proof that the friction cannot be reduced. An oil should be selected of the right viscosity for the speed of the journal, the pressure on it and its fit with this bearing. A loose-fitting bearing requires a heavier oil than one which runs with a perfect fit.

A pure mineral oil is found to be the best for external lubricating purposes and is more generally used. Vegetable and animal oils are apt to become sticky and gummy, and do not work well. Engine oils for high speeds should show a viscosity of at least 170 and should range up to 195 at an ordinary temperature of about 75 degrees. The oil should show a specific gravity of about 30 and the flash test should be between 400 and 450 degrees Fahrenheit. In the case of a slow-speed engine the oil should have a viscosity of from 195 to 215 and a specific gravity of 24, with a flash test between 440 and 460 degrees.

Much attention has been given to the reduction of steam consumption per horsepower, but when it is considered that in small- and medium-sized engines working at full load, 8 per cent. of the power, and at one-quarter load 35 per cent. of the power developed is wasted in journal friction, there seems to be much room for improvement in this line.

The Efficiency of Waterwheels

By Henry D. Jackson

The angle at which water enters and leaves; the friction in guide vanes, buckets and bearings; and leakage between runner and case, all influence the efficiency of a waterwheel, which seldom exceeds 80 per cent. under actual operating conditions.

In many cases owing to the impossibility of testing waterwheels, guarantees of performance are made which are considerably beyond what can actually be accomplished. It sometimes happens that after a wheel is installed, owing to some change in the situation, a test is made possible and the results show that the wheel is not developing anything like what was guaranteed. On an occasion like this, it is not unusual to find that the wheel manufacturer easily points out why the test as made does not show the true efficiency of the wheel; he will claim that the test should not take into account various items which do really enter into the wheel efficiency.

Many wheel manufacturers bring forward tests to show remarkable efficiencies for their wheels, and it is not infrequent to hear of moderate-size wheels showing efficiencies as high as 93 and 94 per cent. and of larger ones showing even higher efficiency. It is very easy to make tests which will show efficiencies of over 100 per cent. if the investigator has not had a wide experience in testing wheels and in measuring the flow of water. A study of waterwheel construction and efficiency will readily show one that extremely high efficiencies, in the nineties, are impossible, except under ideal conditions.

The efficiency of a waterwheel depends upon a great many things. For the best efficiency, the water should enter the wheel tangentially and leave radially. The angle at which the water actually can enter or leave is limited by the mechanical construction of the wheel. The actual direction of the water entering the buckets depends upon the speed of the wheel. The actual direction of the water as it leaves the buckets should be nearly radial so that the energy in the water leaving the wheel will be small. The velocity at the entrance rim depends on the type of the wheel, and bears a definite relation to the spouting velocity of the water. Owing to the fact that it is not possible closely to approximate the theoretically best conditions for either the entrance or exit of the water, there is a considerable loss both as the water enters and leaves. There is also a considerable loss due to the velocity of the water as it leaves the wheel, as it is necessary to have this velocity in order that the water will free itself from the wheel and not choke the discharge passages.

These are also considerable losses of energy due to friction in the guide vanes, friction in the buckets themselves, leakage between the runner and the case and the friction of the bearings in the various parts of the wheel. These losses are of the wheel itself. In addition to these, there are other possible losses due to getting water to and from the wheel, which, if not carefully taken into account, may be very considerable.

In many of the tests reported by waterwheel manufacturers, no account is made of any of the losses existing outside of the wheel itself. In many of the guarantees made by the waterwheel manufacturers, unless the purchaser is wary, he will find when the test is made, that the guarantee simply covers the power delivered by the wheel at the wheel shaft with a head determined exactly at the wheel, and that no account is made of the losses which may exist in other parts of the installation, which should be and frequently are a part of the waterwheel manufacturer's contract.

Often, too, the tests reported are based on wheels which have been prepared especially for the test, having been fitted with roller or ball bearings, and having had many extraordinary precautions taken to reduce friction. The runners may have been made exceedingly close, the blades burnished to a glass finish, the edges of the blades made knife sharp, the guide vanes likewise polished, and all edges made as fine as possible so as to avoid any possibility of eddies. Such conditions would only last during the period of the test; and although a wheel may show exceedingly high efficiency under these ideal conditions, the same wheel when installed for practical purposes would show no such efficiency.

In a very large proportion of the tests recorded in catalogs and elsewhere, the speed of the wheel is not maintained constant, nor is it maintained at a constant percentage of the spouting velocity of the water, but under each gate condition the wheel is allowed to find its own best speed. Under most conditions the purchaser of a wheel desires his wheel to run at a constant speed under all conditions of load, hence, a test of this character means nothing to him, and such a test when shown to him is misleading. To be of any value, tests should be conducted for a certain percentage of the spouting velocity of the water, and this percentage should remain constant through the test, as it is frequently impossible to maintain the head constant during the test.

Different manufacturers rate their wheels under different conditions; that is to say, some rate the wheels under a percentage of the gate opening, and others under a percentage of the water discharged. A 50 per cent. gate opening does not mean 50 per cent. of the water flowing at full-gate opening, nor does it mean 50 per cent. of the power developed under full gate. The best way of rating a wheel is on the percentage of the water flowing under different conditions of gate openings; that is, the wheel should be rated at full gate or 100 per cent. discharge; 75 per cent. discharge, 50 per cent., 30 per cent., etc. Under these conditions the purchaser could readily compare wheels on the basis of the percentage of water discharged, as compared to full gate, which is a constant factor for all wheels.

It is frequently desired to know what the speed of a certain wheel will be under conditions of a certain head when data are at hand for some other head only. As all wheels run at a speed which is practically fixed by the spouting velocity of the water, it is apparent that the speed will vary directly as the square root of the head; hence, knowing the speed at a given head, a constant can be derived by dividing this speed by the square root of the head. To determine the speed at any other head, it is merely necessary to multiply this constant by the square root of the head in question. It is also frequently desired to know what the power of a wheel will be at some different head. This is also readily obtained from the fact that the power depends upon the head which is acting on the wheel, and the quantity of water which will flow through the wheel at that head; hence, the power is in proportion to the three-halves power of the head.

This being the case, the power at any head may be divided by the three-halves power of this head, giving a constant which, if multiplied by the three-halves power of any other head, will give the horsepower at that head. It is possible to determine with fair result the speed of a wheel of a certain diameter at any head, from the fact that the average American reaction runner operates at a speed which approximates 65 per cent. of the spouting velocity of the water, and the spouting velocity of the water is approximately

$$0.95\sqrt{2gH}$$

in which g has its usual significance of acceleration due to gravity and H is the head acting on the wheel.

The maximum theoretical efficiency of the present reaction type of waterwheel will not exceed 97 per cent. and frequently falls short of this. The loss due to the energy in the water leaving the wheel will vary from 1 to 5 per cent., although in some cases it may exceed this. The loss in the guides due to friction and eddies will vary from 1 to 10 per cent. as will the loss in the wheel blades, particularly those of the warped type having exceedingly deep buckets, where the water flows inward, downward and outward. No matter how smooth the castings may be made, there is sure to be considerable friction and considerable eddy loss in a wheel of this character. The leakage loss depends very largely on the size of the wheel; it might readily amount to anything from 3 or 4 per cent. up to 10 or 12 per cent., and the friction loss in the bearings is an item of wide variation although it usually amounts to from 2 to 5 per cent. in good wheels.

Let us then take a wheel having a theoretical efficiency of 97 per cent. and subtract from this the various items of loss. If all of the losses are assumed to be the minimum as quoted, the wheel efficiency will not exceed 89 per cent.; if the losses are average, the efficiency is about 79 per cent. That this latter is nearer to the actual conditions that exist than the former is shown by the fact that nearly all wheel manufacturers list their wheels at 80 per cent. efficiency.

It might be worth while to call attention here to the fact that these losses are those of the moderate-head reaction type, and that the leakage losses and the losses in the buckets and guides may be considerably smaller than the average here quoted in the shorter-bucket, radial, inward-flow wheels of the Francis or modified Francis type, and also that the leakage loss does not exist in impulse wheels.

It is evident, therefore, that anyone purchasing a waterwheel should look into the proposition carefully. When efficiencies of over 80 per cent. are guaranteed, particularly over wide ranges of load, the conditions under which these guarantees are made should be examined into and placed in the contract with a bonus and forfeiture clause in order to protect the purchaser. Care should be taken that the efficiencies are guaranteed under conditions of installation and not under those at some testing flume. The purchaser is interested only in what the wheel will do at his plant and under the conditions there; he has no interest at all in what the wheel can do under other conditions. The head should be that between the forebay and the tailrace, and the power should be that actually delivered by the wheel where it is to be attached to the machinery, so that all of the losses in the wheel installation are taken into account. Attention to these items will frequently save more or less discussion and hard feeling. It is also advisable to see that the type of wheel which is offered is suitable to the conditions under which it is to operate. The American type of wheel, having the long, deep buckets and a small diameter of runner, is not adapted to high heads, although it is frequently sold for such purposes. The buckets, being long and deep with a very marked curvature, cause friction and eddy-current loss.

Concrete for Machinery Foundations

By G. Wise

Owing to its many advantages, concrete is now almost universally used for machinery foundations. Such being the case, it is essential that the plant superintendent or engineer, who is occasionally called upon to install small foundations, be familiar with certain fundamental principles which insure the production of good concrete.

At the present time foundations for heavy machinery are almost exclusively made of concrete on account of its low cost, the availability of the materials that enter into its composition, and the ease with which it can be mixed and molded into irregular shapes and around foundation bolts. Concrete is particularly suitable for this purpose, as when completed the foundation is a solid mass; and as such it more successfully absorbs the vibrations of high-speed machinery than would a similar foundation built of brick or stone. Also on account of the plastic condition of the concrete, when put into place it conforms perfectly to any irregularity of the ground; this sustains the foundation and thus insures a perfect distribution of the load over the soil.

Concrete appeals particularly to the plant superintendent for use on occasional jobs because the necessary materials can easily be secured from local sources and the work done by labor already at hand. This, therefore, reduces the total cost to a little more than that of the materials. The information herein given is intended to be of assistance to those who occasionally use concrete in small amounts.

To most people the mixing of concrete seems to be a very simple process based upon empirical rules which have proved satisfactory by past experience Only to a slight extent is this true, as years of study have shown that there are several fundamental principles in connection with the use of concrete, the understanding and application of which will insure satisfactory and consistent results and remove the element of chance which so often exists.

Materials

The materials commonly used for making concrete are cement, sand and crushed stone or gravel, the latter being above ¼ inch in diameter. The cement is the active constituent, the binder which serves to hold the concrete together; the sand and crushed stone, called by concrete experts the fine aggregate and the coarse aggregate, respectively, are the inert or inactive materials which serve not only to reduce the cost but also to increase the strength.

A pile of sand or broken stone does not consist of a solid mass of material, there being a void space between the various particles. The amount of this void space is usually stated as a percentage of the total volume and has an average value for sand of 40 per cent. and for crushed stone of 45 per cent.

The percentage of voids in the fine and coarse aggregates is the principal characteristic which determines their value in the concrete, and is the basis for the proportioning of the materials. This statement is based upon the principle that in a theoretically perfect concrete every particle of the fine aggregate must be coated with cement and every particle of the coarse aggregate with the cement-sand mortar; also, every void must be filled, the smaller grains fitting in between the larger ones, in order to have a dense and strong concrete. A small percentage of voids is therefore desirable, and to secure this condition the fine and coarse aggregates must consist of particles varying in size.

Portland cement of any standard brand may be considered reliable. It is best, however, to buy it from some reputable dealer who is familiar with it. One bag of cement has a net weight of 94 pounds and a volume of approximately one cubic foot.

The fine aggregate used for making concrete may be any sand which is hard, clean and coarse. A sharp sand, while otherwise satisfactory, contains a large percentage of voids and will not produce as strong a mortar as will a sand with rounded grains. Cement cannot adhere to dirty sand, so clean sand is a necessity. If the sand is otherwise suitable for use, washing with water will remove this impurity. As to size the sand should be well graded from coarse to fine with the former predominating, all material less than ¼ inch in size being classed as sand. Many sands used in making concrete and mortar consist of particles which are small and uniform in size; these sands not only require more cement but also more work in mixing in order to coat every grain with cement. The average weight of a cubic yard of sand is about 2700 pounds.

Crushed stone or gravel is the coarse aggregate generally used and for this purpose it should be hard, clean and well graded in size. The quality of the concrete is greatly influenced by the hardness of the coarse aggregate, the hardest producing the best concrete. Comparative tests show that the relative values of the different aggregates are in the following order: Trap, granite, gravel, marble, limestone, slag, sandstone, slate, shale, cinders.

The aggregate must be clean, as any coating of dirt will not permit direct contact with the cement and will thus lower the strength of the concrete; also a stone with a chalky surface cannot produce a good concrete. The crushed stone should be cubical in fracture as it then packs easily and more closely than does flat-grained material.

All pieces less than ¼ inch in size should be screened out and used as sand, and the remainder should be well graded in sizes up to 1¼ inches. If larger than 1¼ inches they are liable to separate from the mortar when pouring the concrete. In large masses of concrete the work may be strengthened and also cheapened by dropping into the mass of wet concrete, stones as large as can be handled by one man. The average weight of a cubic yard of crushed stone is about 2500 pounds.

The water used must be free from oil, acid, strong alkalies or vegetable matter.

ESTIMATE OF MATERIALS

This is probably best illustrated by citing a particular case. Suppose that an inspection of the soil and the foundation plan shows that the volume of the required foundation is about 210 cubic feet or approximately 8 cubic yards. For machinery foundations a mixture of the materials in the proportion of one part by volume of cement, two parts by volume of fine aggregate and four parts by volume of coarse aggregate will be found satisfactory, this proportion being commonly written as a 1:2:4 mixture.

One cubic yard of concrete of this proportion will require: Cement, 5.84 bags; sand, 0.41 cubic yard; stone, 0.82 cubic yard; water, 26 gallons.

Considering a cubic yard of sand to weigh 2700 pounds and a cubic yard of stone to weigh 2500 pounds, this foundation of 8 cubic yards will require:

Cement, 8 × 5.84 = 46.72 bags.

Sand, 8 × 0.41 = 3.28 cubic yard, × 2700 = 8856 pounds.

Stone, 8 × 0.82 = 6.56 cubic yard, × 2500 = 16,400 pounds.

From the above, the finished concrete may be assumed to weigh approximately 150 pounds per cubic foot, which would make the total weight of this foundation about 31,500 pounds.

MIXING

In order to have a uniform concrete in accordance with the proportion previously decided upon it is best always to measure accurately the materials for each batch; for this purpose a 36 and 16 by 12-inch bottomless box with a capacity of 4 cubic feet will be found convenient. Also, the size of the batch should be such that it can be easily and quickly handled.

After the sand is measured, spread it evenly on a water-tight platform and then spread the cement over the sand. Mix the two thoroughly until the mass has a uniform color and then form a large crater for the reception of the water. Into this slowly pour the required amount of water and assist its absorption by slowly turning the material into the crater. Then thoroughly mix until the mass becomes a smooth mortar. Upon this mortar spread the measured amount of crushed stone or gravel, which has been previously wetted, and mix thoroughly until all the particles are coated with the mortar. If more water is needed add it from a sprinkling can. It will be noticed that the mass becomes more moist as the mixing progresses, which is an indication that the particles are being forced into closer contact.

FORMS, POURING AND FINISHING

For foundations below the surface of the ground no forms are needed except a templet to properly locate and hold the foundation bolts in position, as the concrete can be poured directly in a well made trench of the proper dimensions. In order to allow for adjustment of the foundation bolts it is well, before concreting, to surround these bolts by tin tubes. When the concrete has hardened, these tin tubes can be removed; and after the machine is placed, the holes can be filled with a thin cement mortar.

When above the ground, forms must be provided to hold the concrete until it hardens. These forms must be tight so that the cement mortar cannot run out and they must be strongly made and well tied together to withstand the side pressure of the wet concrete. The forms should be thoroughly wetted before the concrete is poured.

Immediately after the mixing is finished the concrete should be poured in the form and rammed or agitated in such a manner as to produce a dense mixture. In small jobs the work of placing the concrete should be carried on continuously until the job is completed; where this is not possible the old concrete should be well wetted before the new is added. The forms in work of this kind can be removed in from 24 to 72 hours, but after the removal of the forms the concrete should be wetted occasionally for a period of at least ten days.

The surface of the concrete after the removal of the forms will show the imprints of the wood. If this is objectionable it can be removed by bush-hammering the surface; or if a surface of exceptionally good appearance is desired the following method can be employed.

Secure screenings of gravel, crushed trap rock or granite that will pass a sieve of ⅜-inch mesh and be retained on a sieve of ¼-inch mesh. With this as the coarse aggregate make a concrete in the proportion 1:2:3 and use this as a facing mixture, placing it against the forms about 1½ inches thick, just before each batch of concrete is poured. Make only a small batch of this facing mixture and use when needed. As the facing mixture and the concrete are both of a wet consistency, they will readily unite and form a solid mass. This surface is finished by removing the forms as soon as the concrete has set and then, while the concrete is still friable, by washing with water and a stiff brush, rubbing it sufficiently to expose the aggregate of the facing mixture. This process removes all possibility of surface cracking and produces a surface resembling stone in texture.

The Watt Relics at South Kensington

By F. A. Halsey

A micrometer caliper, surface condenser, steam-engine indicator, trussed beams, gearing to replace the engine crank, models showing application of engine power, and several complete and operative engines are some of the relics that have been preserved with such intelligent care.

These form a striking demonstration of Watt's ability as a machinist, engineer and experimental investigator The importance of his work cannot be overestimated.

To an American who realizes the small number of relics which we have of the pioneers in engineering and machine construction, nothing can be more gratifying than to witness the industry and care with which such things have been collected and preserved in England. Of all those whose work we would most wish to have thus cared for, James Watt easily heads the list, and it is precisely his work of which the greatest number of examples remain. It is impossible to overestimate the importance of his work, which constantly grows on one with study. Mr. Chubb has given an account of such things as are kept at Watt's old home and place of business at Soho, and this article will show a few of those which are on view at South Kensington Museum. The man who fails to feel the impulse to uncover his head when standing before these things, deserves expulsion from the fraternity of the engine room.

Two Complete Watt Engines

Of these exhibits, the most striking are two complete and perfect examples of engines as built by Watt The first is illustrated in Fig. 1, which shows his first sun-and-planet engine built in 1788. Its cylinder is 17 inches diameter by 48-inch stroke, the mortise flywheel being 16 feet in diameter. It was nominally of 10 horsepower, but actually indicated 13¼ horsepower, figures which by their smallness are certainly striking in comparison with the size of the machine. It continued at work until 1858, its work being the driving of a metal-working shop of some sort.

The engine illustrates the natural process of transition from the pumping engines previously built by Watt, to engines for driving revolving shafts. The pumping engine, of course, had no shaft, the piston rod being connected to one end of the walking beam, while the pump rod was connected to the other, and the na-

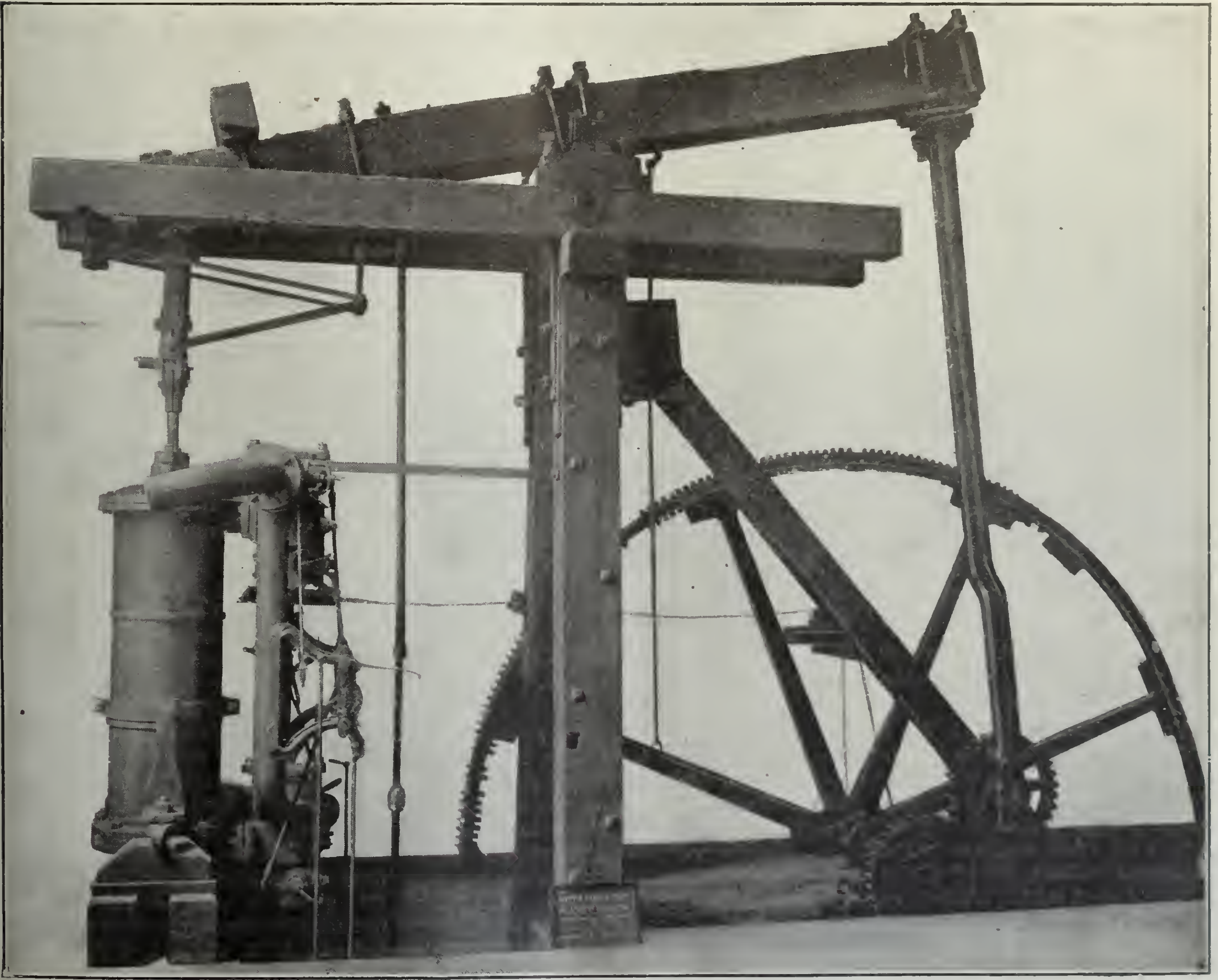


Fig. 1. James Watt's First Sun and Planet Engine—1788

[illegible] method of actuating the valve gear was by a plug rod which hung from the beam, on which suitable tappets moved the valves at the ends of the stroke. In this engine we see precisely the same gear, the plug rod serving also as the air-pump rod, the driving of the valves by means of an eccentric being still unknown. This valve gear, as will be seen, is by no means simple, and the understanding of it in all of its details is not easy without turning over the engine, which is, of course, not practicable. It is, however, clear that the direct action of the tappet was to move the valve to the closed position where it was secured by a latch, the opposite valve being simultaneously opened by the releasing of its latch and the action of a weight. The valves are of the oscillating type, although the exact construction cannot be made out without removing the chest cover. Except for the reversed action, it would require no great stretch of the imagination to see in this an anticipation of the Corliss gear. Moreover, it is a principle of the patent office that reversal of a mechanism does not constitute invention. The governor will be seen behind the valve connections driven by a chain and pulleys. It acts upon a throttle valve of the disk type. We see also Watt's beautiful parallel motion.

Fig. 2 shows another engine in the same collection, which was erected in 1797 and continued in operation until 1885. It had originally a cylinder of 16 inches diameter by 48-inch stroke, but in 1806 the makers substituted a new cylinder of $19\frac{1}{4}$ inches diameter.

The Progress of Improvement

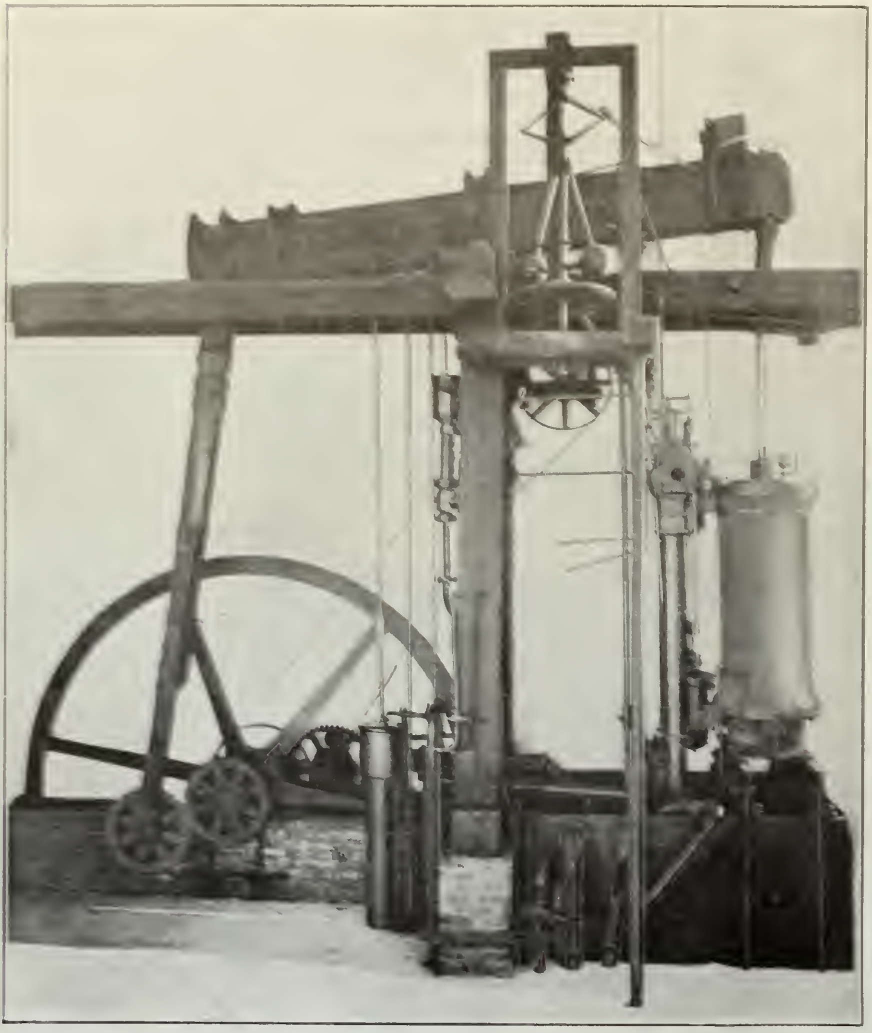


Fig. 2. Another Sun and Planet Engine Showing the Progress of Improvement—1797

In several respects this engine is a marked advance over the preceding one. The valves are driven by a crank motion obtained from a lay shaft geared to the main shaft in such manner as to reduce the speed one-half, this being necessary because with the sun-and-planet gear the shaft made two revolutions for each double stroke of the engine. The action of this crank was, of course, the same as that of an eccentric and, in view of the necessity for a lay shaft, it was a natural preliminary to the eccentric. In addition to this, the engine is fitted with a separate cut-off valve operated by a cam upon the lay shaft. This valve is located at the end of the steam pipe, one valve thus serving for both ends of the cylinder, the cam having two lobes. The connecting rod from the valve crankpin is made with a gab end, permitting it to be disengaged and the valves to be worked by a hand lever when starting the engine. The valves are of the poppet type, the steam and exhaust valve for each end of the cylinder being eccentric with one another and the stem of one passing through the hollow stem of the other.

In addition to these engines there are the main parts of a pumping engine erected in 1777, which, however, has not been photographed and which is so located as to make photographing too difficult for it to be undertaken. It antedates the parallel motion, having the horse-head-and-chain connection between the beam and the piston and pump rods.

Regarding the long terms of service of these engines, it should be said that they have been exceeded by other Watt engines, one of which, believed to be the first sold by Boulton & Watt, a pumping engine erected in 1776, continued at regular work for a period of 116 years, and was occasionally used for six years more. It is still preserved at Ocker Hill, Tipton, in working condition. It is much larger than the engines here shown, having a cylinder 33 inches diameter by 84-inch stroke, and was originally built to assist a water wheel in dry weather by pumping water from the tailrace back to the headrace. Long after the introduction of the rotative engine, this method of driving continued in use for textile mills, because of the more uniform speed when obtained from water wheels. The slide valve was invented by William Murdock, Watt's superintendent, and was employed in some cases, although Watt preferred the poppet

Watt's First Condenser

As a classical relic of Watt's work, the palm must be given to Fig. 3, which shows the original experimental model of a separate condenser. It dates from 1765 and was undoubtedly made by Watt's own hand. It will be recalled that prior to Watt's time the steam was condensed within the cylinder by the aid of water injected therein, and this model was made in order to test his first and most fundamental invention—the separate condenser. It consists of a steam-jacketed cylinder, 1.4 inches diameter, fitted with a hemp-packed piston. From the piston a rod passes downward through a stuffing box and terminates in a hook, from which a weight of 18 pounds, indicating a vacuum of 12 pounds, was suspended. Connected to the upper end of the cylinder is a vertical pipe—

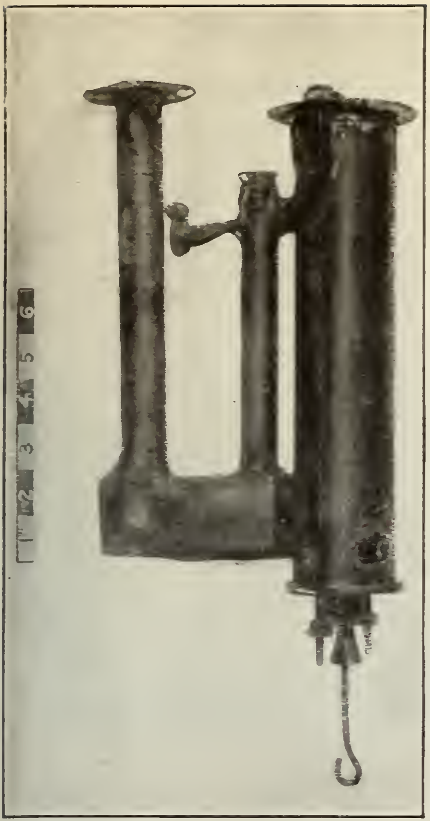


FIG. 3. ORIGINAL MODEL OF A SEPARATE CONDENSER

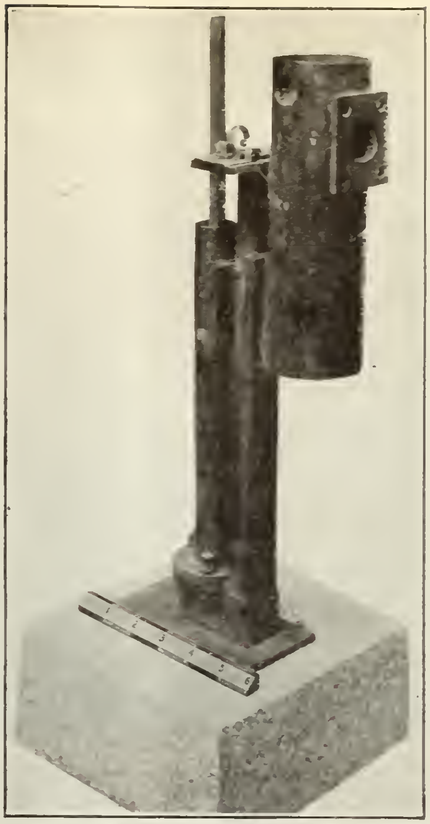


FIG. 4. ORIGINAL MODEL OF A SURFACE CONDENSER

the condenser—closed at the top and connecting below with a metal box from the further side of which passes upward a vertical tube which forms the air-pump barrel.

Most curiously, the model is now assembled in an unworkable manner, the first connection of the condenser being with the steam jacket, which in turn communicates with the lower end of the cylinder. There exists a proper connection with the upper end of the cylinder, which has been soldered over, and other features of the model show that the condenser connection must have been originally by this hole. It is surmised that after his experiments and before his patent was secured, Watt may have made this change in order to mislead anyone into whose hands the model might fall.

It should be noted, moreover, that as the model stands it is not a jet condenser, there being no opening for water into the condenser. The description states that the condenser and air pump were submerged in a tank of water, the condenser being thus of the surface type. At the same time, had there been an opening for water through the cover, it would have been easy for Watt to have removed this cover and replaced it with another in case he was bent on deceiving others. On the other hand, for the purpose of experiment, the cold surface may have given sufficient capacity without the jet.

THE FIRST SURFACE CONDENSER

Fig. 4 shows a model which is only second in interest to the one already shown. It bears a label reading "Original Model of a Surface Condenser," and was probably made shortly after the previous one, and if the previous one was originally a surface condenser this one represents a little more than an increase of surface. It contains an excellent condenser with 140 small tubes soldered into a metal tube plate at each end—the cooling water passing through the tubes and the steam to be condensed being admitted to the shell and outside the tubes.

The pipe next the condenser connects near its top with the condenser and at its bottom with the barrel of the air pump, which has a solid piston. The valves are fitted to the intermediate pipe —the delivery valve at the top of the pipe and the suction valve at its junction with the condenser—the actual pumping being thus done by the rising and falling water column—a superior construction, it will be remarked, because of its eliminating clearance.

Watt was obliged to abandon this con-

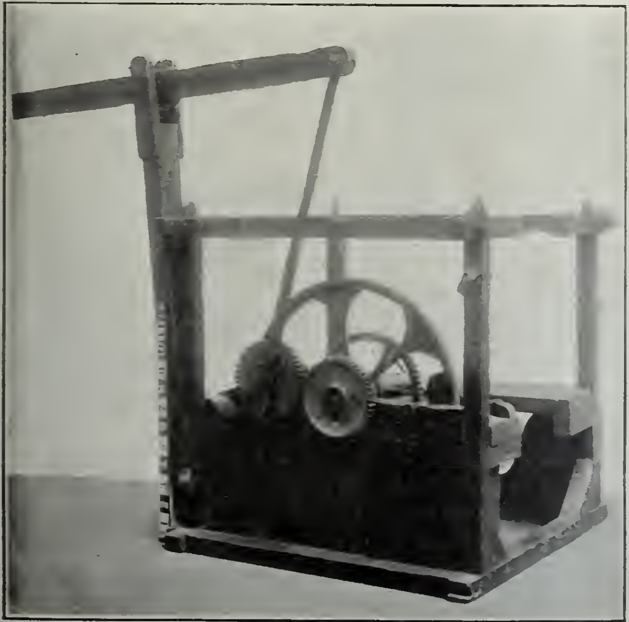


FIG. 5. WORKING MODEL OF A SUN AND PLANET GEAR

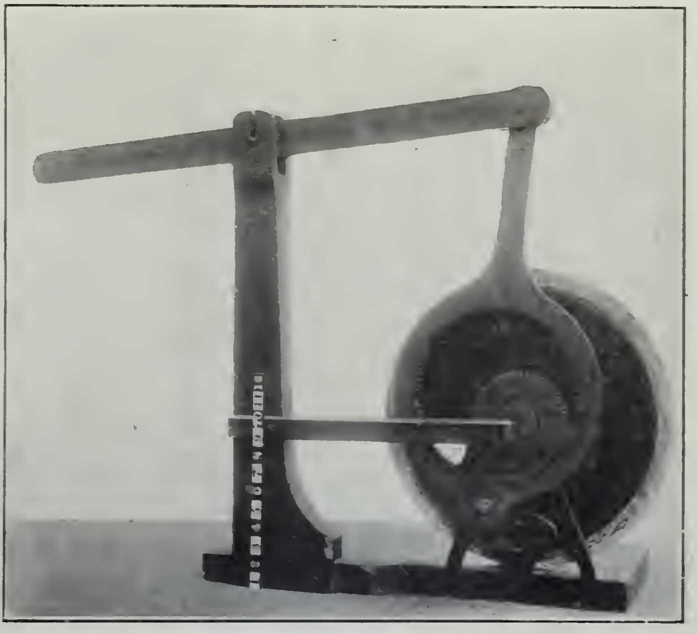


FIG. 6. WORKING MODEL OF AN INTERNAL GEAR CONNECTING ROD

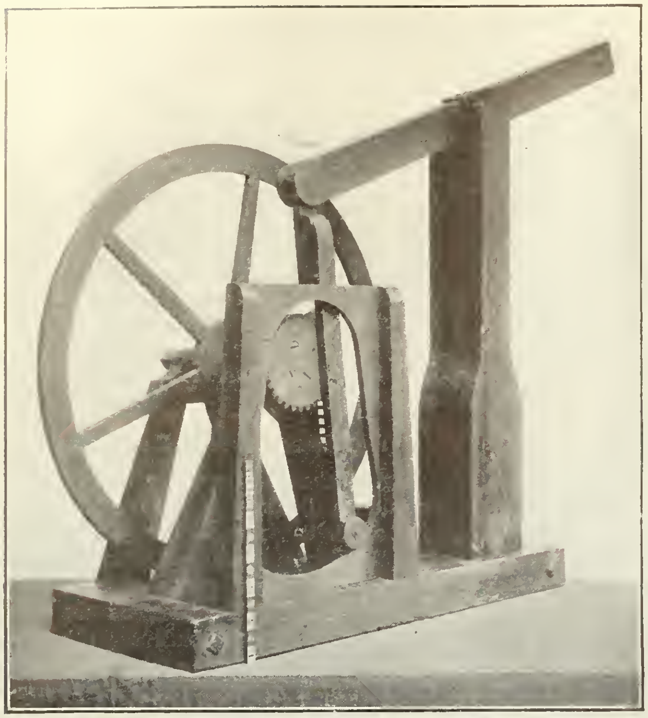


FIG. 7. MODEL OF A LADDER CONNECTING ROD

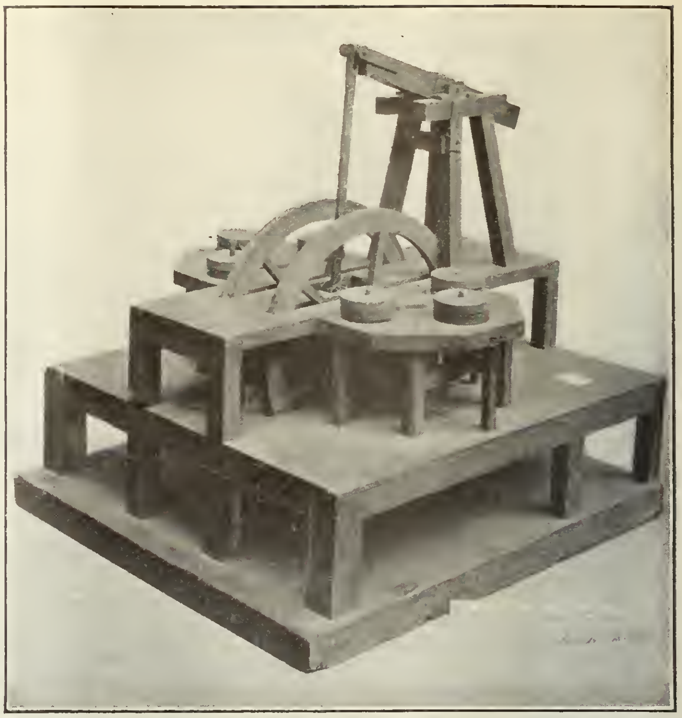


FIG. 8. WATT'S PLAN FOR DRIVING A FLOUR MILL

denser, because of the difficulty of keeping it tight—a difficulty which was not surmounted until long after his time.

STRANGE SUBSTITUTES FOR THE CRANK

Nothing seems so strange in connection with Watt's engine as the sun-and-planet gearing. The explanation of this construction lies in the fact that in the application of the crank he was forestalled by a patent taken out just in time to prevent his using it. Watt regarded the patent as a piece of piracy and refused to make terms with the patentee. In order to avoid this, he was forced to invent a substitute, and the sun and planet gear was the result. That the task was not an easy one is shown by the models of other devices, which he made before adopting this one. He was much given to model making, by the way, as were most of the old masters, and the museum contains a large number of examples other than those shown here, and of scarcely less interest. Among those included is Fig. 5, which is a working model of the sun-and-planet gear. A first cousin to this is found in Fig. 6, which is a model of an internal-gear connecting rod,* while perhaps the strang-

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*The lack of roundness of the internal gear is due to the shrinkage of the material, which is wood.

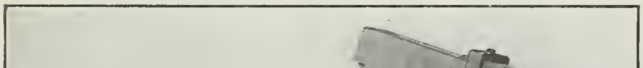

FIG. 9. WATT'S PLAN FOR DRIVING A ROLLING MILL

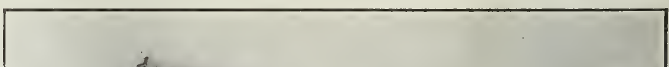

FIG. 10. A WATT MODEL ENGINE

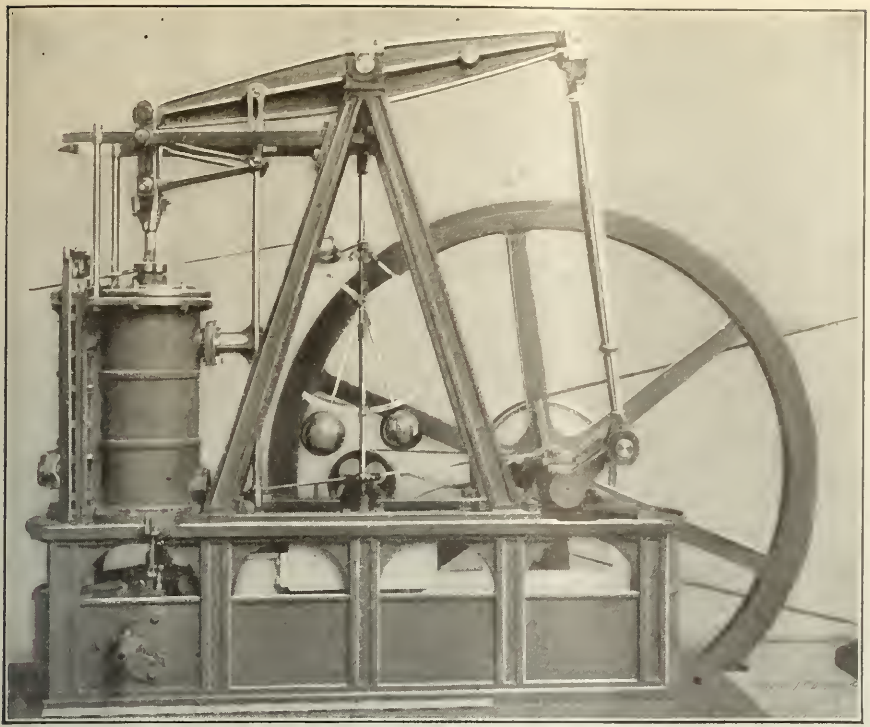

Fig. 11. Another Model Engine Left by Watt

est of all is Fig. 7, which shows a "ladder-connecting rod," than which nothing could more strikingly illustrate the straits to which Watt was put. The main shaft ends in a gear, while the connecting rod carries at its end a long rack composed of rungs like a ladder and capable of acting on either side of the gear. At the lower end of the connecting rod is a roller which works in a large opening of special form in a fixed guide plate, which keeps the rack in mesh with the gear. At the conclusion of the downward stroke, when the last rung is in mesh, the lower end of the rod swings to the other side of the gear, being guided and maintained in mesh by the guide plate, the reverse movement taking place at the end of the upward stroke. Still other devices for the same purpose are shown in model form. In spite of its unpromising features, the sun and planet gear was continued in regular use until about 1800, when the embarrassing patent ran out.

Demonstration Models

It must be remembered that Watt was engaged in the introduction of the greatest innovation that engineering has ever seen. The driving of a revolving shaft by means of steam power originated with him and he had the problem before him of convincing owners of mills that his engine would form a successful substitute for water and horsepower. In order to demonstrate his ideas, he had recourse to many models, and among these is one shown in Fig. 8, to illustrate the application of his invention to the driving of flour mills, in which service, in fact, the engine found some of its earliest applications.

The model shows the method of driving a mill having six runs of stones. The connecting rod has two sun-and-planet motions, one on each side, each of which connects to a driving shaft carrying a flywheel. A bevel gear on each flywheel shaft meshes with a second bevel on a vertical shaft, which also carries a spur wheel meshing with three pinions, one on each of the mill spindles.

Fig. 9 shows a somewhat similar model of a rolling mill, and others showing forge hammers and other machinery driven by the engine are shown at the museum.

Models of Complete Engines

While many of Watt's models had a utilitarian purpose, he also made others for his personal gratification, and, whatever one may think of models in general, he cannot be other than thankful that Watt, and for that matter others, thus left records of their work. Among several of this kind that still exist is the one shown in Fig. 10, which was probably made about the year 1800. It is of far better workmanship than the demonstrating models, being, except for the frame, of metal. Like many other models throughout the museum, a compressed-air system is connected to its own cylinder, by which it is kept constantly in motion. The valves are of the poppet type and of the same construction as in the second large engine already described, and are driven by an eccentric working through a tappet gear. It will be seen that the sun-and-planet gear has disappeared. The steam cylinder of this model is $1\frac{5}{8}$ inches diameter by $4\frac{1}{2}$-inch stroke.

The gem of the collection from the modelmaker's standpoint is shown in Fig. 11, which was bequeathed by Watt to

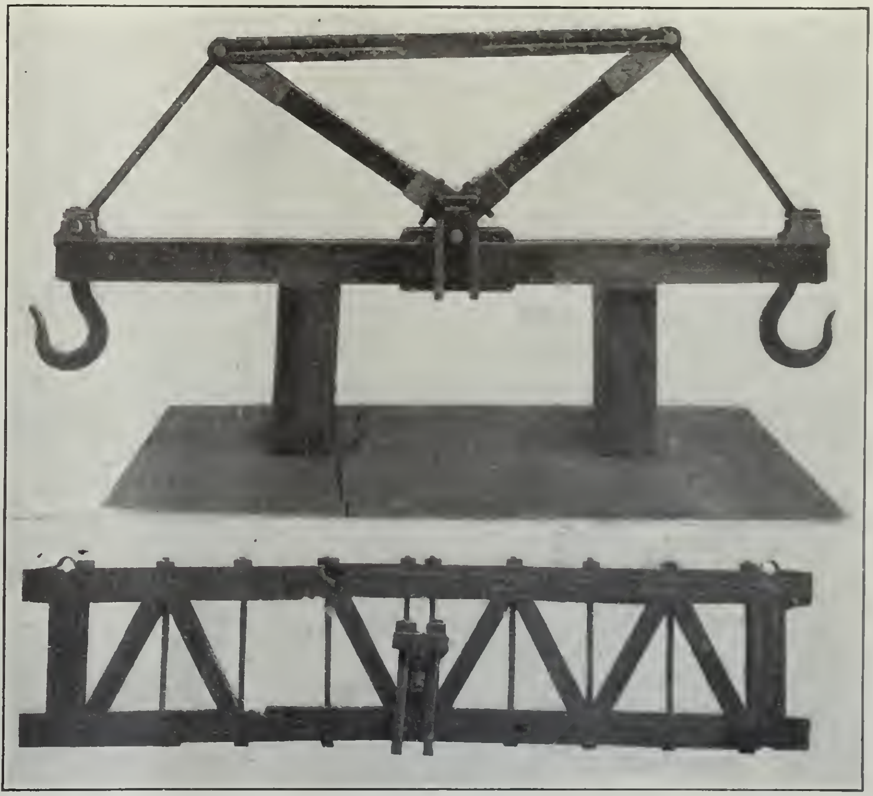

Fig. 12. Test Models of Trussed Working Beams

John Kennedy, of Manchester. It fairly represents the culmination of Watt's work and as a piece of model making it would be creditable to any model maker today. I am not informed of the size of the cylinder, but the bed plate measures about 3 feet in length overall. Unlike the last model shown and the larger engines, this one is fitted with a slide valve of the long "D" form, which is driven by an eccentric. The cylinder is steam jacketed and the engine is fitted with Watt's governor. The frame, including the bed plate, is of iron and of almost ideal form. The engine as here shown, remained substantially the standard construction for many years. The illustrations of "the steam engine," in many of our dictionaries and other books of reference might almost have been taken from this model. Like the last model shown, this one is kept in constant motion by means of compressed air turned into its own cylinder. By the belt shown it drives a modern sectional model, showing the action of valves and piston.

More Experimental Models

The condenser models will illustrate the well known fact that Watt was an experimental investigator, and the next illustration, Fig. 12, will show the same fact in a very different field of work. There is evidence that he had trouble in connection with the walking beams of his engines, and this illustration shows models of trussed beams which were made in order to determine the proper proportions of such beams. One of the models is plainly fitted for attaching test weights, while the other has been tested to destruction. It seems curious, by the way, that while Watt's earliest pumping engines were fitted with trussed beams, the beams of the mill engines which are shown were constructed of nothing but a single stick of timber.

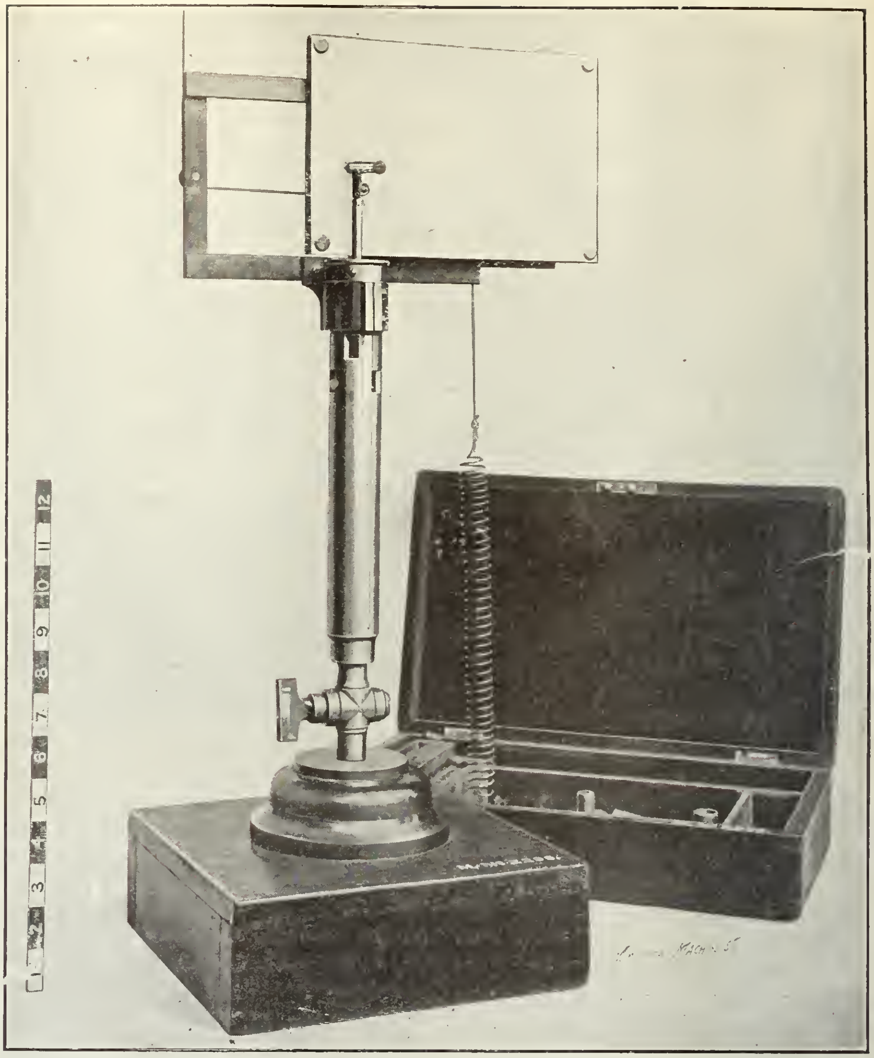

Fig. 13. A Watt Indicator

An Old Steam Engine Indicator

The steam engineer of today will look with a sympathetic eye upon Fig. 13, which represents a steam-engine indicator as used during Watt's time. The actual date of the construction of this instrument is unknown, but it was the property of Watt's agents in Manchester.

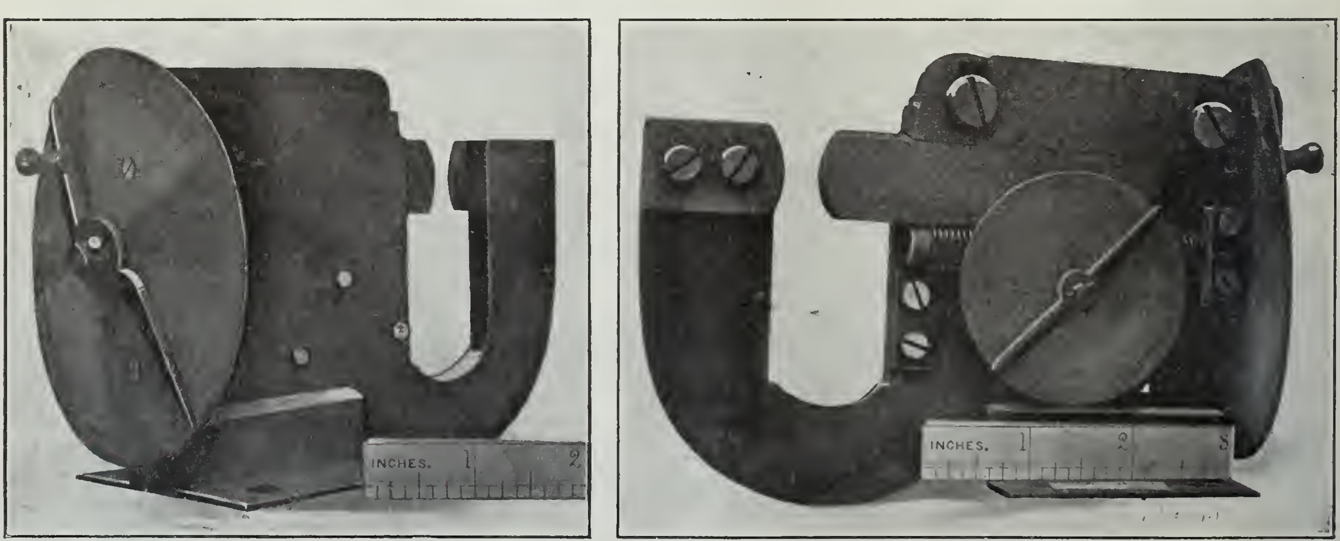

Fig. 14. Watt's Micrometer

It still carries a card which was taken in 1840, by Prof. E. Cowper. The instrument has a brass cylinder 1.13 inches diameter (1-square inch area).

THE FATHER OF MICROMETERS

If the last illustration is of prime interest to the steam engineer, the next and concluding one will be no less so to the working mechanic, for it illustrates a micrometer made by Watt. It has the regulation screw construction, although there is no nut. The screw has no endwise movement and the moving bar is fitted with teeth like inclined rack teeth, which engage the screw. The dial is stationary, the screw being turned by means of the bar which crosses the dial. Whole revolutions are counted by the index on the smaller dial, this index being turned by means of a small worm wheel, which meshes with the screw. The screw has 18 threads per inch, while the dial has 50 main divisions, which are subdivided into halves. The main divisions thus read to the 1/900 of an inch, while the subdivisions read to the 1/1800 part.

The Hudson Terminal Buildings*

By W. F. West

> *Two mammoth buildings in New York City which contain 4000 offices and accommodate 9000 people. During the warm months current for light and power is furnished by the Jersey City power house of the Hudson & Manhattan Railroad Company at a cost of less than one-half cent per kilowatt-hour. During the cold months, when the steam-heating system is in service, the plant at the buildings furnishes the current for even less than this cost.*

The Hudson and Manhattan Railroad Company operates trains through tunnels which connect New York City with about all of the railroad terminals on the New Jersey side of the Hudson river, as well as the electric car line which approach the New Jersey shores. In the downtown section of New York City, the system terminates in two mammoth office structures occupying two whole city blocks, bounded east and west by Church and Washington streets, and north and south by Fulton, Dey and Cortlandt streets. The buildings extend 22 stories above the sidewalk and four below.

In these buildings almost any conceivable thing from a railroad or steamship to tacks and birdseed may be bought or sold. Stores fronting on street, concourse and arcade, are occupied by druggists, automobile agents, jewelers, clothiers, hardware dealers, confectioners, butchers, fruiterers and florists. One of the buildings contains a bank.

The buildings contain the equivalent of 4000 offices, which are lighted on dark days by the equivalent of 30,000 forty-watt, 32-candlepower Tungsten lamps. There are 39 Otis traction passenger elevators. Of these, 22 are expresses, their first stop being at the eleventh floor. In addition to these, there are seven hydraulic elevators for baggage, etc., which are located below the street level. The elevators handle about 40,000 people in and out, per day. The buildings contain about 9000 occupants at present; they are about 87 per cent. occupied. About 50,000 people visit these buildings per day, including the stores. A total of about 120,000 people visit the buildings per day, including the concourse floor; of this number about 70,000 are passengers on the road. The road, according to the records, carried an average of 85,846 passengers per day during the month of August, 1909, and an average of 141,871 passengers during the month of March, 1910. This shows an increase in traffic of about 65 per cent. in seven months. The running time between Jersey City and the Hudson Terminal buildings is three minutes; between Hoboken, N. J., by way of Jersey City to the Hudson Terminal buildings it is ten minutes.

The power house for operating the electric trains is situated in Jersey City. This location affords considerable advantage, particularly in the cost of coal, as it is obtained direct from the mines. This eliminates the cost of transferring it across the river and trucking. The plant is equipped with Babcock & Wilcox boilers, Curtis turbines and General Electric generators. During the warmer season when little or no steam is required for heating, the current for lighting the buildings is obtained from the Jersey City power house. The 11,000-volt alternating current furnished by the power house is converted in an electrical substation at the building to 240 volts direct current for use in the building.

The steam-electric plant for the buildings contains 1500 horsepower of Babcock & Wilcox boilers, fitted with McClave dumping grates, having about 29 per cent. of air space. No. 3 Buckwheat coal, commonly known as barley, is burned. The feed water is put through a Ward feed-water filter and a Wainwright heater. The boilers are fed by steam-driven Worthington or electrically driven Dean pumps, or by 2½-inch Metropolitan injectors. The generating equipment consists of two 400 horsepower tandem-compound noncondensing Rice & Sargent Corliss engines direct connected to 300-kilowatt double-voltage 120- and 240-volt direct-current Western Electric generators. The plant also contains a 144-cell storage battery, with a capacity of 2000 amperes for one hour.

The auxiliaries are as follows: Two steam-driven Worthington pumps and one electrically driven Blake pump for pumping water from the Loomis & Manning filters to the house-service tanks on the roof. One steam and one electrically driven Clayton air compressor for supplying air to six Ansonia sewage ejectors. Two vacuum pumps for the vacuum-cleaning system, two small electrically driven Crocker Wheeler air compressors for pumping air into the hydraulic-elevator tank at a pressure of 300 pounds per square inch. Two Laidlaw-Dunn-Gordon air compressors driven by Westinghouse motors for furnishing compressed air to the road in emergency. Three Dean triplex pumps driven by General Electric motors for the seven hydraulic elevators. Two fresh-air fans driven by General Electric motors, and two exhaust fans driven by Northern Electric motors. One vacuum pump built by the American Steam Pump Company, for the vacuum-heating system. One 80-ton Carbondale refrigerating system served by steam and electrically driven ammonia pumps and electrically driven brine pumps. Brine is carried at zero Fahrenheit and is pumped from the basement to the top floor, 26 stories, under a pressure of 215 pounds per square inch. There are 61 refrigerator boxes throughout both buildings to which brine is furnished.

The heating system contains 140,000 square feet of radiation which takes from 1400 to 1500 boiler horsepower per hour on a cold day. At this rate 60 tons of coal are required per day to heat both buildings. The average evaporation secured is 7.9 pounds of water from and at 212 degrees per pound of coal.

As the capacity of the Jersey City power house is quite large for the requirements of the road at present, and can furnish current for the buildings easily, the question is, which plant is to furnish the current for the buildings? The answer is, the one that can do it the cheaper. The next question is, how cheap can the power house furnish current to the buildings, and not how cheap can the power house furnish current to the road? The road is another proposition. The answer is 3.5 mills, or 0.35 of a cent per kilowatt-hour. This figure was obtained from the additional amount of money which it cost the power house to generate the extra current minus the cost of labor to operate that part of the electric equipment in the Hudson Terminal buildings which could be dispensed with. This figure may seem impossible, but it is not necessary to run another unit to supply the load, which averages from 14,000 to 17,000 kilowatt-hours per day. The only additional cost is that of the extra coal, the handling of same, and a little extra in the way of supplies. This extra load on the power house helps to steady the machines serving the road, the requirements of which fluctuate considerably. The figure for the plant at the buildings is too high in the summer time to think of operating the generating equipment. It is 7.5 mills per kilowatt-hour. This figure is obtained by putting the cost of the auxiliaries where they belong. For instance, in the summer time, with the electric plant shut down, the only expense is that of the auxiliaries, the refrigerating plant and the steam for the kitchens. For an additional sum of $95 to $100 per day the electric plant can be run and from 13,000 to 14,000 kilowatt-hours per day can be generated. But even this low cost is too high to warrant the use of the generating equipment.

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*Abstract of paper read before New York No. 24 N. A. S. E. (Phœnix Association).

In the cold season, as long as the buildings require all of the exhaust steam for heat, the generating equipment can be operated for $21.11 per day, regardless of the amount of current generated. This sum covers extra labor and supplies, such as oil, waste and packing, and three-quarters of a ton of coal. This amount of coal is required to keep the cylinders hot or to evaporate 13,400 pounds of water from and at 212 degrees, the amount of water which is lost from the receivers and muffler tanks per day. This does not mean that the engines do not take 22 pounds of water per horsepower per hour, but it does mean that with a vacuum in the heating-system return line, the water that passes through the engines is in the form of steam and remains steam practically until it leaves the radiators. In other words, to all practical purposes, after passing through the cylinders, it is just as good at one-quarter to one-half pound pressure as is live steam reduced to one-quarter to one-half pound pressure through reducing valves. Reducing valves regulate the pressure in the heating main. When the electric plant is operated, the reducing valves let in enough live steam to the heating system to met the requirements. The more load on the engines the more live steam the valves shut off or *vice versa*. Hence, the more kilowatts generated, the cheaper they are. As there is not enough load for both of the engines through the night, the storage battery is charged at night and discharged during the day.

The minute the atmospheric relief valve, which is set for 1¼ pounds pressure, opens, it affects the cost of the current, for then the amount of the coal and the labor it took to produce the steam that goes out at the roof must be changed against the kilowatts. This is how the cost of the current is figured and as soon as the cost goes to 0.0035 dollar per kilowatt-hour, the electric plant is shut down for the summer.

Construction of Carroll's Gas Engine

In a recent issue we said that a search of the patent records showed that John E. Carroll, who proposes to run the industries of the country with a soda-water bottle, had no patents, excepting one upon an insulated pipe and a freak motor boat. Our search was of the United States patent records. We have since learned that he holds an English and a Chinese patent for "Motive Power Apparatus" which afford a glimpse into the supposed *modus operandi.*

> *The British patent office drawings of Carrol's carbonic acid engine, which are presented in this article show how impractical and absurd are the inventor's claims.*

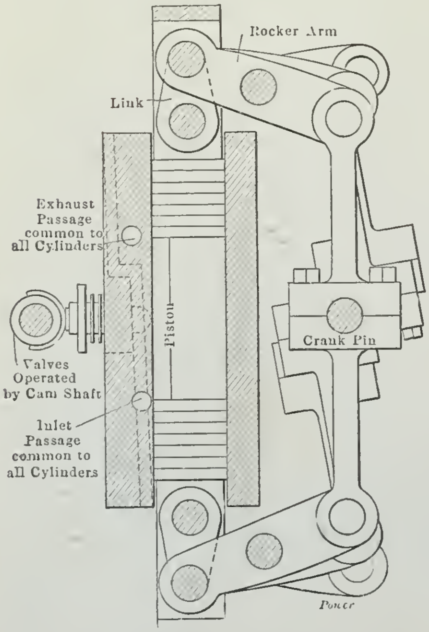
Rocker Arm
Link
Exhaust Passage common to all Cylinders
Piston
Crank Pin
Valves Operated by Cam Shaft
Inlet Passage common to all Cylinders
Power

Fig. 1. Section through Cylinder

Fig. 3 is an elevation of the engine, and Fig. 1, a section through one of the cylinders. The engine has three cylinders, each containing two pistons which are forced in opposite directions by the admission of carbonic-acid gas. Each piston is connected through a link, rocker arm and connecting rod to the crank shaft. The six cranks are set at different angles to equalize the torque. The valves are controlled by a cam shaft operated through gearing from the main shaft.

The exhaust is led into an exhaust or vacuum tank, which is shown in Fig. 4. This is the mysterious "regenerator," and the following description is taken from that in the patent record: "The exhaust pipe leads directly into the inner chamber *A*, adapted to hold a definite quantity of gas. Space *B* is a vacuum surrounding chamber *A*. *C* is a gas chamber connected with *A*; this is also inclosed by a second vacuum compartment *D*, so that when the gas enters *C* there is no interchange of heat by reason of the insulated walls of the vacuum chamber. Additional gas chamber *E* and vacuum chamber *F* are provided, so that when the gas eventually reaches *E* it has passed through all the gas chambers and around the numerous surrounding vacuums at various stages and is then in a *completely liquefied state, owing to the expansion it has undergone.*

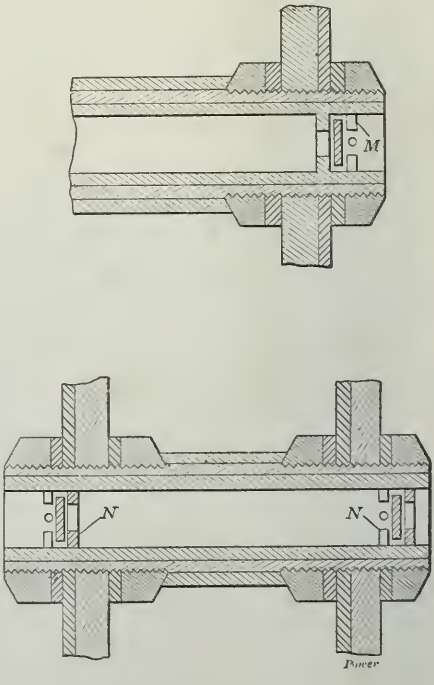

Fig. 2. Check Valves in Exhaust and Outlet Pipes

The exhaust pipe is provided with a check valve shown at *M*, Fig. 2, and the outlet pipes from the various gas chambers contain check valves as shown at *N*, which allow the gas to pass outward. The outlet valves are of *progressively increasing resistance in order to compensate for the increase in pressure of the gas as it passes from one chamber to another.*

The liquid gas, after leaving the exhaust tank, is conducted to a charging tank, shown at *T*, Fig. 4, finding its way through a body of petroleum, represented by *P*. *The liquefied gas, being heavier than the oil, drops below it and then oozes through the oil* into the body of the tank in a gaseous state under considerable pressure, any water that may have been in the gas remaining under the oil. As the quantity and pressure of the gas in the charging tank increase, the supply from the exhaust and liquefying tank ceases until the gas is liberated for use.

At *S* in the charging tank is located a cage containing salt. The gas, after leaving the oil, passes through the salt, which extracts all moisture and renders the gas anhydrous. The gas is then conducted to the engine."

Mr. Carroll seems to have taken great liberties with the laws of nature, making them assume all sorts of forms and shapes to suit his purpose. The "law for the expansion of gases" has been turned "upside down" and taken the form that a gas at a constant temperature will increase in pressure with an increase in volume. Also through the valves the gas is made to flow against pressure. Further, we should like to see carbonic acid gas liquefied by expansion alone. The absurdity of the whole scheme is evident, at a glance, to anyone at all familiar with the fundamental laws of physics, and the efforts to place such a device upon the market is, to say the least, amusing.

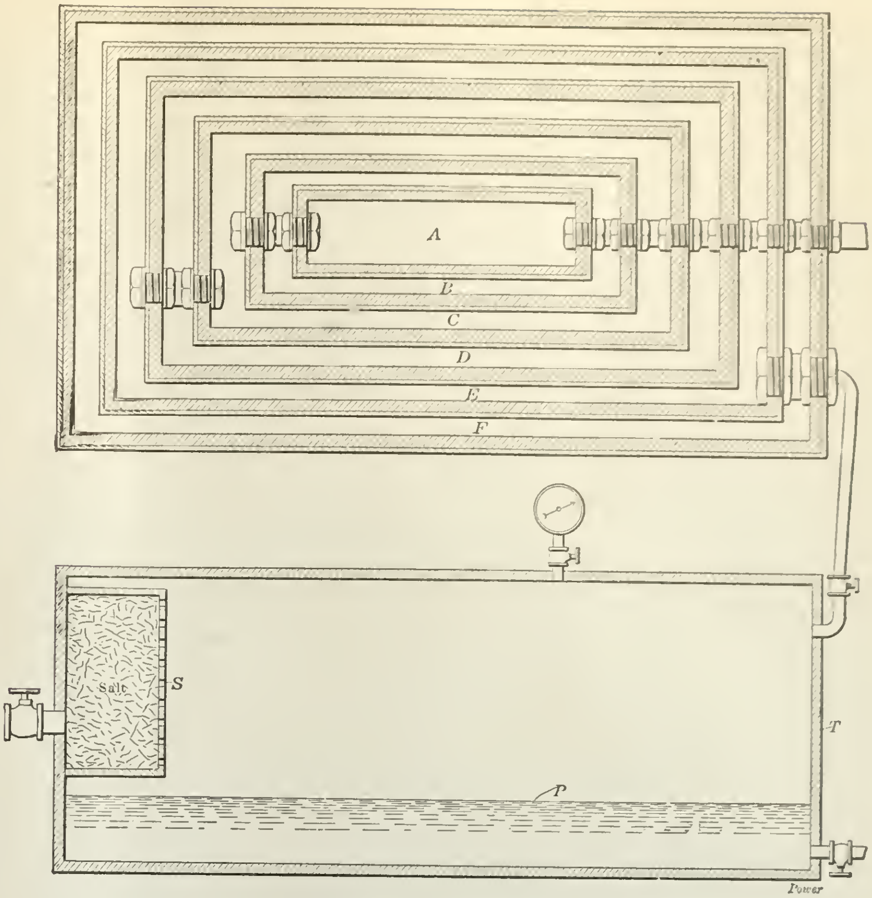

FIG. 4. REGENERATOR AND CHARGING TANK

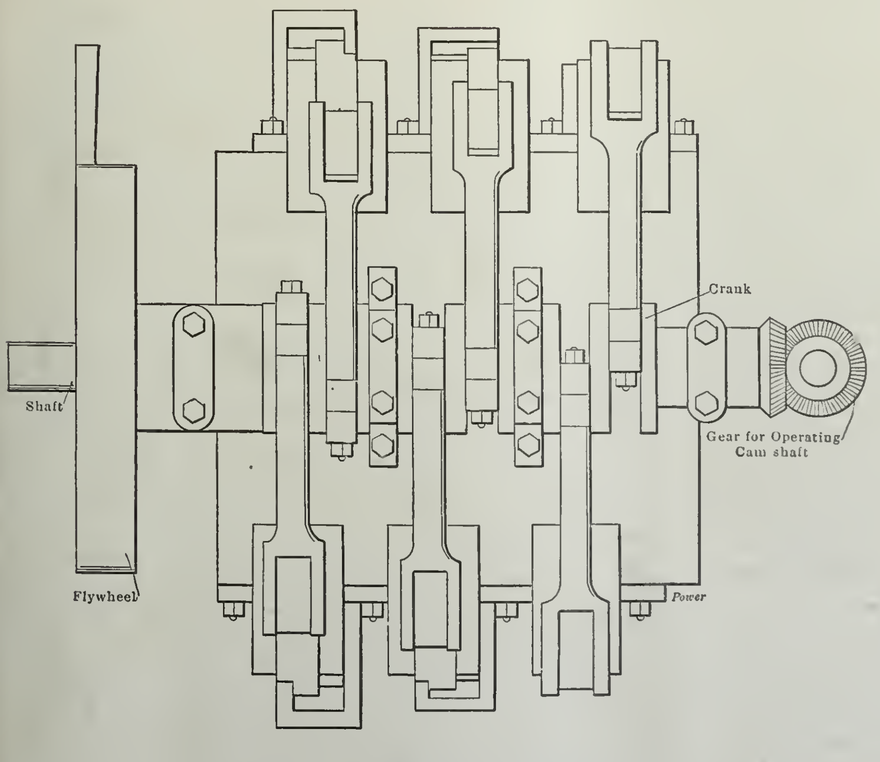

FIG. 3. SIDE ELEVATION OF ENGINE

Lignite for Central Station Fuel

Brown coal or lignite, the production of which in Germany now exceeds 60,000,000 tons per annum, is to be used on a large scale in connection with the supply of electricity to the city of Cologne. According to an agreement which has been concluded between the Municipal Council and the Rhenish company for lignite mining and briquet manufacture, the latter has undertaken to supply high-pressure three-phase current to the city in sufficient quantity to fulfil all purposes, beginning April 1, 1912. The present municipal works and those of districts to be incorporated can be continued and improved by the city, which is also at liberty to close them either wholly or partly and obtain the entire supply from the company. The agreement is for a period of 30 years. The charges will be according to a sliding scale of maximum rates which cannot be increased, but which are susceptible of reduction if the cost of production is lowered by the introduction of new inventions during the period of the agreement.—*Exchange.*

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Primer of Electricity

Brush Contact Resistance

Besides passing through the resistance of the armature winding, the armature current must pass between the brush faces and the commutator surface, and through the metal of the commutator bars. It is not practicable to figure the loss due to the resistance of the commutator bars, and this is so small that it is not worth figuring, if the commutator has been well proportioned. The loss at the brush faces, however, is considerable.

On all except a few special types of dynamos and motors the brushes are made of carbon, for reasons which will be explained in a future lesson. Now, as explained in one of the early lessons, the resistance of carbon varies very much with changes in its temperature, the resistance *decreasing* as the temperature increases. This behavior affects the resistance of the contact between the brush and the commutator somewhat as though the current were passing through carbon entirely; that is, the more current passes between the brush and the commutator, the lower is the resistance of contact, due to the heating of the carbon by the increased current. This decrease in resistance as the current increases has the advantage of preventing the loss due to the contact resistance from increasing as rapidly, with increasing current, as it does in the armature winding and other electrical circuits. That is, the voltage drop at the brush faces does not increase in the same proportion as the current, but very much slower—in fact, it is almost constant, with a certain quality of carbon in the brush and a certain range of current density at the brush face (the current density is the number of amperes passing between the brush and the commutator per square inch of brush-face area).

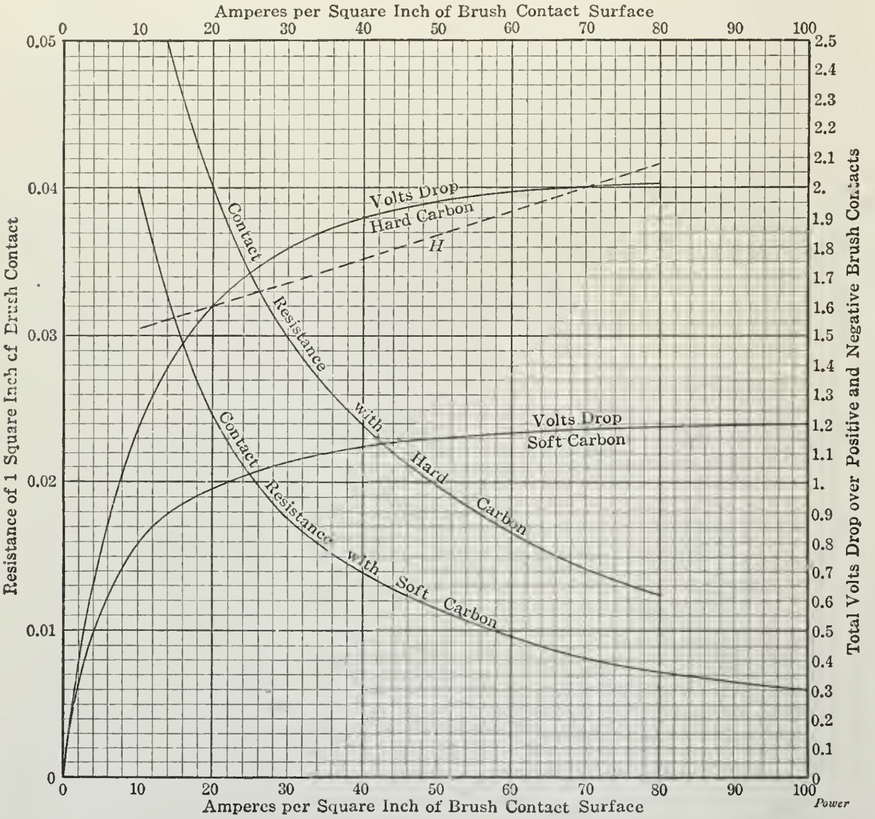

Fig. 55. Resistance of Brush Contact and Resultant Voltage Drop

The quality of the carbon used in brushes varies widely from a very soft, graphitic carbon to a grade so hard that it feels gritty to the fingers. The softer the carbon, the lower its resistance and, consequently, the lower the resistance of the contact between the brush face and the commutator. The voltage drop at the brush face, therefore, is much less with soft-carbon brushes than with hard ones, and the loss in watts is, of course, correspondingly less. From this it would seem desirable to use brushes just as soft as can be obtained, but that is not true in most cases; where the voltage is very low and the current large, soft carbon is commonly used, but it is advisable to use brushes as hard as possible, within certain limits, for the same reasons (to be explained later) that carbon brushes are used at all, when copper brushes would give so much less loss.

Another peculiarity of the carbon brush is that the direction of current flow across the contact affects the relation between increase of current and decrease of contact resistance. When the current flows from the commutator into the brush, the contact resistance decreases much less rapidly, with increasing current, than when the current flows from the brush into the commutator. In some tests made in 1907 by Prof. E. Arnold, a distinguished German scientist, it was found that the drop at the positive brush of a dynamo decreased from 1 volt at a temperature of 104 degrees to 0.8 volt at 168 degrees while the drop at the negative brush decreased from 1 volt to 0.2 volt with the same temperature increase and current.

These various effects are very erratic, and it is impossible to give any rules or formulas that would enable anyone to estimate closely what resistance effects will be obtained with the many different grades of carbon. Besides, the mechanical conditions, such as roughness of the commutator surface, bouncing of the brushes, vibration of the holders in which the brushes are mounted, etc., all affect the effective contact resistance of the brushes.

Merely for the purpose of giving the student a general idea of the relation between the current density at the brush faces and the resistance of the contact with the commutator, the curves in Figs. 55 and 56 are presented. These give the average of a number of tests made with soft and hard carbon brushes of grades commonly used on dynamos and motors in this country. Fig. 55 looks a little complicated, but if the reader will follow the explanation carefully, the complications will straighten out without any effort.

The curve marked "Contact Resistance with Hard Carbon" gives the average relation between the current density at the brush faces and the resistance of the contact between the brushes and the commutator, as obtained from the tests of a number of hard-carbon brushes. For example, with 20 amperes per square inch of brush-face contact, the resistance of one square inch of contact was 0.04 ohm (trace upward along the line marked 20 at the bottom of the diagram until you strike the curve, and then trace horizontally to the left-hand scale). At 40 amperes per square inch of brush contact, the resistance of the contact was only 0.024 ohm, and at 60 amperes per square inch it was 0.0165 of an ohm across one square inch of contact. From this it is evident that the more current there is going across between the brush and the commutator, the lower is the resistance of the contact. It would seem, therefore, that the proper way to work a dynamo or motor is to jam just as much current through the brush faces as possible, or, putting it the other way, to make the brush faces just as small as mechanical strength will allow. This is not true, however, because, while the contact resistance decreases with an increase in current, it does not decrease fast enough to prevent the increased current from increasing the watts lost at the contact, and this means increasing heat, which will ultimately cause trouble. If the brush faces get too hot, a soft brush will soften and deposit a sort of carbon paste on the commutator; a hard brush will also soften at a high temperature, but before that is reached it will crumble off in minute particles which scratch the commutator and lead to various other troubles. Even if no operating troubles were produced, the increase in watts lost at the brush contacts makes it unwise to carry the current density very high.

To return to the diagrams, the curves in Fig. 55 serve to illustrate the difference in the nature of the contact when soft and hard brushes are used and the results produced by changes in the current density at the brush faces. The curve marked "Contact Resistance with Soft Carbon" shows that at 40 amperes per square inch of brush-face area, the contact resistance for one square inch of face area was 0.014 ohm, at 50 amperes density it fell to 0.0115 ohm and at 64 amperes it was only 0.009 ohm.

It is the voltage drop and the watts lost, however, that are of practical interest. The curves marked for voltage drop with soft and hard carbon, in Fig. 55, illustrate the statement that the voltage drop remains almost constant within a certain range of current densities. For example, the total drop over both brushes (that is, the drop at the positive brush added to the drop at the negative brush), with soft carbon, is 1.12 at 40 amperes density, 1.15 at 50 amperes and 1.152 at 64 amperes per square inch. With hard carbon, the drop across both brushes is 1.6 volts at 20 amperes density, 1.8 volts at 30 amperes and 1.92 volts at 40 amperes per square inch.

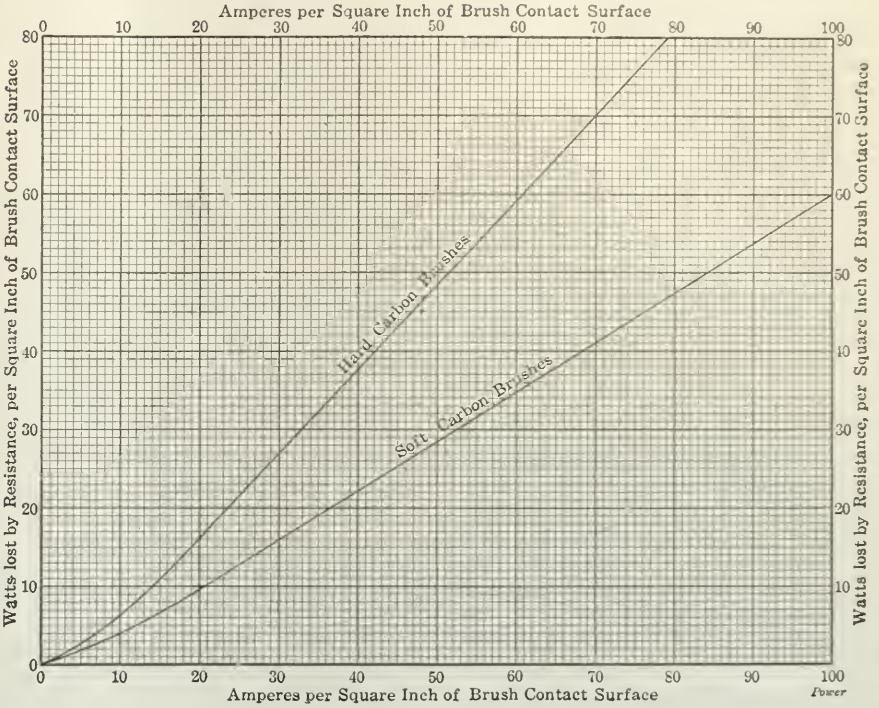

Fig. 56. Average Electrical Loss at Carbon Brush Faces

The curve of Fig. 56 was plotted from the same tests that served for the curves just discussed. Taking the same densities as before, it will be found that the watts lost with soft brushes were 22 per square inch at a density of 40 amperes per square inch, 28½ watts at 50 amperes density and 37 watts at 64 amperes density. With hard carbons, the watts lost per square inch at 20 amperes density were 16; at 30 amperes density, the loss was 27 watts per square inch and at a density of 40 amperes per square inch, the loss went up to 38 watts per square inch. These figures demonstrate what was said as to the increase in losses with increased densities. Suppose, for example, that a machine was equipped with hard-carbon brushes having a total face area of 12 square inches and that the total armature current was 120 amperes. The positive brushes would have a total face area of 6 square inches and the negative brushes the same; therefore, the current density at the brush faces would be

$$\frac{120}{6} = 20$$

amperes per square inch of brush-contact area. The watts lost per square inch of contact area would be 16, and the total loss at all brushes would therefore be

$$16 \times 12 = 192$$

watts. Now suppose that one-third of the brushes were taken off, leaving a total brush-face area of 8 square inches; then the current density would be increased to

$$\frac{120}{4} = 30$$

amperes per square inch. The loss per square inch would be increased to 27 watts, and although the contact area had been reduced to 8 square inches, the total loss would be increased to

$$27 \times 8 = 216$$

watts, as compared with 192 before. Now suppose one-fourth of the remaining brushes were removed, reducing the total brush-face area to 6 inches. The current density would be increased to

$$\frac{120}{3} = 40$$

amperes per square inch and the loss would be increased to 38 watts per square inch, making a total loss of

$$38 \times 6 = 228 \text{ watts.}$$

It may not be quite clear to the student why the curves in Fig. 56 are plotted to show watts per square inch of brush-face area, while the voltage-drop curves in Fig. 55 give the total volts without regard to contact area. The reason is that the resistance of the contact decreases as the area increases, because the resistance of all electrical conductors follows that law. Thus at 30 amperes density, a brush having a face 1 inch square would have a contact resistance of about 0.03 ohm; the current would be 30 amperes (the contact area being exactly one square inch) and the drop would be 0.9 volt at the positive brush and the same at the negative brush (assuming for the moment equal contact resistances). The watts lost at each brush face would be

$$30 \times 0.9 = 27.$$

Now suppose that the brushes (one positive and one negative) had only ½ square inch face area each and the current were 15 amperes. The density would remain 30 amperes per square inch

$$(15 \times 0.5 = 30)$$

and the resistance of contact would be twice what it was before, because the area has been reduced one-half; the resistance at each contact, then, would be 0.06 ohm and the voltage drop would remain 0.9 as before, because

$$0.015 \times 0.06 = 0.9.$$

But to keep the current density at 30 amperes per square inch, the current was halved when the brush contact was halved; therefore, the watts lost at each brush face would be

$$15 \times 0.9 = 13.5,$$

or just half what they were before. The watts lost *per square inch* of brush face, however, would be the same. In the first case there were two brushes each having a square inch of face area, or a total of 2 square inches, and the watts lost were 27 at each brush, or 54 at both; this is

$$54 \div 2 = 27$$

watts per square inch of brush-face area. In the second case there were two brushes of ½ square inch face each, or a total of 1 square inch at both brushes; the watts lost per brush were 13.5, making 27 watts at both brushes, and the total area being 1 square inch, the watts per square inch were obviously 27.

In the explanation it has been assumed that the contact resistance of a positive brush is the same as that of a negative brush. That is not always true, but it does not affect the results because it is the total drop at both sets of brushes and the total watts lost at both sets that are considered, and these total results have been merely averaged between the two sets of brushes to simplify the computations. For example, it makes no difference whether at 30 amperes density the drop at the positive brush is 1 volt and that at the negative brush is 0.8 volt, or whether the positive brush drop is 1.5 volts and the negative brush drop 0.3 volt, so long as we know that the total drop over both brushes is 1.8 volts.

The student is again reminded that while the resistances, drops and losses indicated by the curves in Figs. 55 and 56 may come close to those obtained in practice with average hard and soft carbon brushes, they are not supposed to give hard and fast values—only to illustrate the general performance of carbon brushes. It is possible to get very different figures from the same brushes on two different machines or even on the same machine under different running conditions.

Turbo-generators of 18,000 Horsepower for the Victoria Falls & Transvaal Power Company

The Allgemeine Electricitäts-Gesellschaft recently installed five 18,000-horsepower turbine-generator units at the Rosherville Dam power house of the Victoria Falls & Transvaal Power Company, in South Africa. The specified continuous output of each unit is 12,000 kilovolt-amperes, or 9600 kilowatts with a lagging load of 80 per cent. power factor. The windings of the stator are connected in star or Y fashion, and the potential between terminals is 5000 volts. The frequency is 50 cycles and the speed 1000 revolutions per minute. The stator is equipped with bar-fork windings. The exciters are direct connected and deliver current at 220 volts.

Each generator delivers to a step-up transformer of equal full-load rating, capable of raising the voltage to 20,000 or 40,000.

The guaranteed steam consumption is 5.8 kilograms (12.8 pounds) per kilowatt-hour with a power factor of 80 per cent., steam at 205 pounds per square inch, superheated to 330 degrees Centigrade (626 Fahrenheit), and cooling water at 15 degrees Centigrade (59 Fahrenheit) for condensing.

LETTERS

Effects of a Bad Joint in the "Field" Lead of a Compound-wound Dynamo

The engineer of a small plant sent in a hurried call for a man to locate the trouble with one of his generators and I was sent out on the job. The plant consisted of three 50-kilowatt 125-volt compound-wound dynamos direct connected to gas engines running on producer gas.

The engineer informed me that one of the dynamos refused to carry its share of the load when running with either of the other two. Moving the field rheostat of the faulty machine had little effect except to raise the busbar voltage. The field windings were tested for grounds and short-circuits and found to be all right. The german-silver shunt was carefully examined, the brushes were in the correct position and everything seemed right except for a dark blue discoloration on the commutator. The engineer said this always appeared after a machine had been shut down for several hours, and was caused by the producer gas from a leaky pipe attacking the copper.

After some thinking it became apparent that if the positive lead should be open somewhere between the generator and the busbar the trouble would be accounted for, since any current flowing through the armature of the lagging machine would have to pass through the series field winding of the machine with which it was working in order to reach the positive busbar, thereby strengthening the field of the other machine instead of its own.

Acting on this theory, the three-blade switch of the faulty machine (No. 2 in the sketch) was closed, the circuit-breaker being open and the machine standing still. In this case No. 1 was already on the line and closing the main switch of No. 2 put the series winding of No. 2 in parallel with that of No. 1. This should have diverted half the current from the No. 1 series field winding and reduced the voltage. But there was no

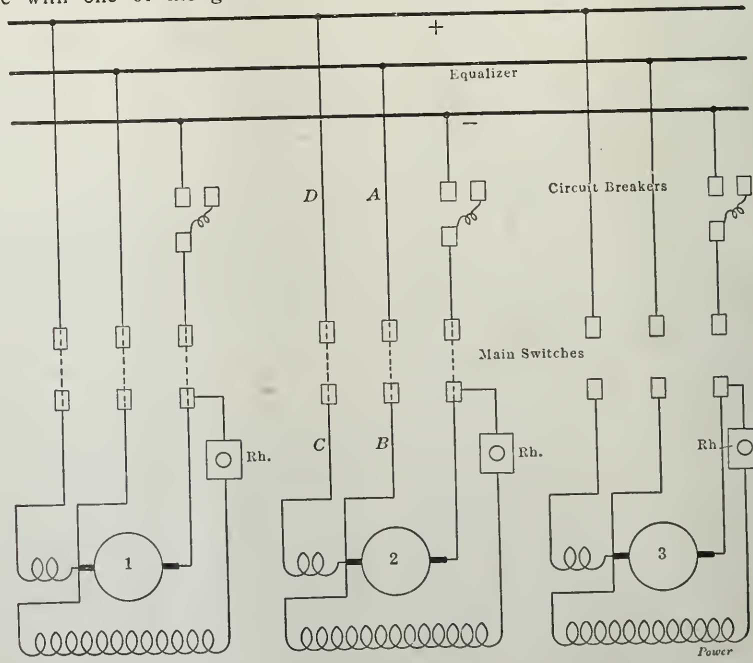

Fig. 1. Diagram of Wiring Connections of Compound-wound Dynamos

effect on the voltage, thus indicating that no current flowed from the equalizer busbar to the positive busbar through the circuit *A, B, C, D.* With the aid of a lamp line it was found that no current would pass through the joint between the positive adapter stud and the busbar, and when the stud was unscrewed it was found to be corroded. The joint had never been actually tight and the action of the gas from the leaky pipe had gradually reduced the contact surface until it formed an open circuit. After this joint was cleaned the trouble disappeared.

Had this happened to the equalizer connection instead of the positive busbar connection, the first symptom of trouble would have been fireworks on attempting to parallel the machines. After this was made clear the engineer took steps to correct the leak in the gas pipe.

A. BODDIE.

Boston, Mass.

Speaking of smoking shunts reminds me of an experience somewhat similar to that of Mr. Roberts, described in the May 31 issue. We had two 500-volt street-railway generators running in parallel and connected as shown by the accompanying diagram. There was no trouble as long as the load was light, but just as soon as the heavy load came on, the shunt strip of No. 1 machine would smoke. It was several days before we could locate the trouble as it did not appear every day. It was discovered by the blade and jaw *P* of the No. 2 main switch getting hot and changing color when the load came on; the minute the load went off the blade was cool.

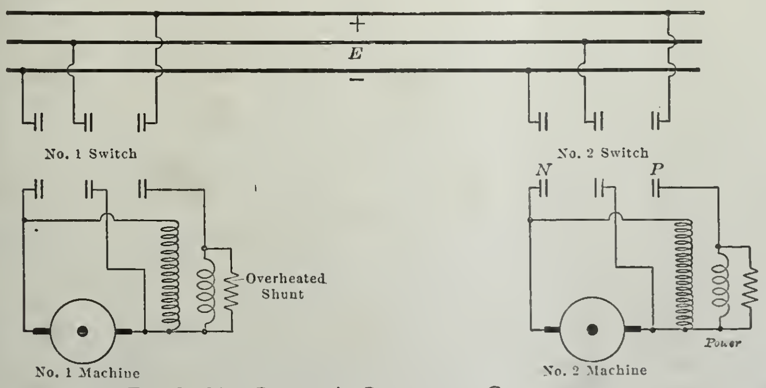

FIG. 2. MR. GORILLA'S GENERATOR CONNECTIONS

The trouble was poor contact between the blade and the jaw. It was found that the blade had some high spots ("blisters") which would not allow the faces of the blade to lie flat against the jaw and utilize its full contact surface. The current from No. 2 armature, instead of passing principally through the jaw *P*, was shunted through the equalizer to the series winding and shunt strip of the No. 1 machine. Of course, No. 1 machine would take most of the load when the heavy load was on, and the shunt would only smoke at those times.

LOUIS J. GORILLA.

Ironwood, Mich.

Circuit Breaker Connections for Generators in Parallel

Two 200-kilowatt 550-volt generators were first connected up for parallel operation as shown in Fig. 1, with the circuit-breakers of both machines connected in the leads from the series field windings. There was a mechanically operated field rheostat on No. 1 generator rigged so that when the circuit-breaker blew out, it would cut in the resistance to prevent the machine from flashing over, as described in the issue of June 7. After the machines were run awhile the circuit-breaker on No. 1 blew out and the automatic rheostat cut in the field resistance. Current still flowed from the No. 2 machine through the series field winding, however, and kept the voltage up and caused trouble at the brushes so that something else had to be done. The circuit-breaker on No. 1 was changed to the other lead, and there was more trouble. When there was a short-circuit, both circuit-breakers blew out, but the line was still getting current. On tracing it out, it was found that No. 2 machine was connected to the busbars by its own negative lead and by the equalizer and the series winding and positive lead of No. 1 machine, as shown in Fig. 2. Then the circuit-breaker on No. 2 machine was changed to the negative lead, and there has been no more trouble.

LEON L. POLLARD.

Fairfield, Me.

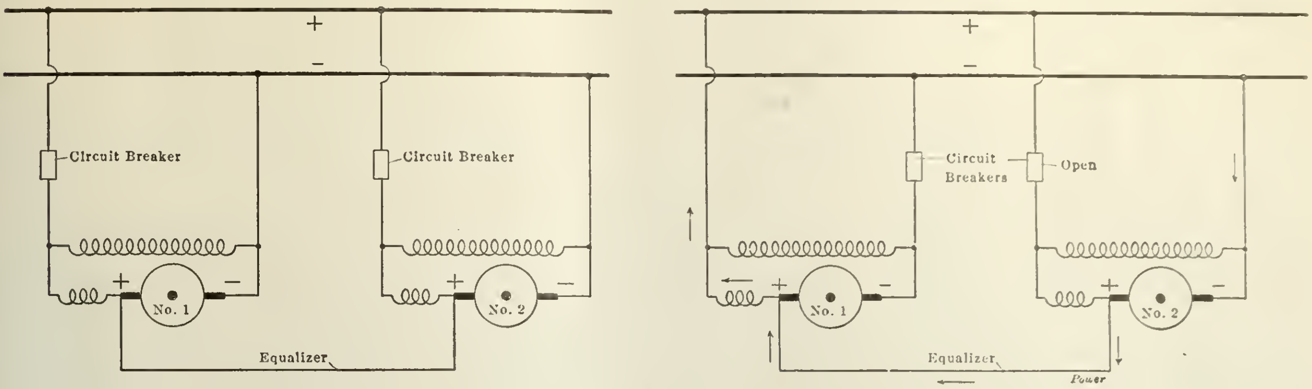

FIG. 1. MR. POLLARD'S GENERATOR CONNECTIONS FIG. 2.

[One of the fundamental and long-recognized rules for connecting two or more compound-wound generators to the same busbars is to put ammeters and single-pole circuit-breakers in the direct leads—that is, the leads which do not come from the series windings. An ammeter in the "compound" lead frequently does not indicate the true current of the machine, and a single-pole circuit-breaker in that lead is useless, as Mr. Pollard has found out.—EDITOR.]

A large Humphrey pump is being designed for construction in Germany, which will deliver over a ton of water per second, with a lift of 200 feet, giving 7000 horsepower in the water lifted. A very high fuel economy is expected, and the pump is to be used to demonstrate the utility and economy of the gas-hydroelectric system advocated by Mr. Humphrey for central electric stations. The Humphrey pump, it will be remembered, is an internal-combustion apparatus, the gas being exploded against the moving column of water itself, and the oscillations of that column being used to effect the intake, compression, etc.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Gas Engines for Driving Alternating Current Generators*

By H. G. Reist

The problem of driving an alternating-current generator by means of a gas engine is not inherently different from that of driving it by a steam engine. If the shaft of the engine turned with a uniform motion, no difficulty would be experienced and no special design would be necessary. It is the variation in angular velocity and speed that affects the driving of alternators.

If the current of a single generator is used for lights or for heating, as in electric furnaces or in electrolytic work, variations in velocity either during a single turn or due to the hunting of the governor will simply increase and decrease the load as the speed varies. If induction motors are driven from a single generator, it is only under peculiar circumstances that any trouble is experienced due to speed variations in the engine, because this type of motor does not have to follow exactly; it is as if it were belted to the engine, the connection being slightly flexible.

A synchronous motor or rotary converter, on the other hand, must keep in phase and behave as though geared to the engine, and must respond to all its speed variations. If it does not keep absolutely even; that is, if its phase relations change, cross currents will flow. When two or more generators are operated in parallel, their behavior is similar, any angular departure of one from the other causing a cross current. The volume of the cross current depends, with any given design of generator, on the angular departure of the generators from each other. This departure may be twice the angular variation of the engine-rotating parts from a mean position, because one may be a maximum distance ahead while the other is in the most backward position.

If the generators were mounted on the shaft so that the relations of the poles to the cranks were identical, and if it were possible so to synchronize them that the corresponding cranks of engines to be run together were exactly together, no cross currents would flow, because the engines would slow down and speed up together. This is not feasible, however, and it becomes necessary to design the engine to run with a fairly uniform rotation. It has been found good practice to limit the variation from a mean position of the revolving parts of the electric generator to 1¼ electrical degrees.

An electrical degree is 1/360 part of the distance from the center of one field magnet pole to the center of the next pole of the same polarity; therefore, on a two-pole generator an electrical degree is one mechanical degree of the polar circle of the machine; if the generator has four poles, an electrical degree will be one-half a mechanical degree of the circle to which the pole faces are bored; if there are six poles, it will be one-third of a mechanical degree. In general, to reduce electrical degrees to mechanical degrees, we must divide the allowable variation by one-half the total number of poles on the generator; so that for a twenty-pole machine the 1¼ electrical degrees mentioned above would be 0.125 actual degree on the circumference of the flywheel. From this it is evident that with a generator of many poles, a more even speed is needed than for one with few poles. For a 60-cycle generator, which at a given engine speed has 60/25 as many poles as a 25-cycle generator, the evenness of running must be 2.4 times that for a 25-cycle generator, at the same speed.

The cross currents between two electrical generators tend to speed up the lagging machine, bringing them more closely into synchronism. If there were no inertia the rotating parts of the generator and flywheel would quickly get into synchronism, almost eliminating the cross currents, but this is impossible. The cross currents are a factor of the amount of inertia with a given natural angular variation, and it will readily be seen that from this standpoint the larger the flywheel the less effect a given value of current or torque will have on the mass. If the flywheel is very large, the current which it may be practical to allow to flow between the machines may not be able to draw them together at all. Hence a large flywheel, while useful in obtaining uniform rotation so far as the engine is concerned, prevents the current flowing between the machines from being very effective in drawing them into synchronism. This shows that it is desirable to obtain uniform rotation in other ways than by the use of an excessively heavy flywheel. Currents flowing between machines occasion losses in the copper and this adds to the heating of the machines. They thus reduce the output with a given rise of temperature and reduce the efficiency.

Certain elements of design may be introduced into an electrical generator to make it less sensitive to slight variations in turning moment supplied by the engine, such as building a generator of poor regulation. The regulation must not be too poor, however; otherwise the operation of the system will be unsatisfactory. A "squirrel-cage" winding in the poles of the generator allows secondary currents to flow in this part of the structure and increase the torque, tending to draw the generators together with a given interchange of current between the two machines. This is of great assistance in parallel operation of generators and should generally be applied on generators to be driven by gas engines. If a flexible connection could be provided between engine and generator it would greatly assist in satisfactory parallel operation, but this connection is generally applicable only on small machines. The ultimate solution lies in the direction of greater uniformity of motion in the engine itself.

Uniformity of rotation of gas engines is dependent on a number of elements of design, such as the number of impulses per revolution, which in turn is dependent on the number of cylinders and arrangement of cranks and on whether a two- or a four-stroke engine is used; the compression and weight of the reciprocating parts; the time of ignition; the weight of the flywheel. The use of a heavy flywheel, however, while one of the simplest expedients, is the least desirable method of obtaining even rotation of the engine shaft, and other means should be used to obtain as uniform rotation as possible.

The following seem to be the desirable characteristics of gas engines for driving alternators:

(*a*) High speed. This will require fewer poles with a given frequency and a greater angular variation will be allowable.

(*b*) A light flywheel. This will allow the current to keep the generators together with a minimum disturbance.

(*c*) Large engines should be built with many cylinders and cranks so placed as to contribute to an even turning moment.

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*A paper read at the Atlantic City meeting of the American Society of Mechanical Engineers.

Jacket Water Temperatures

By John S. Leese

The writer recently made a series of tests on a single-cylinder Simms gasolene engine, in which the load was varied and the revolutions were kept constant. The effect of the temperature of the cooling water at outlet, although not the original motive for the tests, is interesting. The inlet cooling-water temperature was kept constant at 42 degrees Fahrenheit. The results are given in the table herewith.

RESULTS OF TESTS.

Test No.	Duration, Minutes	Revolutions per Minute.	Brake Horsepower.	Temperature of Cooling Water at Outlet in ° F.	Thermal Efficiency on B.H.P., Per Cent.
1	30	1000	1.875	81	15.12
2	25	1004	2.25	88	15.17
3	25	1002	3.45	94	16.12
4	30	1000	3.68	100	16.66
5	25	998	4.49	105	17.29

In each case, before readings were commenced for a test, the engine was run long enough to allow the conditions to become constant.

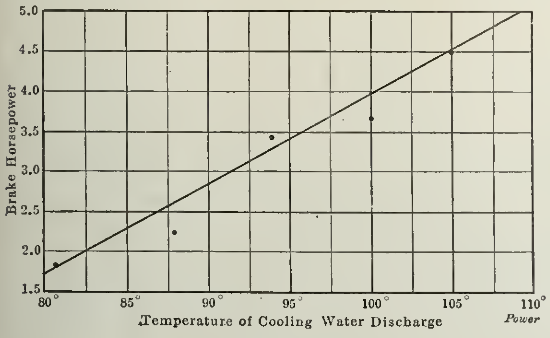


Fig. 1

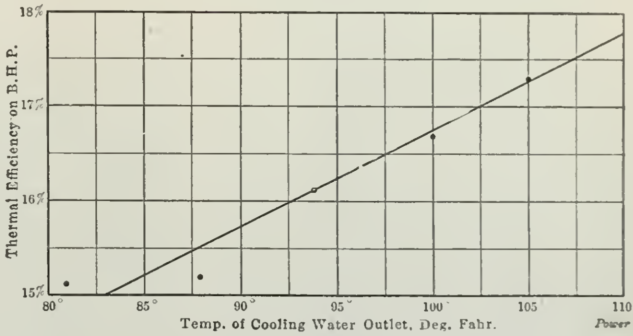


Fig. 2

Fig. 1 shows the relation between load and jacket-water outlet temperature, and Fig. 2 the thermal efficiency and outlet temperature relation. These results confirm the tests made two or three years back by S. F. Edge on a Napier gasolene engine, in which, however, the load and revolutions were kept constant and the cooling water alone was varied. Those tests showed that the most efficient temperature for the jacket water from the point of view of brake horsepower and fuel consumption to be 212 degrees Fahrenheit. I do not wish to be understood to mean that the water should be carried at this temperature in practice; my object is merely to point out that an appreciable loss is entailed by having the water running cold at the outlet, as one finds it in some plants. If the outlet water be allowed as hot as one can bear it on his hand, excellent results are obtained in practice.

Gas Engine Troubles

By James W. Little

Much has been said and written at different times about the erratic behavior of gas and gasolene engines. It has been my experience that a great deal of the trouble is due more to the lack of a thorough understanding of the principles of operation than to any mechanical imperfections in the engines. I dare say, without fear of contradiction from those who have made a study of, and have had experience with, the internal-combustion engine, that the majority of cases of trouble have been remembered with a sense of the ridiculous. Generally speaking, it is something so simple that it is overlooked in the strenuous search for the trouble. Usually the operator "goes after" the engine with a monkey wrench and pulls it apart only to find nothing apparently wrong.

After such a wholesale dismembering it is often the case that the engine, on being put together again, starts up promptly, to the mystification of the operator.

Then again, this operation may be gone over and still the engine refuses to start, and someone else has to be called in to diagnose the case.

It has been said, and with a great deal of truth, too, that nine-tenths of the failures to start, and similar troubles, are due to faulty or completely disabled ignition. In view of this probability, it is rather unwise to take an engine to pieces before investigating the ignition system thoroughly. The learning of a few simple principles would be a better use of time expended.

There are several things that may impair or totally disable the ignition system. Starting with the battery cells, if a battery is used, it is well to see that they are connected according to the directions usually sent with the engine. It is a good plan to scrape the connectors clean, and also the nuts of the binding posts where the connectors lead from cell to cell, then screw these nuts down finger tight and keep them so. Following the wires out to the spark coil and switch, the same suggestion as to cleanliness and tight connections applies. The wires leading to the binding posts on the igniter may be scraped clean, and with the switch closed the spark may be tested by touching the bright ends together and then quickly separating them, whereupon, if everything is as it should be, a good, bluish-white spark will result. If a lot of little sparks are the result, imperfect ignition may be expected, because this usually indicates a short-circuited spark coil.

The igniter insulation may be tested by connecting to the insulated electrode one of the wires and touching some bright part of the engine with the other, taking care that the electrodes of the igniter are well separated. If no spark results, the insulation is all right. It should be seen that the movable electrode is not corroded and gummed so it is with difficulty moved; it should move freely.

The valves of the engine should be tight when closed, as a leaky valve may so spoil the compression that the ignition is unreliable.

One of the reasons why an engine that has stopped, been taken to pieces and put together again will sometimes start, is that an exhausted ignition battery has had time to build up again. When this is the case, however, the battery soon becomes exhausted again and ignition ceases, stopping the engine. [Another reason for this experience, if the engine runs on gasolene, may be that too much gasolene was delivered to the cylinder during the efforts to start, and this has time to evaporate while the engine is being taken down.—Ed.]

Sometimes an engine positively refuses to start some morning, although it ran right up to shutting-down time the night before. Smutty igniter points are often responsible for this. Another common cause of this behavior is leaving the ignition circuit closed when shutting down. When this is done the engine is liable to stop with the timer in contact and run down the battery over night.

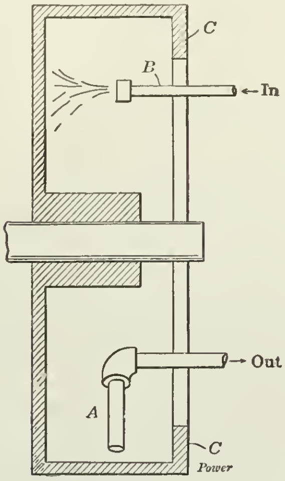
A more recent invasion of the leather-belting field is now reported in the application of paper belting made in England, the claim for it being that it is stronger than leather, will not stretch, and is less subject to climatic changes.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Pulley for Use with Prony Brake

The sketch shows a pulley for use with a prony brake. The outer circumference is turned flat and flanges are left projecting inward as at *C*. A ¼-inch pipe *B* with a cap having a very small hole drilled in its center, is connected to a source of water supply and the interior of the pulley is thus kept supplied with cool water. At *A* is a ½-inch pipe supported so that its end is about ⅜ inch away from the interior circumference of the pulley and pointing as shown in the sketch. This acts as a scoop and prevents the flanged part from becoming filled and overflowing. Such an arrangement prevents the pulley from overheating and keeps the water supply constant.



Cross-section of Pulley

A. H. Anderson.

Chicago, Ill.

Water Wrecks Engine

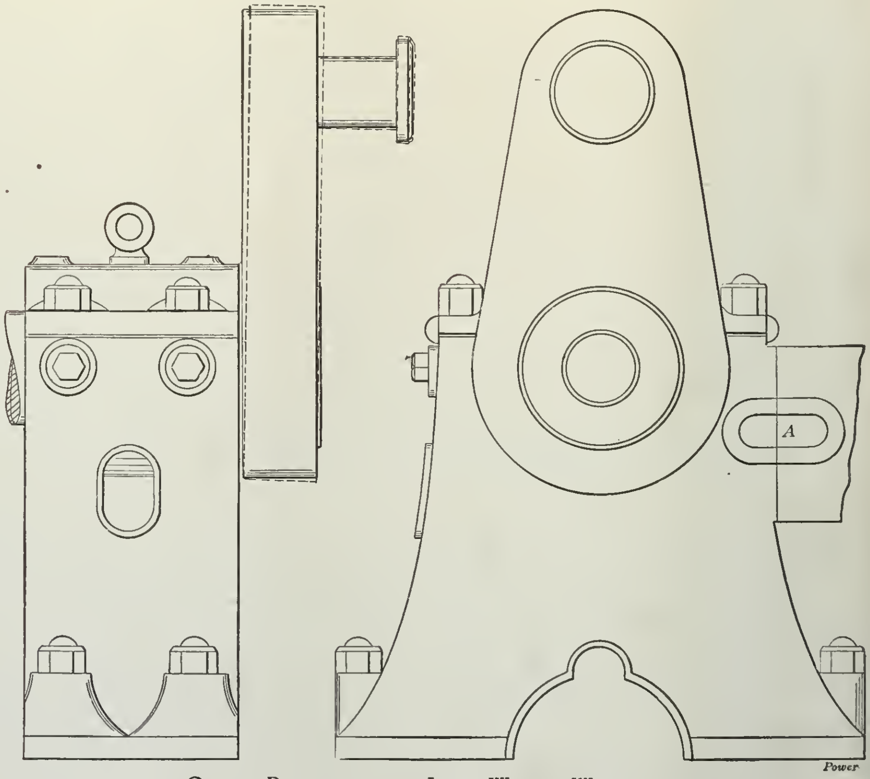
That an engine will act as a pump under certain conditions was demonstrated by an accident which came within the writer's observation a few years ago. The engine was a single 24x48-inch Corliss and could be run either condensing or noncondensing. It was belted to a short jackshaft, that was coupled with the main shafting by a jaw coupling.

On shutting down, a regular routine had been followed which had always been successful until a new man came on watch, and then the inevitable happened. It had been the usual custom to cut out the dynamos, open the jaw coupling on the jackshaft, close the engine throttle and after the engine had been stopped, to shut down the independent air pump and the condensing plant; the exhaust cut-out valve, between the engine and the condenser being left open, as this engine was usually the first one started again on Sunday night.

However, the new man became confused and shut down the air pump first; closed, as he supposed, the injection valve, and then opened the jackshaft clutch, closed the throttle and hooked up the governor. The condenser was on top of the water end of the independent air pump, and had never been fitted with an injection valve, but in lieu of this, the injection water had been regulated always by the valve in the condenser, whose only office was to adjust the amount of the opening to the load, and when this valve was apparently closed it was kept ¼ inch off its seat by four small lugs; these being intended to act as a guard against closing the valve completely, when adjusting it for light loads, or cold injection water.

After the throttle had been closed and the engine slowed down, it being kept moving by the momentum of the flywheel, the vapor was destroyed in the steam cylinder and the piston being absolutely tight, the engine acted as a pump. It was still turning very slowly, the wristplate not having been unhooked, when the water was lifted and the crash came, just after the automatic valve on the condenser had opened and destroyed the vacuum.

Referring to the sketch, the link *A*, as well as the bolts between the main bearing and the bedplate, let go. This allowed the pedestal to shove away from the bedplate and brought the flywheel up against the side of the pit. The crank was shoved out of line 3/16 inch, as shown by dotted lines, and the connecting rod was also badly sprung at the crank end.



Crank Dislodged and Link Which Was Broken

There now began some hasty makeshift repairs, until a new crank could be obtained. First, a new link was made to replace the one at *A* and the bedplate and pedestal were brought back to position; a new set of bolts were made and fitted and the connecting rod was heated and sprung back into alinement. While this was going on it was decided that the only thing that could be done with the crank was to file the pin parallel to the shaft. The shaft being tried for level, it was found to be in perfect alinement, so far as the level was concerned.

The crank was first placed upon the top quarter and a dead level filed across the pin; then a similar level was filed with the crank upon the bottom quarter, care being taken to remove as little metal as possible; next, the same amount of metal was removed from the pin with the crank upon both centers. These four spots were increased to eight, then sixteen, and finally thirty-two. Then the corners were carefully filed off to make the pin round.

An old set of crank-pin brasses was now poured with lead, fitted to the pin, and 3/16 inch planed off one side, and a "dutchman" fitted on the other side, to bring them in line with the connecting rod and the distorted position of the crank. When the pin had been filed as close as possible, the brasses and rod were connected up, and the engine started, and lard oil and flour of emery were fed to the pin, the engine being run light at about 50 revolutions per minute. After a run of two hours the engine was stopped and the brasses were opened up for an inspection of the pin. As the filing had been an excellent job, there were but few hard spots to take off, and while this was being done, the brasses were rebabbitted after melting out the lead, and fitted to the pin. While the crank-pin brasses were being fitted, the brass plate was removed from the crank center, and a hole chipped into the crank on the two opposite sides of the shaft, in order to allow accurate calipering of the shaft to make a gage which was sent to the shop to bore the new crank to.

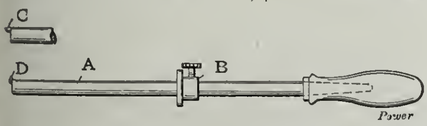
In the mean time the engine was started up and ran under full load for three weeks while awaiting the new crank, and gave no trouble whatever.

E. A. Blackwell.

Kingston, N. Y.

A Gage Glass Cutter

The following sketch shows a gage-glass cutter which has been found to perform its work satisfactorily and to cut as well as any wheel cutter. The stem *A* is made of steel about ¼ inch in diameter and 8 or 10 inches in length. The cutting point is formed by first filing down the end of the stem to form the projection as shown at *C*. This is then heated to a cherry red and bent up as at *D*, after which it is again heated and then hardened by quenching in cold water. This being done, it is dressed to a sharp edge with an oil stone. The slide *B* and the handle are then put on.



Gage Glass Cutter

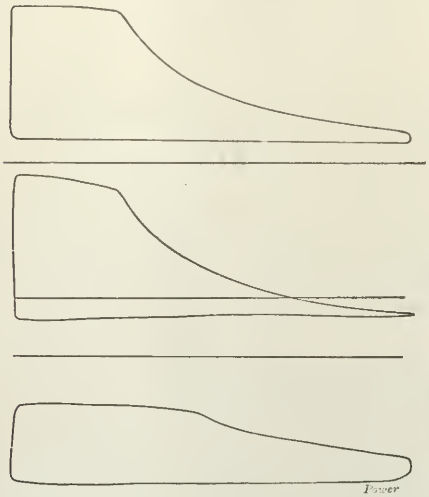
R. L. Mossman.

Tampa, Fla.

Pumping Engine Record

The figure shows diagrams taken from a 36, 60 and 90 by 66-inch triple-expansion pumping engine which were taken during the acceptance test. The steam pressure was 180 pounds, the vacuum, 27.7 inches and the total water pressure was 96 pounds. The average number of revolutions per hour was 1208.

The duty per 1000 pounds of dry steam was 184,476,200 foot-pounds. This is a world's record for a pumping engine.



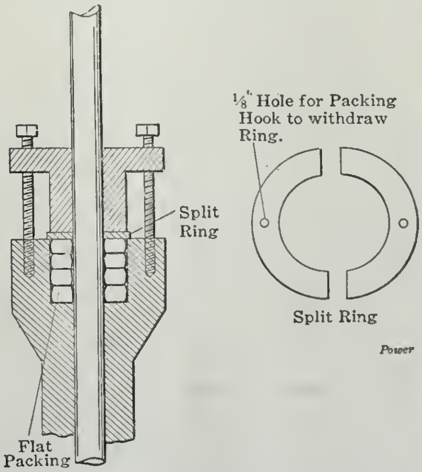
Diagrams from Pumping Engine Holding World's Record

H. R. Blessing.

Philadelphia, Penn.

Means of Keeping Packing in Place

We formerly experienced much trouble in keeping the packing around the plunger of our hydraulic accumulator, which is not of modern design and has the stuffing-box gland made with a decided taper. The depth of the stuffing box is only 2½ inches and a pressure of 2000 pounds per square inch is carried on the piston.



Device for Holding Packing

To remedy the trouble, I had a small steel ring (see sketch) made in halves and placed on top of the packing as a follower. The ring is made of a size to allow 1/16-inch clearance around the plunger to avoid any danger of cutting, and a ⅛-inch hole is drilled in each half to allow the ring to be withdrawn with a packing hook when necessary.

Before using the ring, it was necessary to repack the gland every three or four days, but now a packing will last for about three weeks.

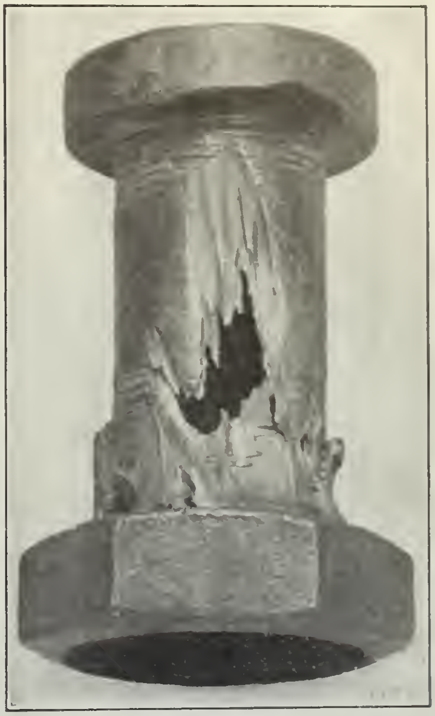
C. C. Harris.

Springfield, Mass.

A Case of Rapid Erosion

The writer's attention was recently called to the very rapid erosion which took place in a gasket union shown in the accompanying photograph.

This union was taken out three weeks after it had been installed. It connected a steam trap to a 6-inch water leg, 440 feet from the boiler, and carried steam which was delivered to the engine at 130 pounds pressure. The trap and joint were located in a brick and cement pit and were covered with water in which there was considerable sand and grit.



Union Showing Effects of Erosion

A probable explanation of this erosion is that the escaping steam stirred up sand and grit, which was sucked into the current of escaping steam and the combination worked like a sand blast. The writer would like to know if any of the readers of Power ever had a similar experience, or if anyone can offer an explanation other than the above.

E. M. Hall.

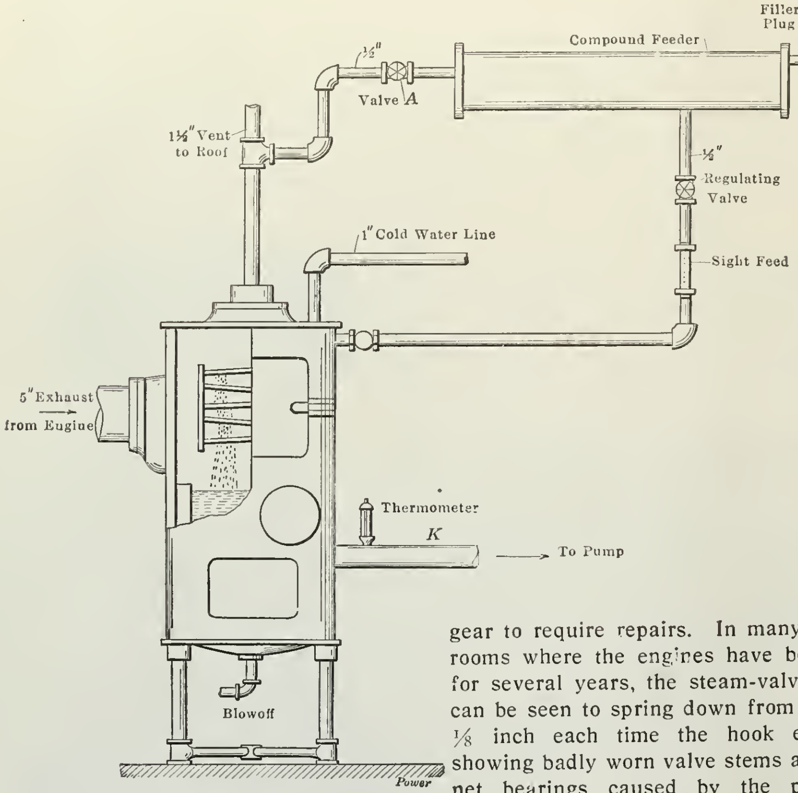
Lexington, Mass.

Boiler Compound Feeder

The accompanying sketch shows a type of boiler-compound feeder which I am thinking of installing, but should first like a few opinions regarding it from Power readers.

The feeder is made of a piece of 4-inch pipe 2½ feet long, capped at both ends and tapped for ½-inch pipe. The filler plug consists of a 1¼x½-inch reducer. At *A* the feeder is connected to a 1½-inch vent pipe leading to the roof.

I figure that with the valve *A* open, a little pressure will be created which with the weight of compound in the feeder will more than balance any tendency to back up through the regulating valve. Also, steam entering from *A* will circulate the compound and act as a vent. Backing up should be considered, for when the engine carries a heavy load there is enough back pressure in the heater to cause the cold-water regulating-valve float to kick up quite a rumpus. The sight feed, which is made from fittings from an old lubricator, is adjustable and independent of valve *G*, and will take the place of a gage glass on the compound feeder, thus saving the unnecessary opening of the feeder or heater to make sure it is feeding all right.



Proposed Location of Compound Feeder

The compound and cold-water feeding together and attaining the temperature of 210 degrees Fahrenheit, ought to cause some scale precipitation in the heater which is the point aimed at. The feeder could be put on the pump-suction line as at *K* with much less trouble, but whatever precipitation that takes place in the heater would then take place in the feed line. This is to be avoided as far as possible as it is much easier to clean the heater than the feed line.

Charles T. Reed.

Elizabeth, N. J.

Operation of Dashpots

One has but to observe the valve gear of an engine and to converse with experienced engineers, to become convinced that the steam-valve stems and their bearings are the first parts of the valve gear to require repairs. In many engine rooms where the engines have been run for several years, the steam-valve stems can be seen to spring down from 1/32 to ⅛ inch each time the hook engages, showing badly worn valve stems and bonnet bearings caused by the powerful downward pull of the dashpot.

Some builders have recognized this fact and have placed the drop levers inside of the bonnet thus lessening the strain, but this plan has its drawbacks and if the dashpot were always operated carefully and properly the change would not be necessary. In almost every instance of badly worn steam parts the dashpots will be noted to be acting with too much force.

Repeated experiments, using an indicator, with the dashpot seating rapidly or slowly will prove to the satisfaction of any engineer that a rounding cutoff will remain rounding and a square corner will still be square in either case. This being true there is clearly an advantage in favor of the slow movement.

The weight due to the atmospheric pressure on a 4-inch plunger would approximate 188 pounds provided a perfect vacuum were attainable in the vacuum chambers. Making due allowance for some air entering, there must still be quite a weight, the lifting of which from 80 to 100 times a minute represents considerable power expended. But that is not the point. The more important fact is the strain upon the pins, the downward wear of the valve stem and bearing, the rapid wear of the hook-plate edges, and the friction and wear on the knockoff cams and friction plates. Also, there is added friction and strain on the wristplate and governor connections and a tendency to noise, shock and rattling throughout the entire valve gear.

The writer's method of operating dashpots is not new but it has been suggested to many appreciative engineers who had the impression that because certain attachments for the control of dashpots were furnished with the engine, they should never be changed or disturbed. The correct way is to get the best possible fit to the cup leather or packing ring and then provide means for killing the surplus vacuum by admitting air into the vacuum chambers, using the least possible vacuum necessary to do the work. At all times the vacuum should be strong enough so that the plunger will strike bottom when the exhaust valve is opened on the air-cushion chamber. This can be accomplished by the use of an in-and-out-check valve attached to the dashpot. By this means the pot is always under control of the operator who by the simple operation of the air cock and cushion valve can vary the seating speed at will.

F. C. Holly.

Yazoo City, Miss.

Condensed versus Fresh Water for Making Steam

I have been a subscriber to engineering magazines for almost forty years and have never seen an explanation given of why fresh water will make more steam than condensation, or returns from a heating system.

A great many of the old-time engineers know it to be a fact but have never given it the necessary consideration.

We all know that fresh water does not make clear ice; it is full of air globules and must be boiled to extract the air before it will make clear, solid ice. For the same reason a hot-water pipe in a bath room or kitchen will burst with a slight frost much sooner than the cold- or fresh-water pipe. This is because whatever elasticity there may have been in the water has been removed by boiling the air out of it. Nearly the same results are obtained by heating the feed water in an open heater. If fresh, cold water is pumped into a closed heater, and heated to 212 degrees, all the air globules expand to a great many times their bulk and become part of the steam immediately upon entering the boiler. This air must be heated to the temperature of the steam at the expense of fuel. I have no means of making a test but I fully believe that it is more economical to run the condensation into the sewer and use fresh water through a closed heater if there is enough waste heat to bring the temperature to 212 degrees without using live steam. Besides, the exhaust steam is impregnated with cylinder oil and it is almost impossible to keep it out of the boiler.

CHARLES STEVENSON.

St. Louis, Mo.

How Should Boilers Be Set?

I would like to hear what the readers of POWER have to say regarding the proper hight a 72-inch by 18-foot boiler with seventy 4-inch tubes should be set above the grates. It is our intention to set the boiler 42 inches above the grates. The bridgewall is to be 30 inches high and hollow, communicating with the outside through the register door and with the furnace—8 inches below the top of wall—by every other brick being left out. The boiler-feed pump exhausts into the ashpit.

Would anyone recommend "An Economical Boiler Furnace" illustrated and described in POWER, June 23, 1908?

HENRY SHOUT.

Paris, Ky.

An Example of Self-Education

I have some bright young men working for me, and the other day I walked into one of our engine rooms and asked one of our assistants if he could figure a problem, and gave him the following:

Two mills each burn 100 tons of coal per day; in one the stack temperature of the waste gases is 450 degrees, in the other 200 degrees. Find the saving in tons of coal per day in the second plant over that of the first.

He submitted the following solutions which pleased me very much:

Eighteen pounds of air are required per pound of coal for best combustion.

$$200{,}000 \times 18 = 3{,}600{,}000$$

pounds of air required for 100 tons of coal.

Heat required to raise one pound of air one degree Fahrenheit is 0.23751 B.t.u.

Therefore

$$3{,}600{,}000 \times 0.23751 \times 250 = 213{,}759{,}000$$

B.t.u. going up the stack in plant No. 1 in excess of that in No. 2.

The B.t.u. in each pound of coal = 14,500.

Hence,

$$213{,}759{,}000 \div 14{,}500 = 14{,}742$$

pounds of coal required to heat this air from 200 to 450 degrees.

And,

$$14{,}742 \div 2000 = 7.371$$

tons of coal per day wasted.

This young man after one year in the high school, on account of the death of his father, came to work for me. He has always been a student, and there is hardly a day that I do not catch him with a book and pencil in his hand figuring out some problem. For the last two years he has taught in one of our city evening schools, instructing firemen and engineers, as well as attending to his duties at the mill during the day. This goes to show what a person can accomplish by trying, and I predict a great future for this young man.

GEORGE H. DIMAN.

Lawrence, Mass.

A Substitute for Oil Packing

Not having any specially prepared oil packing at hand, I was compelled to pack the piston rod of an oil pump with ordinary steam piston-rod packing. While this kept most of the oil in the cylinder, a considerable quantity leaked out along the rod and soaked through the packing. After emptying the drip pans beneath the pump several times, and doing much extra wiping and cleaning, on account of the superfluous oil in the vicinity of the pump, a barrel of soft soap caught my eye, and I decided to use some of its contents in an attempt to overcome the difficulty.

I removed the packing, put a little soap in the bottom of the gland and rubbed a liberal quantity of soap into each ring of packing. I then replaced the packing and the pump has since been running steadily, without the loss of a particle of oil through the gland.

ROBERT ROE.

Hamilton, Ohio.

Explosion of Pipe Line

The line was of 1-inch extra-heavy pipe connecting to two three-stage air compressors supplying two Diesel engines of 450 horsepower each. Being a temporary installation, it was made up of numerous short lengths connected by couplings, unions and tees. These fittings were of the tight-joint type and of steel and were supposed to withstand 2000 pounds per square inch.

The writer had thrown the load off one unit at the switchboard and had signaled for the assistant to shut it down when, with a deafening report, six of the fittings burst and flew in all directions. Four tees and two ells let go, while two tees connecting the branch lines between the others held all right. There were two different weights of metal used in the fittings and five of the lighter grade burst while only one of the heavier grade let go.

The cause may have been due to several things: The line was poorly supported and in carrying air at 1100 pounds pressure it became very hot. The fittings were uneven in thickness and had probably crystallized under the continued heating and cooling. It is possible that there was enough oil in the line to ignite and cause an excessive pressure, although the gage showed 1150 pounds at the time of the explosion. There are relief valves, set at 1200 pounds, on each compressor, but there were none on the line so the supposition is that the flow of air having nearly stopped, the temperature rose rapidly and igniting the oil was sufficient to burst the fittings.

G. H. KIMBALL.

Southbridge, Mass.

Pipe Covering Materials

In discussing pipe-covering materials mention is seldom made of mineral wool. It is said to be short lived. A couple of years ago we covered a number of pipes with mineral wool, and were told to wait three months and see where our mineral wool would be. We have waited but it is still there. Furthermore, in a plant not far from here mineral wool which was put on ten years ago is said to be as good as ever.

In tests conducted by Professor Ordway, a covering made of carbonate of magnesia and a small percentage of asbestos transmitted 2.5 B.t.u. per square foot per minute. In tests conducted by George M. Brill, mineral wool was found to transmit 1.29 B.t.u. per square foot per minute, or about half as much as magnesia.

In the March 29 issue of POWER, J. H. Boughton cites an instance of the saving effected by the use of what was probably 85 per cent. magnesia. He states that the condensation was reduced from 74.4 to 11.58 pounds per hour; that is, 67.87 pounds of steam were saved. If mineral wool had been used the loss would have been only half as much, and the steam saved would have equalled 68.61 pounds. This ought to be worth saving.

In the case referred to by H. N. Dawes in the June 7 issue, magnesia was selected in preference to sponge felt, although sponge felt was only 3 to 4 per cent. less efficient than magnesia.

Professor Kent gives a table of coefficients of radiation in which at 115 pounds absolute, that of bare pipe is 684; 2-inch cement composition (probably 85 per cent. magnesia) is 209; asbestos flock 210, and mineral wool 93. Asbestos is a poor heat insulator and must affect any substance with which it may be combined.

R. MCLAREN.

Berlin, Ont.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Bad Management

Mr. Sprague's article on "City Council Engineering" in the June 14 issue, is another case of history repeating itself, as per the editorial in issue of May 31.

But why it should be this way is a question not easily understood. Nevertheless it is a fact as hundreds of cases in all parts of the country bear witness. The same thing is true of the purchasing of supplies for the steam plant. The managers in the delusion that they are economizing will purchase what they think is just the proper thing and Mr. Engineer must use it, although in thousands of cases it is not the right thing at all. The result is trouble and needless expense. In some cases, this state of affairs is brought about by good salesmanship and ignorance; in others, it is due to prejudice in favor of a certain machine to such an extent as to be blind to the possibilities of others; all of this amounts to bad management.

A case similar to the one spoken of in the above mentioned editorial, with the exception that vertical boilers were installed in place of horizontal ones, is fresh in my memory.

The old plant consisted of fire-tube boilers in a splendid boiler room. A saving could have been made by utilizing a portion of the old foundations. Those parts which were not used would have interfered in no way; consequently, their removal would have been unnecessary. In spite of all that the engineering force could do or say the management could see nothing but a vertical type of a certain make. This was due, possibly, to the fact that they had once used fire-tube boilers, built by the same firm, with satisfactory results, and took it for granted that the water-tube type was first class. Another item which caught their eye was the attractive first cost. They overlooked the fact that a neat sum could have been saved in freight as they were close to the factory of a company building a first-class horizontal boiler, just the one for the place.

Then there were the old foundations, eight in number, to remove. This was no small task, as they were of concrete and of ample proportions, compelling the use of dynamite and a large force of men. The new foundations were expensive as the boilers extended 4 or 5 feet below the floor line.

The piping was far more expensive and less stable than it would have been with the other style of boiler. The boilers occupied all of the overhead space, it being just possible to house them without interfering with the roof. The management got the idea that the renewal of tubes would be a remote possibility. But the subsequent renewal of tubes and the attendant perforation of the once splendid roof quickly banished this dream.

The new boilers were rated at 250 horsepower each, and guaranteed to carry a 25 to 50 per cent. overload, whereas, in thorough tests, using a good automatic stoker, the best they ever developed was 226 horsepower.

The changes made a bad matter worse, for while there was insufficient boiler capacity before, there was less after, to say nothing of the useless expense involved. The management was then prevailed upon to purchase another boiler, but still they seemed to know more than the engineers and concluded to install a surface condenser, a good idea as far as it went. There being a large tank pump in a pit in the engine room they conceived the idea of placing the condenser over the pump, thereby securing the condensing water as it left the pump on its way to the tank overhead, at the same time saving the cost of digging a pit. The bottom of the condenser was brought considerably above the lowest part of the engine cylinders. This arrangement exists still, as far as I am aware.

JOSEPH STEWART.

Hamilton, O.

Installing Globe Valves

I was interested in R. A. Cultra's excellent article in the June 21 issue on "Installing Globe Valves" and I quite agree with his methods of installation with the exception of that for blowoff valves. I wonder if he ever stopped to think of the consequence of the thread stripping off the stem when the pressure is below the disk. If he has two valves on the line, he, of course, would be able to close the other and save his water, but I have seen lots of plants with only one valve to each boiler and none on the line.

For a blowoff valve I prefer an iron-body valve with a heavy cast-iron disk with a babbitt lining which can be renewed at a cost of only a few cents and a few minutes' work when the valve leaks. The pressure should be on the top of the disk.

We have had fifteen 2½-inch valves of this type in service for seven or eight years and never have been bothered by them clogging with scale. One thing about them that appeals to me is that the fireman cannot open or close them quickly enough to damage anything even if he wants to. I have seen so much abuse of blowoff cocks by firemen that I would not accept them as a present if I could possibly get the regular blowoff valve.

C. B. SMITH.

So. Framingham, Mass.

Engineers' Wages

No employer deliberately throws away money which he might put into his pocket. It is evident that many employers do throw away money by employing low-price engineers. These two statements appear to be contradictory, but they are not when we consider that a large number of engineers are employed by men who know nothing of their work, whose only method of judging their engineer's efficiency is by seeing the line shaft turn, and also that, outside of public-service power plants, the total cost of power is likely to be so small a part of the whole expense that a 10 per cent. saving there is lost sight of.

With long-distance power transmission coming into vogue to such an extent, it is easy for an employer to figure exactly what he can afford to pay for his power; he knows what his coal, oil, fuel and outside repairs cost and he can readily tell from his books just exactly what his engineer is worth. If the engineer wants to know himself, he can figure it almost as closely if he has knowledge of the price paid for coal and the price of electric power.

Suppose an engineer in a small plant figures that he is worth $25 per week when he draws $18, what is he to do? If he goes to the "Old Man" with his figures, it is likely that he will be told that all that he has said is perfectly true but that there are a dozen men ready to take his job if he does not want it.

In such a case the employer cannot be blamed if he hires another man at the same price, particularly if the other man proves to be equally as good. What I would like to suggest is a rather novel variety of union and one that will meet the approval of manufacturers and employers.

First, I believe in a union for education. Of such there is a plenty. Next, I believe in a union for publicity. Of this kind there is none.

If I should try to sell engines by waiting until I had one made and then going up and down the street in my old clothes, calling at one office after another until I found a customer and selling it at his price, I would land in the poor house. I believe that a man who has nothing to sell but his labor and brains should use the utmost diligence in advertising himself. This does not mean bragging; it means that he should keep a daily record of every plant that he handles and keep it in such a way that the cost per horsepower or kilowatt can be told at once for any given time, including all cost for repairs, etc. I believe that a graphic record accompanied by actual figures for proof, turned into the hands of the proprietor, would help matters by giving him a better idea of the actual cost, by calling to his attention the fact that he has a power plant and by showing him whether his engineer is growing more expensive or less so as time goes on. I also believe that another chart and set of figures should be sent to the headquarters of the union so that those who are interested may verify the computations, compare the results with the local market value of a horsepower and advise the engineer either that he has a "cinch" and should "freeze" to the job or that he is working too cheaply and should get more. Other members of the body should also be warned that such and such a job pays so many dollars a week and is worth so much more to any man equally as capable as the incumbent. By a process of education and comparison engineers would ultimately find their level and the plants that can be made to be worth $30-men would get $30-men and the plants that are worth $15-men would get that kind.

When an employer finds himself confronted by an engineer who can prove in cold figures that he is worth $25 and cannot get anyone else whom he dares to trust, for less than that, because others will know that that is what the position is worth, then he will pay $25. On the other hand, if an engineer is getting $20 a week and earning it, he need not be surprised or offended if someone else comes along and says, "Here, I can make that job worth $25 because I can save $5 worth of coal a week," and gets his job away from him by raising the price. Employers are "from Missouri" and must be shown; dollars and cents appeal to them. For one man here and one there to begin making spasmodic reports on the cost of power to their employers would do no good. It requires organization, uniform report cards furnished by the union, etc. It requires, also, full information about every power plant whether the engineer belongs to this organization or not. All this information should be available to every man who operates engines, whether he belongs to the organization or not.

E. F. HENRY.

Worcester, Mass.

I herewith give the scale of wages and hours of service which prevail here:

Engineers, first class, $4 per 8-hour day; engineers, second class, $3.50 per 8-hour day; firemen, $3.15 per 8-hour day; water tenders, $3.35 per 8-hour day; pump men, $3.75 per 8-hour day; coal wheelers, $2.75 per 8-hour day.

Overtime work is paid for as straight time, as is also Sunday work.

I would like to see tabulated data from all of the States. They would be interesting at least.

L. EARLE BROWN.

Roundup, Mont.

In answer to Waldo Weaver, in the May 17 issue of POWER, I will say that I am not in a very large or prepossessing plant. I have charge of two 16-foot by 66-inch return-tubular boilers suitable for 150 pounds working pressure, and one 14x36-inch Corliss engine and all of the necessary appliances of an exhaust-pressure and live-steam gravity-heating plant for drying draining tile and brick.

I do my own firing and repair work and think that I am a practical and capable mechanic. I did the installation and steam-fitting work with my own hands and all of the other work that an engineer might be able to do in a plant of this kind. Everyone knows that brick-yard surroundings are not the most congenial and it is hard to keep out the dust, yet I am not ashamed to have anyone visit my little plant.

The ordinary load is 90 horsepower. It varies, however, from just the friction load to 90 horsepower and sometimes to as high as 150 horsepower for a short time. The night fireman gets $2 per night with comparatively light work and small responsibility and no repair work. My wages are $75 per month with two Sundays off per month, if I can get the work out of the way without working every Sunday. The boilers are cleaned while the plant is in operation during the week, every other week, under my direction and with my help.

My hours of work are nearly 12 hours per day.

LEROY H. WHEAT.

Emmetsburg, Iowa.

Drop in Steam Pressure

In reply to H. E. Satterfield, who under the above caption, in the June 21 number, asks for information in regard to what is a reasonable amount of drop in pressure between the steam pipe and cylinder, there is no doubt in my mind that people seldom appreciate the amount of money lost through having steam and water piping too small. I remember once indicating in Chicago a 1200-horsepower vertical engine. By putting an indicator on the high-pressure steam chest and on the steam header, only 10 or 15 feet away, I found a drop averaging about 10 or 15 pounds, I forget which; the steam pipe was of a size considered ample.

The superintendent of a large mill recently told me that when indicating the low-pressure cylinder of a steam engine, he always used a separate indicator close to the opening on each end. He had previously found that otherwise the instruments read too low, because the low-pressure steam did not travel fast enough through the sizes of pipes ordinarily used. Personally, I never encountered this difficulty, because I have always used two indicators, but I can very well believe that his finding is quite correct.

The cost of labor, brackets, etc., is usually a considerable portion of the total cost of piping installed, and it does not pay to use the smallest practicable size. An even more important matter is that of sharp bends.

WALTER T. RAY.

Spartanburg, S. C.

Cylinder Lubrication

In the June 7 number, Mr. Longstreet tells of feeding a high-grade cylinder oil, costing 35 cents a gallon, into the high-pressure cylinder with a steam pressure of 160 pounds and 85 degrees of superheat. He feeds the same oil into the low-pressure cylinder; the only difference is that but half the quantity is used. The pressure in the low-pressure cylinder is 15 pounds and there is no superheat. Now, either this oil is too good for the low-pressure cylinder with its low temperature and wet walls or too poor for the high-pressure cylinder with its high temperature and dry walls.

In our plant we use an oil costing 90 cents per gallon in the high-pressure cylinder and one costing 30 cents in the low-pressure cylinder, fed at the rate of one-half gallon of each kind per day of 24 hours in an engine 22 and 40 by 48 inches with a steam pressure of 125 pounds and a receiver pressure of from 5 to 10 pounds, according to the load.

A company not far from where I work got a new cross-compound 20 and 36 by 46-inch engine, running at 126 revolutions per minute with a steam pressure of 150 pounds. One day the chief asked me to come over to see the porous cylinder on his engine. When I got there they had the high-pressure cylinder head off. The walls were as dry as though they had never seen oil. But like some of those marine-engine cylinders we read so much about, it was getting along all right without it. I was surprised when told that they were feeding a gallon of oil to it every 12 hours and the same amount of the same oil to the low-pressure cylinder.

We took off the low-pressure cylinder head and found that the low-pressure cylinder was getting enough oil; in fact, I thought too much.

We put the heads back, started up and fed a good stream of oil to the high-pressure cylinder. When we removed the head again the cylinder was as dry as before. The chief said, "I told you so, she is porous." But as there was no oil on the outside, I could not quite see it. I sent over to our plant and got a gallon of our high-pressure cylinder oil and we started the engine up again and fed our oil at a rate of about two drops per minute. After about an hour's run we stopped the engine and took the head off again. The cylinder walls felt quite oily to the touch.

I got an order for some high-grade cylinder oil at 90 cents per gallon and a box of cigars for not mentioning the "porous" cylinder.

A. W. Griswold.

Adams, Mass.

Ignorance among Engineers

I have read with much interest the article by our "Canadian Cousin" in the issue of June 21 on ignorance among engineers. Ignorance is more serious with the employers than the engineers. The employer often does not know the conditions, for if he did, he would put an experienced engineer in charge and not have a blockhead at the throttle merely to keep down the wage rate. The American engineer is not seeking to put his brothers out of employment but to pass laws which will force employers to hire only such men as are qualified to fill the positions and are capable of operating steam plants in a manner that will insure safety to life and property.

Mr. Leicester states that the installing of water columns without valves in the steam and water connections would not be allowed in British Columbia. This may be true, but he overlooks the fact that under some conditions it is absolutely necessary to have stop valves on the water-column connections, especially in marine service where large Scotch-marine boilers with several fire boxes are used. These boilers are generally provided with several water columns. To shut down one of these boilers in mid-ocean during a storm is liable to result in disaster. It is therefore necessary to have stop valves on both connections to close in case of accident to the gage cocks.

With stationary plants of large size there are usually many small boilers which are shut down often for cleaning and repairs. The mud in the water has choked up the angle valve in the water connection to the column in so many cases that the Massachusetts boiler board has made a law that where valves are used on the steam and water connections to the water columns, straight-way or gate valves must be used. These, too, generally corrode and leak when it is necessary to close them in cases of emergency.

Mr. Leicester states that there cannot be established any certainty of a clear passage for both water and steam by merely opening the drain from the water glass. This is true if there is not a ¾-inch drain to the water column as well as a drain from the glass.

The blowing down of the gage glass and water column depends on the kind of water used. In several plants I have been obliged to blow out the glass every hour, and the column every two hours. I have had a drain pipe to the column plug up in two hours' time so that 125 pounds pressure would not start it until a wire had been punched through, the valve being only slightly open and of the gate type.

I do not see how our "Canadian Cousin" can feel safe in blowing down the water column only once or twice a week. We have a very good vegetable water here, taken from the city supply which comes from a large pond; but under no conditions do I allow the boilers to run without testing the gage glass and water column every four hours. I use the surface blowoff and the bottom blowoff pipes each twice a day. This may seem to some engineers a useless waste of water and fuel, but we find about 1/16 inch of scale every six weeks and this amount of care is required to keep it from accumulating.

R. A. Cultra.

Boston, Mass.

Location of Throttle

Considerable has been said of late about the proper location of the throttle.

It goes without saying that the natural and most convenient position for the wheel is on the front side, handy to the valve gear. The only reason advanced for placing it elsewhere is that in case of a runaway this would be about the least desirable part of the engine room to visit.

There are several ways of overcoming this undesirable feature. The simplest and least expensive would seem to be to place the throttle wheel on the opposite side of the cylinder and use a bypass valve for handling the engine, keeping this closed when the main throttle is open.

In the case of an ordinary Corliss or similar type of engine, unless the valve gear has been deranged, it would be simply a waste of valuable seconds to attempt to close the throttle as a single sweep of the hand would bring the governor to its lowest position and shut off steam in a small fraction of the time required to wind in the "mile of thread," as one writer put it.

After all, there is only one really satisfactory solution of the problem; that is, install a reliable engine stop with push buttons located in various convenient parts of the room, including, in some cases, the boiler room and basement. When the stop is once installed it should not be kept for an ornament but should be used at regular periods to make sure that it is in working order so that it may be depended upon in case of emergency. With this arrangement not only the dangerous position is avoided but much time may be saved.

Leon Roundy.

Concord Junction, Mass.

Boiler Disaster of the Early Morning

I read with interest the editorial in the May 10 issue of Power on the early morning boiler explosion.

As it seems truly to be a case of unbalanced pressure at the initial opening of the stop valve, the thought suggested itself to me, should not all systems of boiler leads, headers and connections be provided with bypasses?

The inexperienced, and usually it is left to such men to cut in the boilers, seem not to realize the amount of steam that can get through a large stop valve just started from its seat.

For this reason, I would suggest the bypass system. We bypass nearly all other large valves, but, to my knowledge, a bypass has not been used for this particular service.

Since thinking of it, I have looked up the idea in all of the textbooks that were available, but could not find a thing mentioned in regard to the subject.

In view of what was said in the editorial about the heat and pressure stored on one side of the stop valve, and the large condensing area on the other, it would seem that the use of some method of warming up would be commendable practice.

James W. Little.

Fruitland, Wash.

Standard Colors for Station Piping

The article on "Standard Colors for Station Piping," in a recent number of Power interested me very much.

The increased size and complication of steam plants makes such a system necessary.

The tagging of valves and pipe lines does not fill the requirements, as the tags come off and are lost and they cannot be seen distinctly from a distance or through smoke or steam, and in an emergency nobody will know where anything is because everybody has been depending on tags.

Painting the pipes different colors is not very good as colors do not look the same to all men, or in different lights. A green in daylight becomes a blue in the light of the arc lamps, and a blue changes to black under the same conditions.

When pipes have different colors the men are liable to get into the habit of calling the pipe lines by the colors, without connecting the colors with what is inside of the pipes.

Also, there are so few colors that will wear well in the heat and dirt of a power plant, most of them soon become useless and the expense of repainting the piping is great.

I think that a system of markings should be arranged so that anyone could tell at a glance what is inside of a pipe and the direction of the flow.

All of the important valves should be numbered, not by tags but by figures painted on the covering of the valve body as large as possible and on the side which can be seen the best from doorways or gangways.

The numbers should preferably be painted in black as black wears and shows up well.

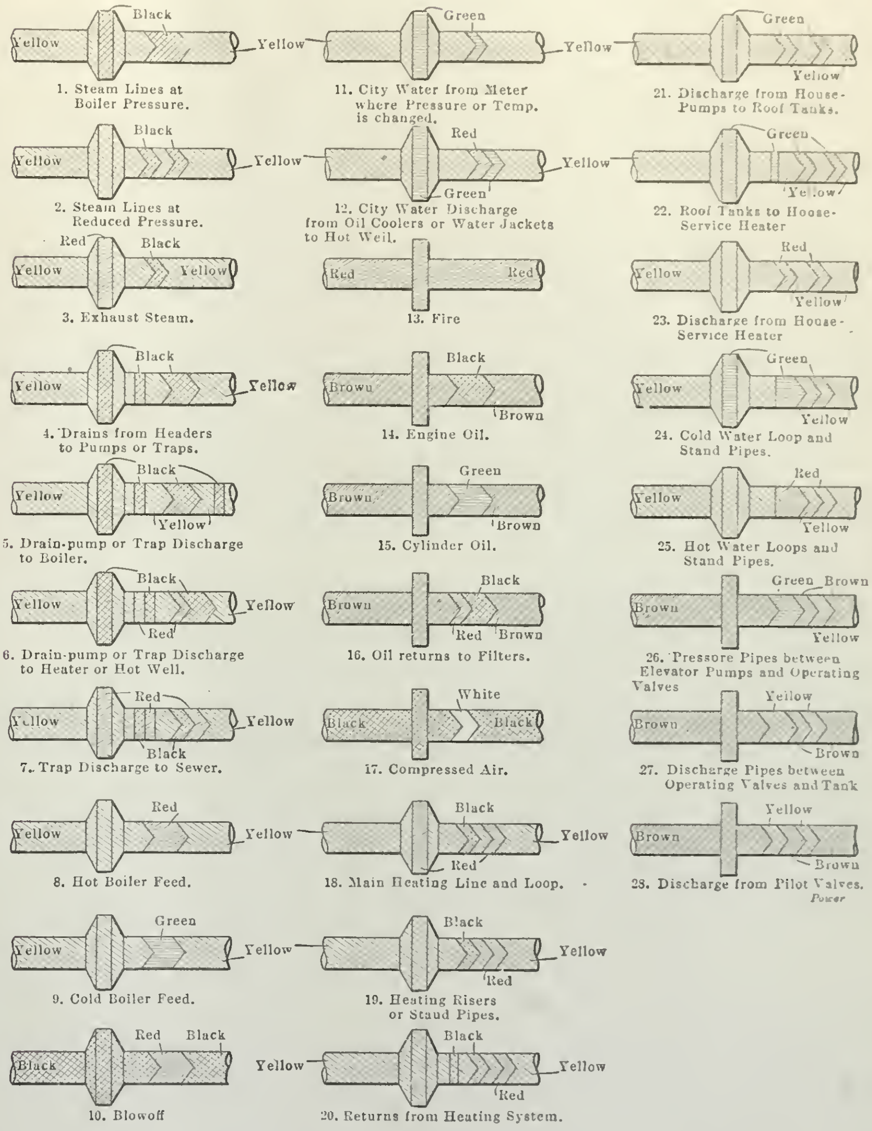
An arrow should be placed to one side of the number to show the direction of flow where the other pipe mark does not show. The boiler stop valves need not be numbered, let them be known as boiler stop valves. The valves on the boiler leads to the main header may be numbered from No. 1 up with numbers corresponding to the boiler numbers. All cut-out valves on the main header or on connecting bends between the headers may be numbered from 101 up with numbers running in the same direction as the boiler numbers. The valves leading to the loop, or to the engine leads where there is no loop, may be numbered from 201 up, and so on, as far as it is desired to carry out the system.

The men would soon learn the numbers of the valves and, in an emergency, it would be better to tell a man to close valve No. 304, than to try to tell him to close the stop valve on the loop for No. 4 engine. The probability is that he would be more likely to close the right valve and not shut off the valve for No. 5 engine by mistake.

Where a valve is not in plain sight from the regular passages, there should be in addition to the numbers on the valve, a number on the wall with an arrow pointing in the direction of the valve.

As a suggestion for a pipe-marking system, I offer herewith a chart which was worked out for an office building.

The colors will all wear well and will not change so as to be mistaken. The colors used are red, black, green, brown, white and yellow. The idea is to paint the markers on the pipe as large as possible and where they will show up the best. The point of the mark shows the direction of flow. Where flange marks are needed, and there is no flange where the mark is wanted a band can be put on the pipe of greater width than the other marks. The marks are to be put only where they will be needed: where a pipe branches, in passages where the pipes go through a wall and in the pipe shafts. It may be well to have at least one mark on each pipe in every room.



1. Steam Lines at Boiler Pressure.
2. Steam Lines at Reduced Pressure.
3. Exhaust Steam.
4. Drains from Headers to Pumps or Traps.
5. Drain-pump or Trap Discharge to Boiler.
6. Drain-pump or Trap Discharge to Heater or Hot Well.
7. Trap Discharge to Sewer.
8. Hot Boiler Feed.
9. Cold Boiler Feed.
10. Blowoff
11. City Water from Meter where Pressure or Temp. is changed.
12. City Water Discharge from Oil Coolers or Water Jackets to Hot Well.
13. Fire
14. Engine Oil.
15. Cylinder Oil.
16. Oil returns to Filters.
17. Compressed Air.
18. Main Heating Line and Loop.
19. Heating Risers or Stand Pipes.
20. Returns from Heating System.
21. Discharge from House-Pumps to Roof Tanks.
22. Roof Tanks to House-Service Heater
23. Discharge from House-Service Heater
24. Cold Water Loop and Stand Pipes.
25. Hot Water Loops and Stand Pipes.
26. Pressure Pipes between Elevator Pumps and Operating Valves
27. Discharge Pipes between Operating Valves and Tank
28. Discharge from Pilot Valves.

SCHEME FOR PIPE IDENTIFICATION

The high-pressure steam, reduced-pressure steam and exhaust-steam lines (Nos. 1, 2 and 3) are simple, and there is no danger of getting them mixed. All drain lines between pipes or headers, and the drain pumps or traps are marked the same as the pipes which they drain with the addition of a narrow black band. This band is in back of the marker (No. 4).

Where the drain pumps or traps discharge to the boiler direct, the pipe is marked the same as the drain pipe with the addition of a black band ahead of the mark (No. 5).

Where the traps discharge to the hot well or feed-water heater the marks are touched with red and black (No. 6).

Where traps discharge to the sewer, the marks are touched with red, ahead of the marks (No. 7). The hot feed water, cold feed water, blowoff and city water pipes (No. 8, 9, 10 and 11) cannot easily be mistaken.

Red lead is the best paint for fire lines if they are of black iron. A great many plants already have them painted so.

These marks can be put on a large pipe system at small expense, and in the engine room, pipe shafts and around the building they would last for a long time. In the boiler room and basement they would not last so long, but they can be renewed very cheaply. When they are once in place any fireman or handy man can renew them. Charts with pipe marks and explanations should be printed.

C. G. HARDEN.

Chicago, Ill.

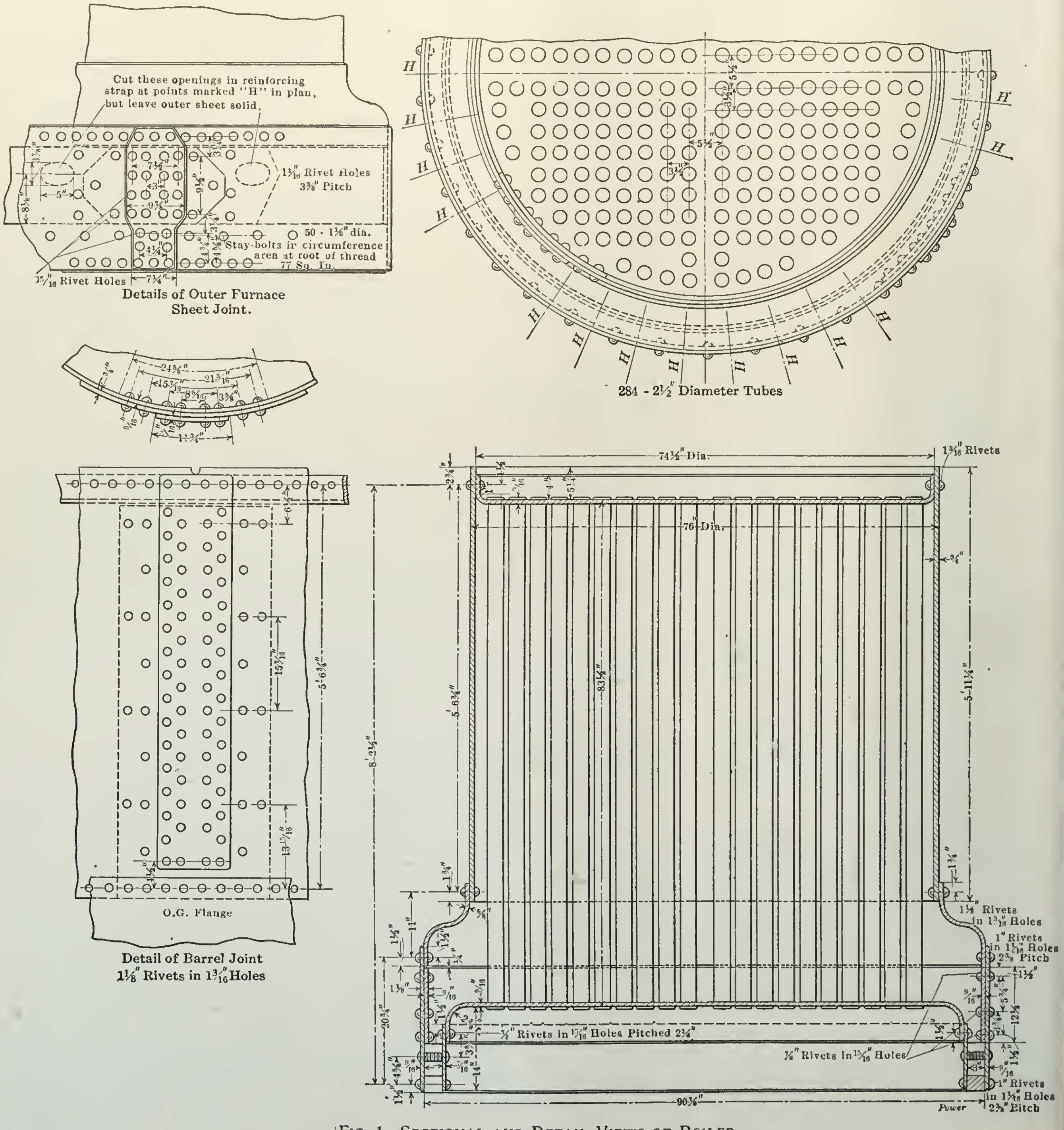
Interesting Test of Manning Boiler

In the Manning boiler there is a considerable vertical distance between the lower tube sheet, and the ogee which connects the outer sheet of the water leg to the barrel. This distance is necessary for the location of handholes which must be on a level with the crownsheet. The condition will be evident from an inspection of the sectional drawing, where, however, no handhole is shown, and is described by S. F. Jeter, M.E., in an article entitled "Features of the Manning Boiler," in our issue of February 15 of this year.

> *To demonstrate the reinforcing effect of a continuous strap riveted to the inside of the sheet extending from the lower edge of the ogee to the top row of stay bolts, hydraulic pressure was applied. The reinforced section showed considerably less distortion than the barrel.*

In the water leg it is desirable that thin material should be used for the outer as well as for the inner sheet, to afford flexibility on each end of the staybolts. The connection of the inner and outer sheets by the staybolts reduces the tensile stress on the outer sheet and makes the use of a thin sheet allowable for the stayed portion. In order to strengthen the portion above the stays it is the practice of the Bigelow company to rivet a continuous strap to the inside of the sheet extending from the lower edge of the ogee to the top row of staybolts.

Notwithstanding this construction has shown no distress in service and has been approved by the boiler-insurance com-


Cut these openings in reinforcing strap at points marked "H" in plan, but leave outer sheet solid.
1⅟₁₆" Rivet Holes 3⅜" Pitch
50 - 1⅛" dia. Stay-bolts in circumference area at root of thread .77 Sq. In.
1⅟₁₆ Rivet Holes
H
74½" Dia.
1³⁄₁₆" Rivets
76" Dia.
8'-2½"
5'-6¾"
83½"
5'-11¼"
1⅛" Rivets in 1³⁄₁₆" Holes
1" Rivets in 1⅟₁₆" Holes 2⅞" Pitch
⅞" Rivets in 1⁵⁄₁₆" Holes Pitched 2¼"
⅞" Rivets in 1⁵⁄₁₆" Holes
1" Rivets in 1⅟₁₆" Holes 2⅞" Pitch
20¾"
90⅜"
O.G. Flange
Power

Details of Outer Furnace Sheet Joint.

284 - 2½" Diameter Tubes

Detail of Barrel Joint 1⅛" Rivets in 1³⁄₁₆" Holes

FIG. 1. SECTIONAL AND DETAIL VIEWS OF BOILER

panies and the Massachusetts Board of Boiler Rules, the criticism has been advanced that unless the joint between the plate and this strap is made perfectly tight, pressure can act between them and nullify any reinforcing effect of the strap. A common expression of this criticism is that the boiler so constructed is like a cask with the hoops inside.

To settle the matter beyond question the Bigelow company, at the suggestion of Mr. Jeter, who is their mechanical engineer, constructed the model shown in the accompanying drawing and photograph, being a section out of a full-sized boiler of 225 horsepower capacity, including the lower barrel course and one row of staybolts. In other words, the model is a full-sized 225-horsepower boiler with the bottom portion of the water leg and the top portion of the tubes left off.

This boiler was arranged with gages to measure the distortion in various parts, and on Tuesday, July 12, in the presence of various representatives of the boiler inspection and insurance companies, members of the Massachusetts Board of Boiler Rules, professors from the Sheffield Scientific School, and competing boiler manufacturers and others, it was tested by the application of hydraulic pressure. We hope in a future article to deal more particularly with the various distortions produced as the pressure was applied in increments of 50 pounds. For the present it is sufficient to say that when the pressure got a little above 700 pounds, six of the tubes and a plug which had been screwed in to close the opening between the abutting edges of the shell plates in the butt joint where it is not covered by the outer strap (as is common in making this joint) were leaking so badly as to make it difficult to increase the pressure. When the tubes had been reëxpanded and the plug calked the pressure was run up rapidly to just within 800 pounds, when one of the tubes in the outer row collapsed, opening a considerable outlet in the upper head. Another tube, also in the outer row and nearly opposite, collapsed just before but did not produce any opening.

The essential feature of the test and that which is justifiably gratifying to the makers of the boiler is that the distortion of the reinforced section was very much less than that of the barrel, of the safety of which there is no question. The gage in the middle of the barrel showed that its circumference increased 1.98 inches, while the circumference of the water leg at the center of the reinforcing strap increased only 0.24 of an inch. The distortion of the barrel was thus about eight times as much as that of the construction under investigation. A gage applied just above the staybolts showed no distortion whatever, showing not only that the properties here are perfectly safe but that no effect was produced by leaving off the lower rows of staybolts in the tested boiler.

The tubes were 2½ inches in diameter and of standard thickness, but had been used before. There was no change of shape either of the barrel or firebox section under pressure in the region of the joint. Wooden forms made to span the joint and fit the boiler near it before pressure was applied, fitted equally well during and after the test.

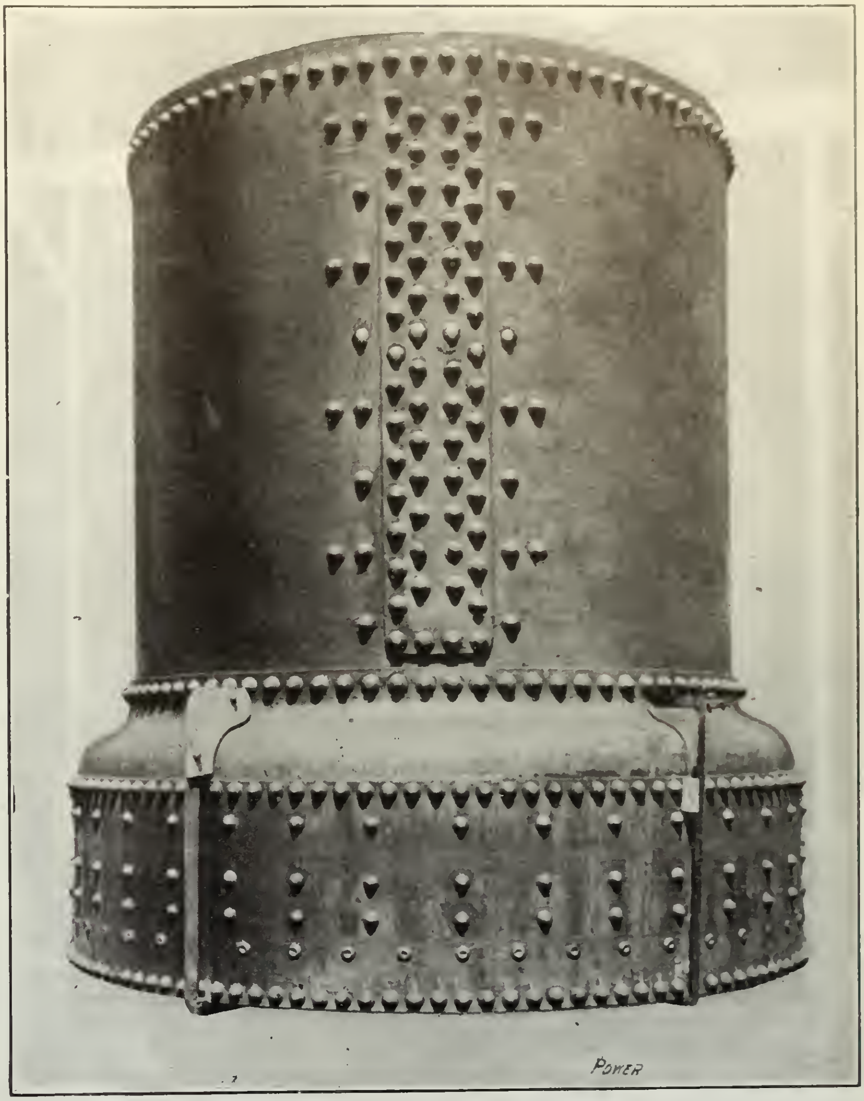


Fig. 2. The Boiler as Prepared for Testing

Unique among the events in which public-utility corporations participate, was a banquet given by the Chamber of Commerce, of Long Beach, Cal., to the officers and executive staff of the Southern California Edison Company at Hotel Virginia, Long Beach.

It was in celebration of the selection by the company of a site on the harbor of this Pacific coast port for a steam-generating plant which will ultimately be one of the largest in the country.

The banquet was attended by about two hundred and fifty citizens. In the addresses expressions of hearty good-will toward the company and a hearty appreciation of what it is doing for Long Beach and southern California, were features that have few parallels in the relations between commercial organizations and corporations.

In describing the proposed plant, W. A. Brackenridge, vice-president and general manager of the Southern California Edison Company, said: "The first building will be of sufficient size to contain two turbo-generating units with a total output capacity of 40,000 horsepower.

"Contracts have been let for a part of the machinery and boilers. These are substantially all of the plans that have been definitely decided upon.

"It is expected that the machinery now under contract will be installed and ready for operation about the first of the coming year."

Tubular Boiler Explodes in Laundry

By F. L. Johnson

A boiler built of inferior stock, carrying four times its safe working pressure, explodes; completely wrecking one three-story building and damaging several others. One man was killed and five other persons were seriously injured.

Just before six o'clock on the afternoon of July 5, as the employees of the Lakeport steam laundry, Laconia, N. H., were preparing to leave for home, the boiler exploded, wrecking the entire structure, which was a three-story building. One man was killed by a large water tank which fell from the roof, and seven other persons, two men and five women, were seriously though not fatally injured by falling timbers and flying pieces of the wreck.

The damage was not confined to the laundry building alone, for the force of the explosion shattered the rear wall of an adjoining brick building, blew out the entire plate-glass front of two stores, wrecked a pool room and a barber's shop. One man was thrown through a plate-glass window and clear across the street, while his wife, standing nearby, was uninjured.

As the work for the day was over, the various outlets for steam were closed and the heat in the furnace and boiler setting was sufficient to raise the steam pressure beyond the safe limit. The safety valve, which has not at this writing been found, did not blow. But it is stated on good authority that it was set at 85 pounds and that it blew freely on the Friday before the day of the explosion.

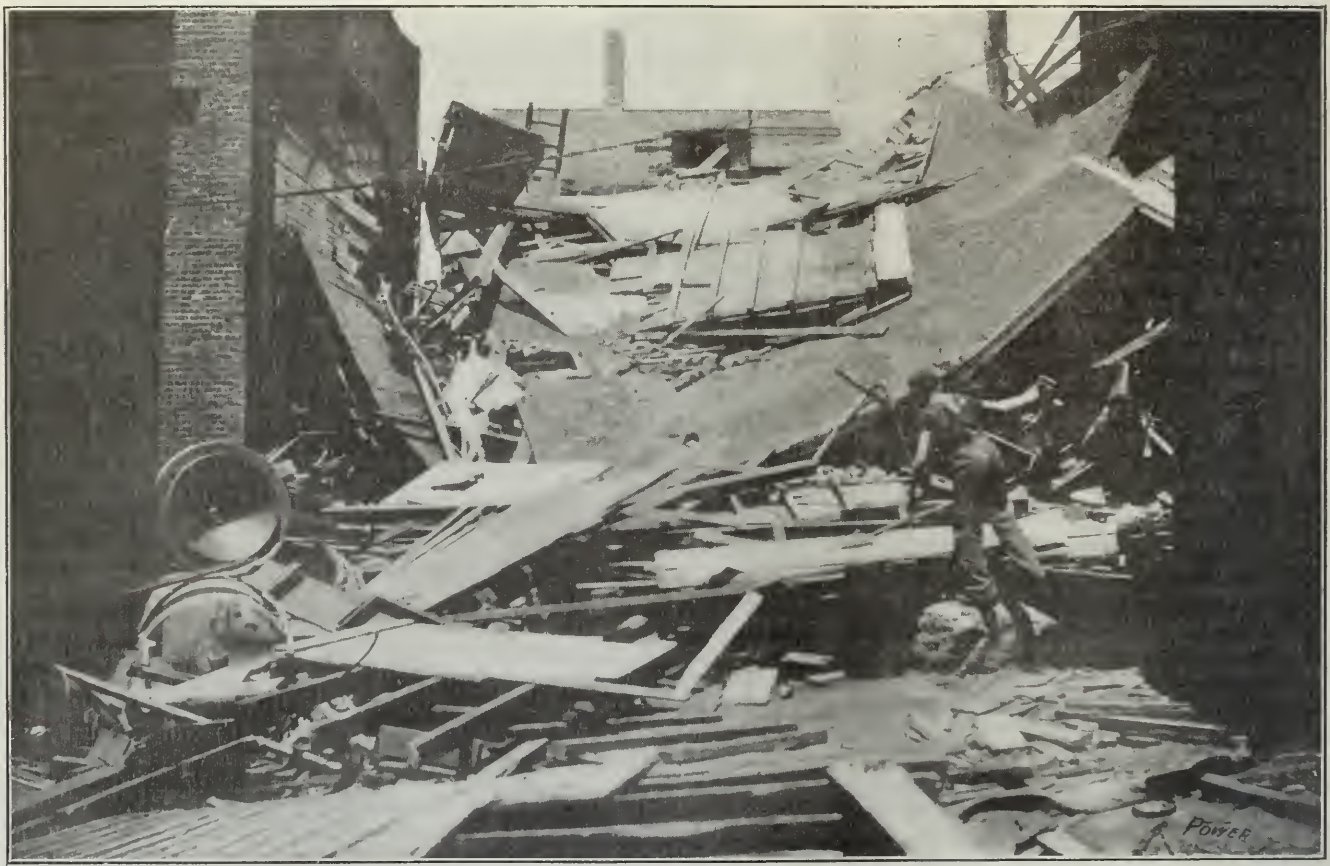


Fig. 1. Appearance of Laundry Building After Explosion

It is probable that the initial rupture started near the middle of the length of the boiler in the longitudinal seam, as the greater part of the shell sheet comprising the front, middle and a part of the rear course, opened outward and nearly flat, was blown directly through a brick wall parallel to and near the boiler setting. Part of the rear course, which was torn completely around the entire circumference, through solid metal, and the rear head, holding some of the tubes, were driven through the partition into the laundry office and turned through about 90 degrees and lay at right angles to their position in the setting.

Portions of the front were blown down an alley about 200 feet, ricochetting from one side to the other. The front head and the tubes that went with it were carried diagonally to the front corner of the boiler room where they were buried in the wreck of the two floors above.

The boiler, which was built, it is said, by the Lowell Boiler Works, about 15 years ago, was 36 inches in diameter by 9 feet long, was of 1/4-inch plates with 5/16-inch heads and thirty-four 2 1/2-inch tubes. The segment above the tubes was stiffened by a flat bar 4 inches wide and 1/2 inch thick riveted to the head about 3 inches above the tubes.

It is not probable that the boiler was designed for the 85 pounds pressure under which it has been operated for the past ten years.

In the longitudinal single-riveted seams the pitch of the 11/16-inch rivets was 2 inches, giving, if the rivets were of an equal strength with the plate, an efficiency of over 65 per cent. for the strength of the joint.

The first illustration, Fig. 1, shows the wreck of the building under which the parts of the boiler were buried. At the left is the brick wall through which the shell was blown. The photograph, Fig. 2, shows the part of the rear course from which the rest of the shell was torn through solid metal, the front head and five tubes which went with it. The small cut, Fig. 3, is an enlargement of the

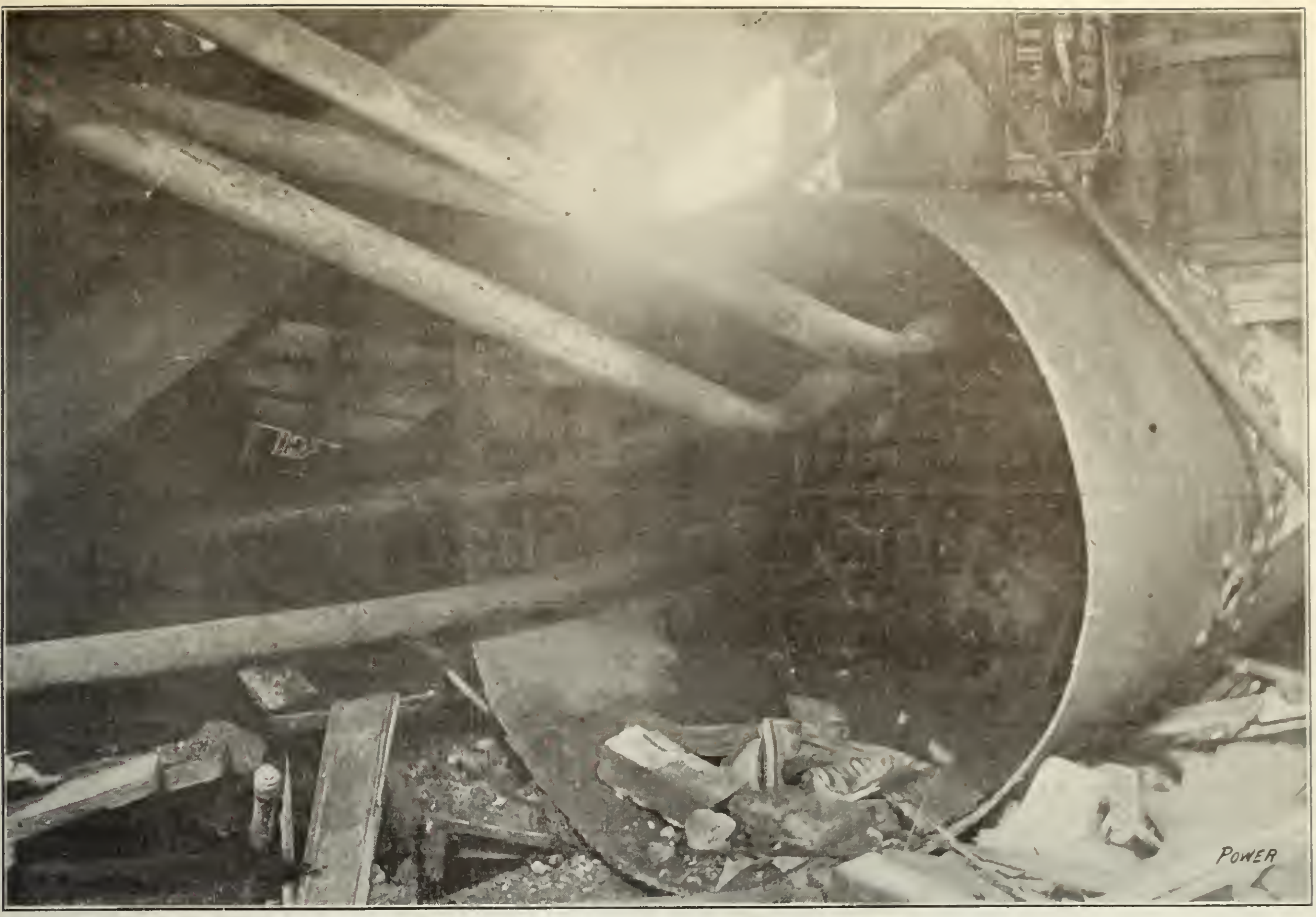


FIG. 2. REAR COURSE AND HEAD TORN FROM SHELL SHEET THROUGH SOLID METAL

fracture of a test piece from the shell sheet which failed at 24,800 pounds stress per square inch.

It was impossible to get a photograph of the sheet which was blown through the walls. But from the exposed edges, however, several pieces were obtained. To get these cutting was unnecessary as with an 18-inch Stillson wrench pieces 6 inches in length were easily broken from the sheet. Test pieces were made from these and pulled apart at the engineering laboratory of Columbia University. The elastic limit was low, the pieces showing no elongation and snapping like glass under the test, some of them having a tensile strength of only 24,800 pounds per square inch.

Though with some justice New Hampshire has been called the dumping ground for condemned Massachusetts boilers, this one had not, like some others which have expolded in the State, been condemned by the Massachusetts inspectors. But it is needless to say that such a boiler would not be allowed to operate under similar conditions in Massachusetts, for it was old and unsafe, built of tank iron and with single-riveted lap longitudinal seams.

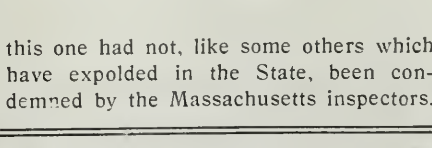


FIG. 3. ENLARGED SECTION OF TEST PIECE WHICH FAILED AT 24,800 POUNDS STRESS

There is no mystery about the cause of this explosion. It was a case of simple overpressure. There was no evidence of low water, leaking joints, scale or excessive corrosion. Except for the unwarranted pressure carried at times the boiler had the best of care and the fact that not a single rivet was sheared while the plate tore as readily through the solid metal as along the seam shows like the physical test that the shell plates were of remarkablly poor quality.

Central Station Heating*

By F. H. Stevens

For a number of years after its introduction, little was known as to the proper methods of installing central-heating systems, and to this lack of information a number of failures were due. But today, with the modern methods of construction and management and a knowledge of the different materials used for insulating the distributing mains, many heating plants are operating successfully and are producing attractive returns upon the investment.

It ill behooves the manager of any electric-light or power plant whose station is located within a mile of the business section of a city to ignore the subject and consider district heating prohibitive in his particular case without first getting definite information from some reliable source.

Almost from the first, the advocates of central-station heating have had the opposition of the consulting engineer and the engine builder, and it is only lately that these have been forced to recognize the system as an important revenue producer. The engine builder's objection lay in the fact that it raised the water

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*From a paper read at a meeting of the National District Heating Association, at Toledo, O.

[illegible] of the engine a considerable amount, sometimes doubling the steam consumption per kilowatt-hour.

This at first might seem to justify the opposition, but let us carry the investigation a little farther. Assume, for the sake of comparison, that the electric load is 5000 kilowatts and that the prime movers are compound Corliss engines, the manufacturer's guarantee being 20 pounds of steam per kilowatt-hour when operating with a 26-inch vacuum. This would make a total consumption of 100,000 pounds of steam per hour.

Now let us assume that the engines are to be operated noncondensing and exhaust into a heating system, resulting in a maximum back pressure of five pounds; and that the consumption of steam by the engine will be 45 pounds per kilowatt-hour, or a total of 225,000 pounds per hour.

In either case there are 5000 kilowatts of electric current for sale, which at 5 cents per kilowatt-hour would represent an hourly revenue of $250.

In the case of the plant operating with a heating system, we have in addition to the $250 revenue from the sale of electric current, 225,000 pounds of steam for heating, the sale of which at a rate of 60 cents per 1000 pounds would produce $135, making a total revenue of $385.

As to the additional cost with the heating system, it required 125,000 pounds more steam to produce 5000 kilowatt-hours of current. Assuming that the coal costs $2.50 per ton, water 5 cents per 1000 gallons, and firing 30 cents per ton, with an evaporative efficiency of 7 pounds of water per pound of coal burned, the manufacturing cost of steam would be 21 cents per 1000 pounds, or a total extra cost of $26.25. These figures show a net gain of $108.75 in favor of the heating system.

The above is also true of plants of smaller size and the ratio of increase in revenue is approximately the same. The best results are obtained where due consideration is to be given the heating system when selecting the prime mover to be installed.

Ball Bearing Line Shaft Hangers

Part of the discussion following the reading of a paper by Henry Hess at the June meeting of the American Society of Mechanical Engineers. The original paper was presented in the June 21 number of POWER.

Among other papers presented at the June meeting of the American Society of Mechanical Engineers was one on "Ball-bearing Line-shaft Hangers," by Henry Hess, an abstract of which was printed in the June 21 issue of POWER. Following is part of the discussion upon this paper which took place at the meeting:

F. B. GILBRETH—Is there any information regarding the manner in which concrete buildings and so called mill buildings affect ball-bearing hangers? Many engineers believe that the lack of vibration in the concrete building saves considerable cost of maintenance in line shafting, hangers and bearings.

MR. HESS—The relative necessity of frequent realinement in the case of mill buildings as compared with the more rigid concrete buildings, answers itself. That has nothing to do with roller bearings or plain bearings, as such; when a building moves, the other things have to be adjusted.

C. W. BAKER—Do the ball bearings stand the dust present in concrete buildings better than plain bearings?

MR. HESS—If dust is allowed to get into a ball bearing it will be destroyed, as would any other bearing. It is, however, a very simple matter to keep dust out of a ball bearing. As to the effect of dust in concrete buildings, experience has shown that it does not get into the bearing. Regarding lubrication, I might say that we have in our own shops line shafts which have been lubricated only three times in five years. It is not a question of the amount of lubricant but a question of its becoming hard and gummy, which is no fault of the bearings. We have found pale-yellow vaseline to be a good lubricant in certain classes of service.

MR. JACKSON—It is stated in Mr. Hess' paper that the ball-bearing hanger is as safe and reliable at 600 revolutions per minute as at 200, whereas the use of such high speeds with plain bearings is beset with so many difficulties in the way of reduction in size of line shafting, pulleys, drop hangers, etc., as to take it practically out of consideration. I think these difficulties would apply equally to ball bearings.

MR. HESS—There is no reason why a plain-bearing line shaft cannot be run at a high speed, but it necessitates great care in maintenance. You can run a ball bearing at 600 revolutions per minute or at 1600 revolutions per minute and pay no attention to it from year's end to year's end, but I doubt if there is a man present who would be willing to run a plain-bearing line shaft at 600 revolutions per minute without giving it frequent attention with regard to alinement and other features which are liable to go wrong and give trouble.

Proposed N. Y. State License Law

Features of the bill now before the New York Legislature providing for the division of the State into districts, appointment of chief and district examiners, appeals from examiner's rulings, grading of licenses, etc.

While the laws of New York allow municipalities to enact ordinances governing the inspection of boilers and the licensing of engineers and firemen, no provisions have been made for the operation by competent men of boilers and engines in all the territory not embraced within the limits of a few cities. That this state of affairs is wrong has long been the opinion of the members of the National Association of Stationary Engineers and in an attempt to improve the condition of its members and conduce toward safety in the operation of steam-power plants, the License Committee of the State Association prepared and had presented to the General Assembly at Albany a proposed Engineer's and Fireman's License Law, which embodies the thought of the committee as to what will best serve the interests of the operator and the public.

With the usual exceptions of low-pressure heating boilers, agricultural engines, locomotives, boilers and engines under the jurisdiction of the State and of the United States all engine and boilers must be operated by licensed men only. The State is to be divided into districts, each of which will be under the supervision of a district examiner appointed by a chief examiner who is himself to be appointed by the Governor with the consent of the Senate and who shall hold office for six years or until removed for just cause, vacancies caused by death, removal from office or resignation are to be filled in the same manner as provided for in the original appointment.

The salary of the chief examiner is to be $2500 per year and that of the district examiners to be $1800 each.

An applicant for license as engineer or fireman must be an American citizen or must have declared his intention to become such and have had not less than three years' practical experience as engineer or fireman or have served three years as an apprentice at engine, boiler or pump construction and one year in boiler or engine room.

Regular licenses continue in force for one year and may be renewed from year to year without examination.

Three dollars is the fee to be paid for an original license and two dollars for each annual renewal.

All engineers and firemen not included in the exemption clause and operating engines and boilers within the State will be notified by the district examiner to apply for license and will be given sixty days in which to prepare for such examination for license.

Any engineer operating under a local municipal license may upon payment of the regular fee have issued to him a license of the same grade as the local license.

Any person dissatisfied with any ruling of a district examiner may upon the payment of ten dollars appeal to the chief examiner. A jury of five licensed engineers, two chosen by the aggrieved, two by the district examiner and one by the four already selected shall decide by majority vote the justice of the decision of the examiner and in case the ruling of the examiner be reversed the money paid by the appellant shall be returned to him. Five dollars per day will be paid to each member of the jury and their decision will be final.

Four grades of license are to be granted:

Chief unlimited in horsepower.

First class. To have charge of and operate any boiler or boilers and any engine not exceeding 150 horsepower.

Second class. To have charge of and operate any boiler or boilers and any engine not exceeding 75 horsepower.

Third class. A special license to operate any particular plant provided that the owner of such plant makes the application and shows that a duly licensed engineer cannot be procured. Such license may be issued for a term of thirty days and may not be renewed.

Two years' experience in any lower grade is necessary for an examination for the next higher.

The term "have charge" is defined and construed to refer to the person under whose supervision any boiler or engine is operated.

The horsepower of any boiler is rated at ⅓ square foot of grate surface per horsepower for a power boiler, and 1½ square feet for a heating boiler.

Engine horsepower is based on 40 pounds mean effective pressure for a simple engine, 50 pounds for a condensing engine and 75 pounds on the high-pressure piston of a compound engine.

Operation of boilers and engines by unlicensed persons is a violation of the law and both owner and operator are liable to punishment by fine or imprisonment or both.

Marine engineers holding United States license may be given examination and granted a license of the same relative grade as the one held.

Any engineer who has for three consecutive years operated any plant may be granted a license of the grade required for such plant without examination.

Clean and Near Clean Boiler Tubes

It is of course well known that scale on the heating surface of boiler shells or tubes reduces the heat-transmitting qualities, but of the extent of this effect little is actually known.

RESULTS OF TESTS SHOWING INFLUENCE OF SCALE.

Coal. Eight parts No. 3 buckwheat One part soft	Ordinary Firing.		Forced Firing.	
	Before Internal Cleaning.	After Internal Cleaning.	Before Internal Cleaning.	After Internal Cleaning.
Hours run	5½	5½	5	5
Water used	55,661	92,015	89,981	114,965
Water evaporated per hour	10,120	16,730	17,996	22,993
Equivalent evaporation per hour	10,767	17,802	19.204	24,466
Average horsepower	312	513	558	709
Coal burned, total	9,670	12,945	14,830	16,875
Coal burned per hour	1,760	2,351	2,966	3,375
Pounds water evaporated per pound of coal as fired	5.75	7.11	6.00	6.81
Equivalent water evaporated per pound of coal	6.12	7.56	6.47	7.25
Pounds of coal burned per square foot of grate	11.6	15 60	19.7	22.4
Average steam pressure	175	180	180	180
Average steam temperature	493	467	531	483
Average temperature at gauge pressure	378.8	379.3	379.3	379.3
Degrees of superheat	114	87.7	151.7	103.7
Average flue temperature	492	457	608	509
Average feed water temperature	202	202	200	202
Average room temperature	70	79	69	84
Average temperature of air supply	56	66	56	69
Average air pressure in ash pit	0.24 inch	0.35 inch	1.10	0.85
Flue draft	0.50 inch	0.45 inch	0.50 inch	0.40
Average price of fuel per ton as fired	$1.772	$1.772	$1.772	$1.772
Cost to evaporate 1000 pounds of water (cents)	12.99	10.59	12.44	10.97

Along this line the experience of a central-station engineer may be of interest.

In the boiler room of a public-service corporation there was a light coat of scale over the inner surface of the tubes of the 400-horsepower water-tube boilers. It was not thought that the thickness of the scale was enough to make any measurable difference in the efficiency of the boiler and to prove the truth of this thought a series of tests was made. Two sets of conditions obtained in the plant. One in which for a great part of the day the boilers were worked below their rating and the other where they were forced to near the limit of steam-producing capacity.

Except for the difference in heat transference due to the presence and absence of internal scale the conditions were as nearly identical in all of the tests as possible.

The results though not exactly as expected were of great value, showing in a marked manner the difference in heat-transferring qualities between clean and near clean boiler tubes.

In the last line of the report of the test given herewith will be found the summary which speaks for itself.

What He Wanted

Years ago an old lumberman who had made a comfortable "pile" in the lumber business in Wisconsin decided to put up a new mill near Marinette. For the engine he went to Milwaukee and this was how he specified what he wanted:

"Now, Mr. Reynolds, I don't care whether the engine has paint on it or not, but what I do want is an engine which will stand up to its work and have a good healthy exhaust."

That time-worn maxim, "look after the little things and the big things will look after themselves," can find no better field for application than in the engine room.

Inquiries of General Interest

Heating Surface and Boilers

How is the heating surface of tubular and water-tube boilers calculated?

J. C. S.

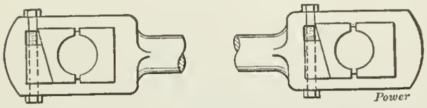
By finding the area of all the surface which is exposed to the hot gases of combustion on one side and covered by water on the other. In all tubes the inside area is taken as the heating surface.

Does Keying Alter Rod Length?

I have a connecting rod in which the keying is done by wedge blocks, as shown in the sketch. Does keying alter the length of this rod?

J. I. C.

If the wear at both ends of the rod is equal, the length will not be changed by keying. But if the wear is greater at one end than at the other, the length of the rod will change every time it is keyed. Suppose wear to take place at the crosshead end of the rod, which in the sketch is at the right. It is seen that the movable half of the box must to take up the wear travel toward the end of the rod. This would carry the crosshead pin further from the crank pin and thus lengthen the rod. If wear takes place at the crank end, keying will bring the crank and crosshead pins nearer together and thus shorten the rod.



Safe Pressure on Stayed Surfaces

In a 60-inch boiler the distance from the tubes to the top of the shell is 23 inches. There are 10 stays, each good for a working stress of 6000 pounds. What is the safe working pressure of the head?

E. L. R.

The area supported by the stays is 668 square inches or 66.8 square inches for each stay. As the stress on each stay must not exceed 6000 pounds the safe working pressure on the head will be as many pounds per square inch as 66.8 is contained in 6000, or 89.82.

Valve Setting

How is the valve on the Armington and Simms engine set?

T. O.

In this engine as in all shaft-governed engines, the valve is set by lead. That is, the valve is given the proper lead at each end of the stroke. In this type of engine, beyond seeing that the valve stem is kept at the right length, the operator has no control over the valve setting.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Universal Leaks in Boiler Seams

If you had a boiler, carrying high-pressure steam, and you moved it into another building and used it for heating purposes at a low pressure of steam and in the moving you did not strain the joints, yet the joints leaked, but before moving they were tight, how would you account for it?

C. J. P.

Such an occurrence in ordinary boiler operation is so improbable that only a thorough investigation of all the conditions of setting and operation would reveal the cause.

Overhead Heating Coils

I wish to heat my shop with exhaust steam in overhead pipes. Can you give me any rules for doing it?

F. A. H.

The heating rules used by engineers provide for a certain ratio of heating surface to cubical content heated. This varies from 1 in 40 to 1 in 200 or more, according to the size of the room.

For a well built shop 1 in 150 is a fair ratio. The coils should not be less than 1¼-inch pipe and should be far enough from the ceiling to allow a free circulation of air around them. All the coils should have an angle to provide for unequal expansion and should have a pitch of 1/16 to ¼ inch per foot of length for drainage.

Grate Surface and Cylinder Volume

How many square feet of grate surface burning 15 pounds of coal per square foot per hour will be required to supply steam at 35 pounds pressure to an engine 5 feet in diameter by 11 feet stroke, making 20 revolutions per minute with 5/8 cutoff.

W. B. R.

The volume of steam used per hour will be

$$5^2 \times 0.7854 \times 0.625 \times 20 \times 60 \times 22 \text{ or } 323{,}977.5$$

cubic feet. For condensation clearance and leakage this should be increased 50 per cent. or to 485,966.25 cubic feet. One cubic foot of steam at 35 pounds gage pressure weighs 0.1198 pound, and

$$485{,}966.25 \times 0.1198 = 58{,}198.75$$

pounds. If one pound of coal evaporates eight pounds of water, 120 pounds will be evaporated per hour for each square foot of grate and it will require as many square feet of grate as 120 is contained in 58,198.75 or 485 square feet.

Chief Requisites for Boiler

What are the chief requisites of a boiler?

S. B. R.

It should be capable of developing its rated horsepower with easy firing, a moderate draft and average fuel, and in cases of emergency permit of being forced to at least 50 per cent. above its rating.

Copper and Brass Pipe

Where should copper or brass pipe and fittings be used in a modern power plant?

A. B. E.

Copper pipe should not be used at all. Brass pipe and fittings should be used for the feed-water line and in all small unions brass is preferable.

Location of Storage Batteries

I would like a few suggestions on the placing of storage batteries and method of finding proper number of cells.

H. L. L.

Storage batteries should be located in a room separate from everything else, to prevent damage to other apparatus by the acid fumes. The room must be unusually well ventilated and should be kept at a fairly uniform temperature of about 60 to 75 degrees if possible.

The total number of cells in a battery should be 0.54 × the voltage required; thus, for 110 volts, use 60 cells; for 240 volts use 130 cells, and so on. Some of these must be "end cells," to be cut in and out during discharge and charge. If you use an automatic booster, then the number of cells need be only 0.4 × voltage.

Speed of Piston and Crank Pin

Does the crank pin of an engine travel faster than the piston and if so how much and why?

C. M. E.

The crank pin travels over the semi-circumference of a circle in the same time that the piston covers the diameter. The semi-circumference of a circle is 1.5708 times the length of its diameter and the average speed of the travel of each is as 1 to 1.5708.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for June, 1910, *was* 139,000.

July 5	34,500
July 12	34,500
July 19	34,500
July 26	34,500

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

Boiler Explosions

Steam-boiler explosions in this country occurring at the alarming rate of about two for every working day of the year furnish more than justification for continued, but thus far futile, appeals to the legislatures of the several States and to Congress for the enactment of compulsory boiler-inspection laws.

In most of the States a boiler of any quality of material or degree of workmanship may be installed anywhere and may be operated by men of any or no degree of intelligence or experience.

In the matter of roads, bridges, water supply, contagious diseases and in a thousand and one other matters affecting the safety of life, limb and property, society protects its members. If a man or a corporation should seek to establish a powder mill in the heart of a thickly settled community the measures taken to prevent it would be neither uncertain nor dilatory. If a man should be found suffering from a contagious though not necessarily fatal disease, such as smallpox or yellow fever, his isolation would be immediate and complete.

But a boiler with its possibilities of destruction on a par with those of the powder mill and exceeding those of a polluted water supply or a victim of contagion may be placed without a single protest where men, women and children are thickest, and with its operation and upkeep intrusted, as is all too frequently the case, to one utterly unfit, its ultimate failure is the surest thing on earth.

When the state of affairs existing in this country is compared with that in England, it becomes disheartening and this feeling can only be emphasized by a like comparison with the records from Germany, where in the thirty years ending with 1907 there were in the entire empire, both on land and sea, seventy-two fewer boiler explosions than took place in the United States in the single year of 1909, while for the whole period there were more than eighteen times as many failures in the United States on land alone as in Germany on both land and sea, and the number of fatalities was in still greater proportion, being more than twenty-seven to one.

Against the United States there is, in the matter of preventable boiler explosions, a terrible indictment which demands action on the part of those responsible. Complete immunity may not perhaps be possible, but, going abroad again for ways of doing things that are better than the American ways, it is found that in Manchester, England, the Steam Users Association has not in the fifty years of its existence lost a life or a boiler from a preventable accident.

This shows the possibilities of intelligent inspection in Great Britain and it is highly probable that the same exercise of brains and common sense in the United States would be productive of equally good results.

Flywheel and Pulley Rims

It would be interesting to get statistics at the present time which would show the relative mortality of flywheels and pulleys during the past ten years as compared with similar figures for the twenty-five years preceding. Flywheels do burst occasionally now, as any self-respecting wheel will do if run at a sufficiently high speed. It would seem, however, that these accidents should be less numerous as more care is certainly taken than heretofore in the design of the rims and rim joints for large wheels.

Previous to 1895, most cast-iron flywheels were made with flanged joints in the rim midway between the spokes, with a strength much inferior to that of the remainder of the rim. The designers of such wheels seemed to lose sight of the fact that the rim was subjected to a bending moment between the rims in addition to the ordinary centrifugal tension. James B. Stanwood and Prof. Gaetano Lanza were among the first to point out the nature of the stresses in the wheel rim and the presence of a dangerous bending moment at the rim joint. Experiments made by Professor Benjamin at the Case School laboratories verified the theories proposed and showed that a rim joint having an internal flange frequently had a resistance of less than one-fourth that of a solid rim. While cast-iron pulleys with whole rims frequently attained speeds before bursting of more than 400 feet per second, wheels with flanged joints failed at less than 200 feet per second. Whether as a direct result of this discussion and these experiments or not, a change in wheel design soon began to develop until at the present time we find very few flywheels having this type of joint. Where the flanged joint is used, it is now placed directly over the arm.

The gradual decline in belt transmission and the prevalence of electric generators has reduced the number of wide thin rims and brought about the introduction of the rim having great depth and little width, one which is better adapted to resist bonding and which can be joined by links or tee bolts so as to give a comparatively high efficiency. Hollow rims with reinforced joints can be made to have a theoretical efficiency of 100 per cent. Such have not, however, been tested to destruction.

Although flywheels are the principal offenders in power-house accidents, high-speed pulleys have also given more or less trouble. The solid-rim pulley, whether of steel, wood or cast iron, is perfectly safe at all ordinary speeds. The general introduction of the split pulley on shafting has brought about some bad designs and poor construction. This is particularly noticeable in the ordinary steel-rim pulley where the rim is parted midway between the arms and fastened by light ordinary lugs riveted to the rim and joined with small bolts. These joints possess almost no resistance to bending and cause the wheel to fail at comparatively low speeds.

The difficulty with wood pulleys is the method of attaching the arms to the rim, and failure usually occurs at this point.

Balance weights are another source of trouble in all kinds of high-speed pulleys. The washer of cast-iron bolted or riveted to the inside of the thin cast-iron rim promptly shears its way to freedom and frequently reduces the strength of the wheel to about one-fourth what it should be. The wrought-iron slugs which are frequently used for balancing wooden pulleys have much the same peculiarity and are usually the initial cause of rupture.

Whether we have reached the limit of speeds in rotating machinery remains to be seen. If any further increase is desirable it will be necessary to pay more careful attention to some of these peculiarities of wheels and pulleys. A linear speed of 200 feet per second should be safe for any solid rim, whether of wood or iron, and half this speed should be perfectly safe for all pulleys whether having jointed rims or not. There are, however, many wheels running today which would be in serious danger if run at the latter speed.

Some Weird Definitions

At the recent convention of the National Electric Light Association, there was presented a report on terminology by a committee composed of men who ought to know better. Among the injudicious changes and obscure definitions recommended were the following:

"Corrective Motor," instead of "rotary condenser"; meaning a synchronous motor operated with an overexcited field to improve the power factor of a circuit carrying an inductive load.

"Feeder"; restricted to power supply, instead of being applied to every circuit in which the volt-amperes are the same throughout its length (ignoring the drop in the circuit wires).

"Reactance Coil," meaning "a coil for producing difference of phase or for eliminating current."

"Quarter Phase" instead of "two phase."

It is regrettable that such an important subject should have been given the inadequate consideration indicated by the report.

Specialization

The present tendency of the industrial world is toward specialization of work and knowledge. This holds true through the various spheres of employment. The manager of a large engineering enterprise is not necessarily a man of vast knowledge, but is a specialist in the methods of organization and in the ability to obtain results through other men; just as much as the electrical engineer is a specialist in laying out the electrical equipment of a power plant, or the expert fireman is, in getting the most steam from a pound of coal.

This specialization, while tending to narrow one's field of application, nevertheless promotes a higher efficiency in the individual, and if taken advantage of, increases a man's worth to his employer. Of course, leisure hours should be devoted to the broadening of one's intellectual field and to the improvement of social conditions, but the working hours should be devoted to seeking the depths of that particular industry which a man has selected as his life's vocation.

However, one should guard against becoming a machine during working hours. It pays to keep alert every moment of the day, and if anything new or out of the ordinary is observed a mental note of it should be made, or, better still, it may be jotted down in a notebook for future reference. *Nothing is worse than neglected opportunity.*

Brooding over low pay and long hours does not improve conditions, but, like all worry, incapacitates a man for better work. He should not get discouraged if his superiors do not agree with him. It should be an incentive to determine whether the fault is his or theirs. If found to be the former, he should have the courage to see his shortcomings and set about to remedy them, keeping in mind that genuine satisfaction is attained only through honest effort.

A man should become an expert in his work, however simple or complicated it may be, as only an expert enjoys his work fully. To arrive at this stage of proficiency, it is necessary to patiently and intelligently master the details of the work, understand the principles, not being satisfied that a thing is so, but always anxious to know why it is so, accumulate experience, train the memory and improve the methods; and before long he is a proficient worker, invaluable to his employer and ready to look with pity upon the other fellow who started out with equal opportunities, but who has wasted his time from a half-hearted discharge of duty and lack of ambition.

A Worn-out Delusion

Whether or not there even existed any foundation for the absurd belief that a manufacturer can prevent his competitors from knowing what he is building we do not know, but we do know that there is not a shred of justification for that belief now. Within the last few years we have repeatedly had the experience of being unable to obtain permission to publish detailed descriptions of new apparatus or machinery because the manufacturer did not wish to give his competitors the "advantage" of ascertaining what his constructional details were; in almost every instance we have found that the principal competitors had complete working drawings of the sacred details within a few weeks after the first machine was completed.

The worst case of this kind that we recall at the moment was that of a gas-engine builder who refused permission to publish details of a new valve gear and admitted at the same time that he had in his possession complete blueprints of all the principal competing engines, and had obtained them almost as soon as they had been issued. Just why he considered himself so much smarter than his competitors in the matter of stealing blueprints, we never learned.

As a matter of cold fact, pending patent applications are the only valid excuse for withholding from publication the description of anything new. When there is no element of this kind involved, a manufacturer's attempt at secrecy almost always indicates either that he is pitiably small-minded and conceited or that he is afraid to submit his designs to the critical consideration of the engineering public.

The law as to the relation between heat and mechanical energy has been violated many times recently, for in our experience every increase in heat was accompanied by a corresponding decrease in energy.

Boiler explosions are getting to be about as common as thunder showers.

There is a great difference between the technical journal of today and that of twenty-five years ago. If you do not think so, compare your latest copy of Power with that for the corresponding month of 1885.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Naylor Patent Vortex Steam Separator and Receiver

This device fulfils the office of separation of entrained water from steam, preventing loss in the cylinder from the particles of water, which abstract heat from the cylinder walls of the engine, thereby reducing its economy and power.

This separator is constructed with very large volumetric capacity, which greatly increases the available working pressure at the engine by storing a large volume of steam adjacent to the point of use.

The velocity of the steam in the pipe is thereby lowered, reducing its friction as well as lessening pulsations of the current which cause pipe vibrations. The action of the steam in entering the separator is as follows:

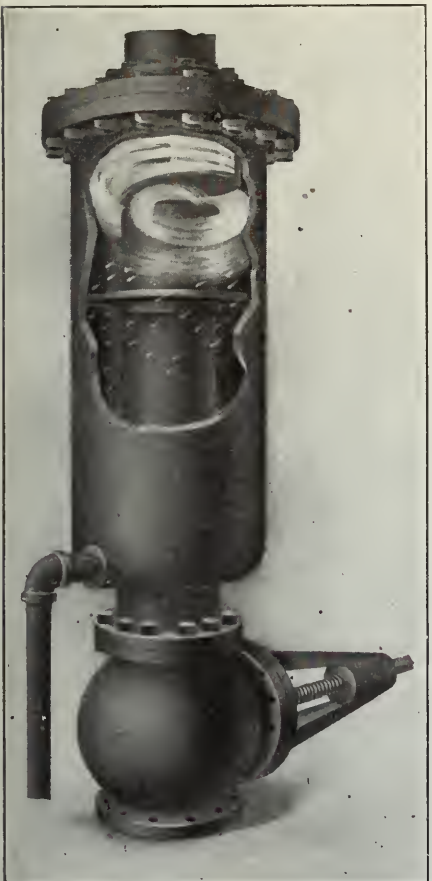


Fig. 1. Vertical Receiver Separator

Steam enters a series of four stationary buckets having tangentially curved blades and curved bottoms. The effect of these blades of double curvature is to deflect the steam from the axial line of the pipe and set it revolving freely and at great speed in a circumferential path at right angles.

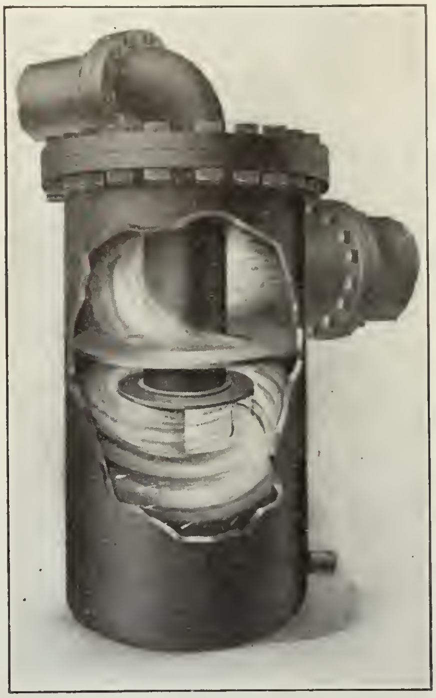


Fig. 2. Vertical Receiver Separator for Compound Engines

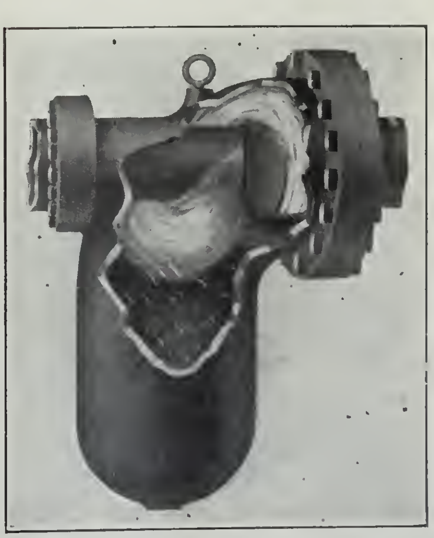


Fig. 3. Horizontal Receiver Separator

The cloud of moist steam and entrained water shown in the sectional views of the separator illustrates the whirling condition occurring in the separator.

At the center of the separator there is an extended pipe from the outlet, which takes the dry steam from the vortex of the whirl, the point at which it is of the driest possible character.

A diaphragm plate, with circumferential openings and interposed diagonal lips, permits the entrained water to pass to the lower space of the receiver where it passes out through a drainage opening.

The diaphragm plate divides the interior of the separator into two distinct chambers, the upper one being the whirling or separating chamber, while the lower one is a still chamber to receive and hold the water to be drawn off.

This separator will carry a large amount of water, but ample provision should be made to get rid of the normal flow of water as well as any extraordinary quantities which may come over from the boilers due to priming or other causes.

These separators are built of any capacity and are made both in horizontal and vertical types by Hewes & Phillips Iron Works, Newark, N. J.

Improved Tomlinson Type "C" Condenser

Since the description of the Tomlinson type C condenser in the March 22 issue, an important improvement has been made in the method employed to build up a vacuum when starting, or in case it should be lost when the turbine is running.

The Tomlinson type C condenser is supplied with two centrifugal pumps mounted on the same shaft and driven by a direct-connected motor or small steam turbine. The larger pump removes the water from the condenser and discharges it into the hotwell, while the smaller one takes water from the cold well and discharges it through a specially designed air-ejector nozzle to the hotwell, the latter acting in the same manner as a dry-air pump.

The manner of forming the vacuum on starting, or in case of loss of vacuum with this condenser, as improved, is accomplished by turning the discharge from the air-ejector pump into the main injection line between the condenser and the main injection valve, through a connection provided for that purpose and shown in the accompanying Figs. 1 and 2. This pump is of ample capacity to furnish the necessary water to form sufficient vacuum to lift the water from the cold well through the main injection line which enables the vacuum to be formed with a full load on the steam unit without shutting steam off the condenser or interfering with full-load conditions.

As the wet-vacuum pump and air-ejector pump are on the same shaft and run by the same motor, any failure to remove water from the condenser will cut off the forced injection supply and eliminate all danger of flooding the system. Furthermore, it makes an arrangement entirely self-contained and independent of any outside source of water supply, thus reducing the auxiliaries to a minimum.

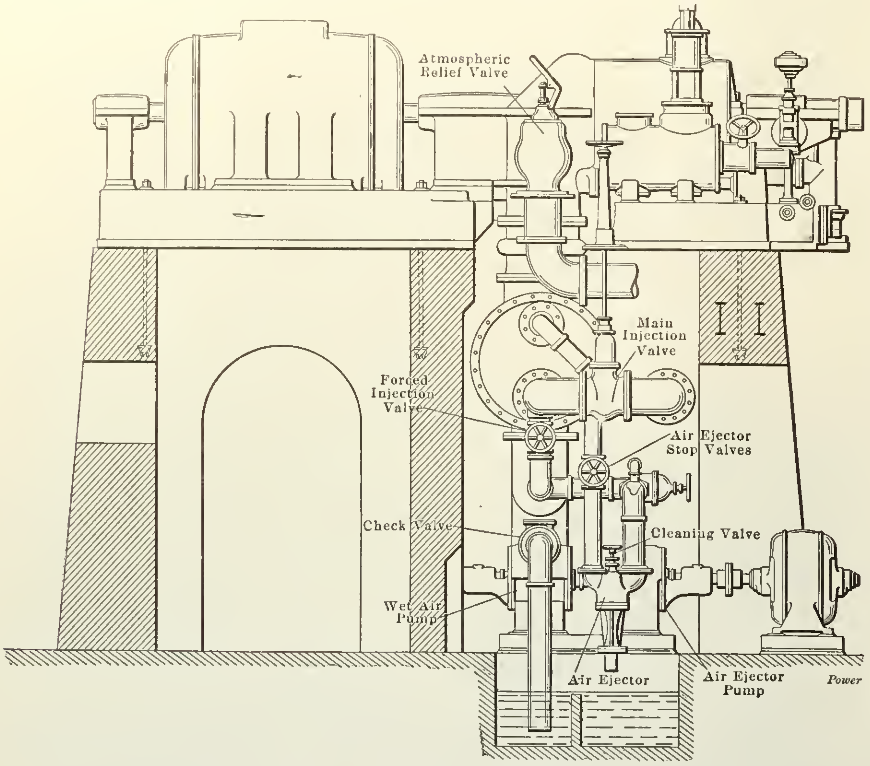


FIG. 1. SHOWING FORCED INJECTOR CONNECTION ON DISCHARGE OF DRY AIR PUMP

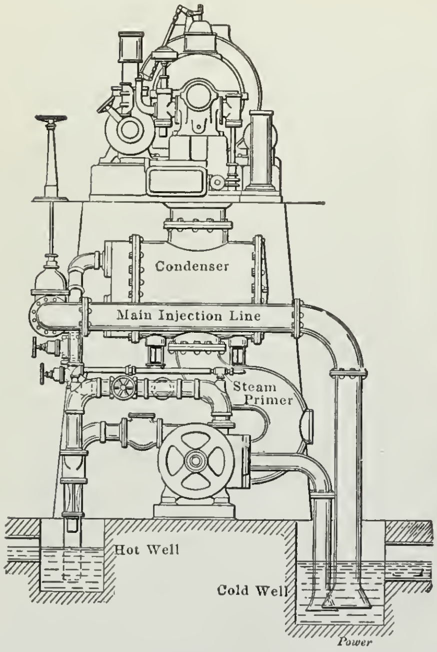


FIG. 2. END ELEVATION WITH CONNECTION SHOWN IN DOTTED LINES

The Harrington Automatic Stoker

The tubular-boiler proposition is one which gives considerable trouble when the question of automatic stokers is being considered, as plants having this type of boiler installed are frequently placed in low-ceiled, inaccessible locations, where deep excavations for ashpits are impossible, or room in front of the stoker is limited.

It was with these particular conditions in mind that the stoker illustrated herewith was developed, and the apparatus has been designed to meet the requirements for an automatic arrangement for burning coal in the smaller boiler installations. This stoker consists of a series of grate bars set at a slight incline to the horizontal, and extending from front to rear of the furnace. The rear ends of all of these bars are supported by a transverse, vertical, adjustable beam, and it is the position of these fixed bars which determines the normal grate surface. The other half of the bar is extended forward to include a yoke in which slides an eccentric block. The eccentric blocks include eccentrics on a transverse shaft, Fig. 1, which is the main drive shaft of the stoker. This shaft is operated by means of a train of cast-steel spur gears securely babbitted in the side frame of the stoker and are operated from a line shaft supported on the boiler fronts, through a sprocket chain having a maximum speed of 100 lineal feet per minute.

The forward ends of the bars are covered by a dead plate which extends the full width of the stoker and acts as a bottom to the coal hopper and a surface upon which a reciprocating pusher may travel. The hopper is designed, in connection with the pusher, so that at each stroke of the latter a certain variable amount of coal can be pushed onto the dead plate. The strokes of the pusher are made with sufficient frequency so

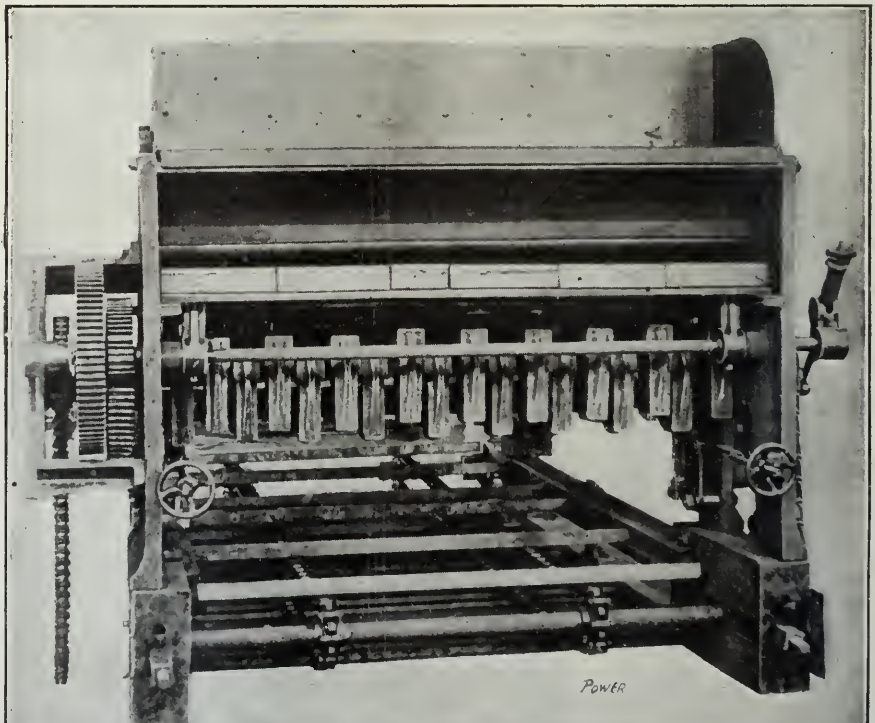


FIG. 1. FRONT VIEW OF HARRINGTON STOKER

that the flow of fuel across the dead plate and onto the grate bars is reasonably constant. During the progress of the fuel over the dead plate it is subject to the heat of the fire and becomes ignited and partially coked. The movable bars operated by the eccentrics previously mentioned can be moved either in parallel or one-half of them can be moved forward while the others are returning, according to the character of the fuel being used. The effect of this upon the fuel is to move it slightly forward at each revolution, depositing it upon the fixed bars, which under the influence of the slight incline and the resistance offered by the fuel, hold the fire about stationary while the bar is returning below the normal grate surface.

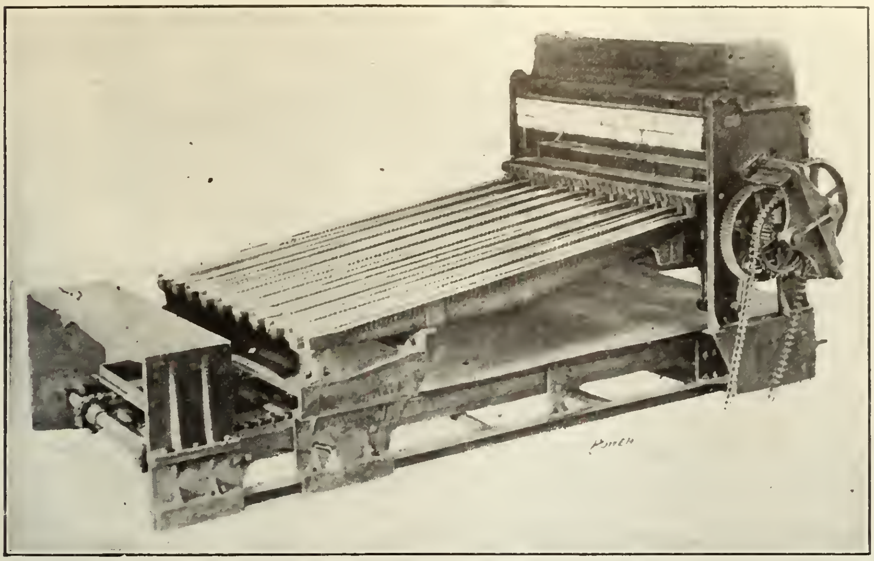


Fig. 2. Side View, Showing Driving Gears and Arrangement of Grate Bars

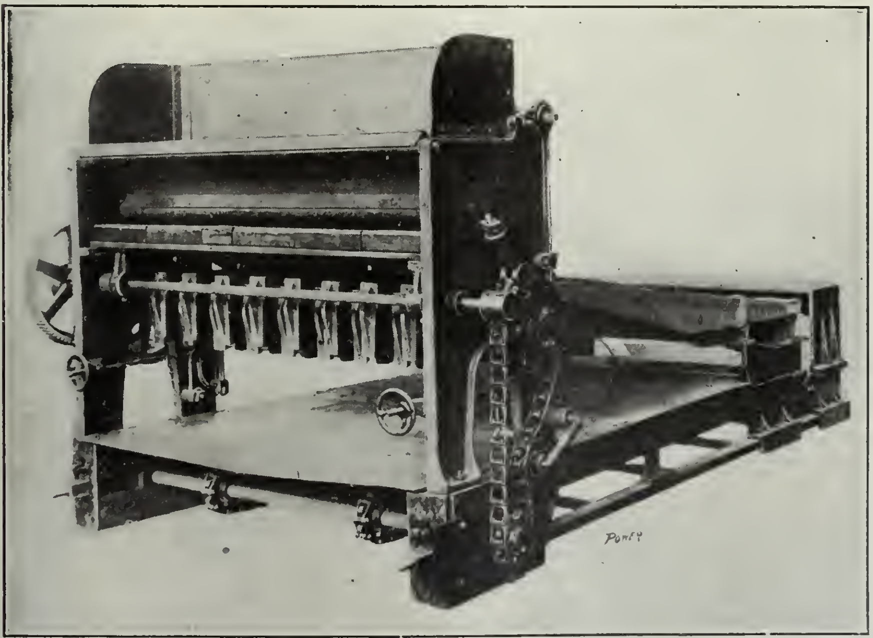


Fig. 3. Showing Chain Drive to Ash Drag.

As the fuel progresses through the furnace its volume diminishes and becomes more and more porous, requiring less motion to the fuel bed. This decrease in motion is obtained by allowing the back end of the bars to slide horizontally on their rear support as indicated in Fig. 2. A comparatively dense fire at the rear of the stoker is thus effected.

As ordinarily supplied, this stoker is provided with a mechanism for carrying the ash to the front of the setting, where it is accessible and easily removed by hand. The ash drag renders it unnecessary for a deep excavation under the boiler. The depth of the ashpit can be as little as 3 feet if necessary, which will be sufficient to contain the ash for a period of several hours.

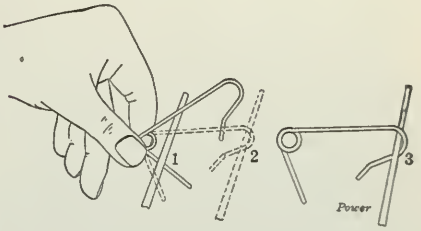
These stokers are built in five sizes in width, ranging from 3 feet 9 inches to 6 feet 2 inches, and are made in two lengths of grate bar, the standard being 7 feet in length, and the other 5 feet. The 5-foot bar is used where excessively small grate surfaces are required, the minimum possibly being 19 square feet.

The Harrington stoker is manufactured by the Green Engineering Company, Commercial National Bank building, Chicago, Ill.

Trill Indicator Hook

A handy device, known as the Trill "Catch-em-quick" indicator hook, made by the Trill Indicator Company, 34 East South street, Corry, Penn., is shown in the accompanying illustration.

It is simple and efficient, and renders the operation of connecting the indicator cord to the standard on a crosshead a very simple matter, even though the engine may be running at 400 or 500 revolutions per minute.



Trill Indicator Hook

Its operation is simple, as shown. The loop of the hook is held between the thumb and index finger, allowing the pin on the standard to strike the straight part of the hook when it is within about one inch of the end of its travel. Connection is made instantly without danger to either instrument or operator.

The largest field for future evolution in power transmission is toward high-voltage direct current. Its advantages over alternating current, considering the line alone, would be very great. The effective voltage would coincide with the maximum, and the insulator requirements would be reduced 30 per cent. The number of conductors would be reduced to two at most, and possibly one with grounded return would be satisfactory. Line-charging current and self-induction would not exist and the question of power factor would be eliminated. The whole problem of power transmission would be vastly simplified and costs would be reduced for construction.

Instruments for Alining and Leveling Shafting

Figs. 1 and 2 show a special architect's level and a portable target with self-adjusting jaws for gripping shafts which are being put on the market by the Kinkead Manufacturing Company, 7 Water street, Boston, Mass. These instruments are used for leveling and alining shafting.

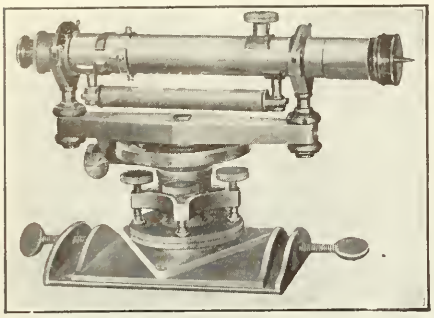


Fig. 1. Special Level

Fig. 3 shows the instrument where the target is hung from a 1-inch shaft at the driven end and moved along to a 4½-inch shaft at the driving end, which is accomplished without altering the target.

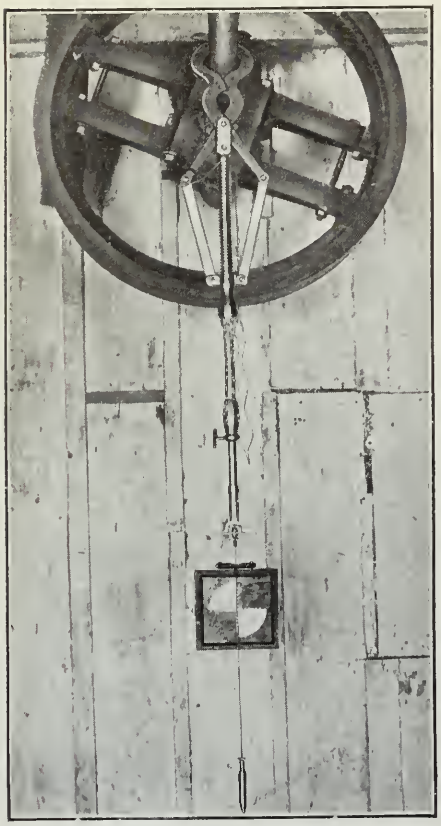


Fig. 2. The Target

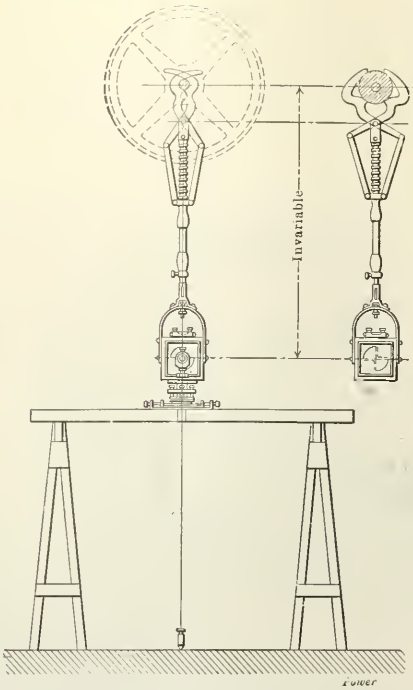


Fig. 3. Varying Sizes of Shafting without Changing Center of the Target

In Fig. 4 is shown the shafting hanging from a post or beam, with an obstruction under the center of the shafting. In such a case the portable target is hung on the shaft and adjusted free from obstructions, as shown, using a counterweight *A* on the offsets to hold the target down. Then the target is hung on the end of the shaft from which the operator is to handle the opposite level. It is leveled accurately and made plumb from the pin central over the target to the floor, after which the point is marked and the target moved 12 inches. Another mark is made and a line is drawn between the two points. The architect's level is then brought into use.

Fig. 5 shows the method of using the device where the shafting is heavy and the pulleys large, the leveling and alining being done on a horizontal system which enables the operator to work from the floor. An extra spirit level is attached to the face of the target by means of a special casting designed for the purpose. The target is hung horizontally on the shaft, supported with a stick grooved at the top, and by means of a thumb screw on the extra casting the spirit level is accurately adjusted. Then by plumbing from the center of the target a point on the floor is determined and a mark made. This operation is repeated every 12 inches along the shaft, and then the line is drawn. The architect's level is set over this line and the work of alining completed.

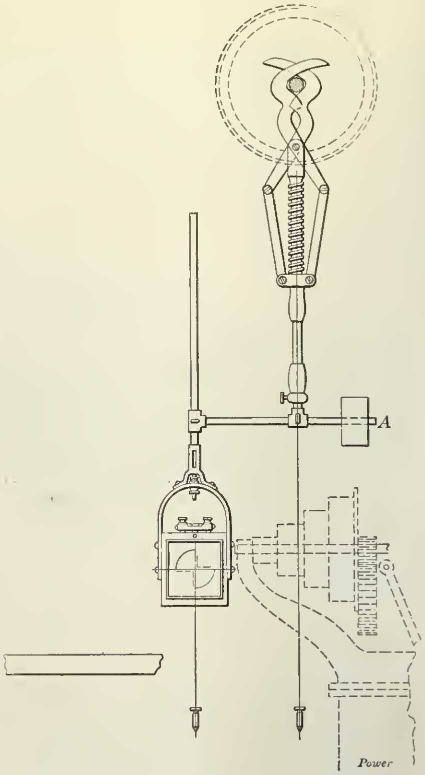


Fig. 4. Shafting Hanging from Posts or Beam with Obstruction Under Center of Shafting

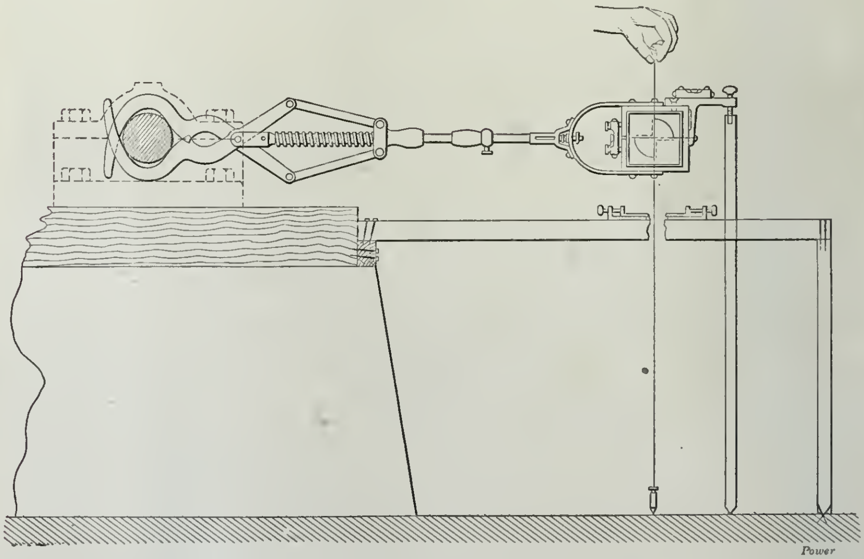


Fig. 5. Alining Pier Shafting

Mud Drum Explodes

On July 16 a cast-iron mud drum exploded on a Babcock & Wilcox boiler at the plant of the Rochester Railway and Light Company, Rochester, N. Y., wrecking the boiler and injuring three men, one of whom will die. The drum ripped through the reinforced ligaments between the tube holes. Similar explosions have probably been due to initial shrinkage strains set up in the foundry by removing part of the sand while the casting was hot, thus exposing a part of the surface to the atmosphere while the rest was covered.

NEW PUBLICATIONS

Engineering Thermodynamics. By C. F. Hirshfeld. Published by D. Van Nostrand Company, New York, 1910. Boards; 160 pages, 4x6 inches; 22 illustrations. Price, 50 cents.

This is the second edition of Professor Hirshfeld's excellent little treatise. As the reviewer is unfamiliar with the first edition the relative merits of the two cannot be discussed. The intrinsic merit of the present edition, however, is extraordinarily high, despite a few rather inexact statements. The chapter on entropy, though covering only five of the small pages of the book, is one of the most lucid and accurate expositions of the subject that we know. The author does not follow the frequent slovenly practice of saying that entropy is equal to

$$\frac{dQ}{T}$$

but carefully explains that this represents the *change* in entropy; that is, $d\phi$. True, he rather carelessly says that

$$\frac{dQ}{T}$$

"has received a name," but that is easily forgiven in view of the beautifully clean-cut discussion of the subject which follows.

The only important lapse that the reviewer has noticed is the inclusion of the mechanical equivalent of heat in the first law of thermodynamics, but this is common practice with textbook writers. Saddling a "law" with an approximation, however, does not give it the definiteness necessary to inspire respect. On page 39 the author states that work done (in foot-pounds) by the expansion of a gas is *equal* to the change in the sensible heat of the gas, which is not strictly true, of course; the formula immediately following, however, includes the coefficient (778) necessary to establish the mathematical equality.

The foregoing criticism is intended merely to indicate the small amount of effort necessary to eliminate flaws from this admirable little work; it is not due to any carping desire to find some slight excuse for adverse comment.

Massachusetts Engineers at N. A. S. E. State Convention, Report of Which Appeared in Last Issue

PERSONAL

John A. Stevens, representing boiler-using interests, and William M. Beck, representing operating engineers, were reappointed by Governor Draper to the Massachusetts Board of Boiler Rules. Their services were apparently appreciated and both men are to be congratulated.

Sylvester S. Howell has become associated with Paul M. Chamberlain, engineer, Marquette building, Chicago, under the firm name of Chamberlain & Howell. The firm will carry on the designing and consulting-engineering practice established by Mr. Chamberlain.

L. R. Pomeroy has been appointed chief engineer of the railway and industrial division of J. G. White & Co., Inc., with headquarters in New York City. Mr. Pomeroy has for a long time been considered an authority on railway-shop equipment, operation and construction, and his attainments should form a valuable addition to the organization.

OBITUARY

John Denison Evarts Duncan died at Ann Arbor, Mich., on July 13. The news of his untimely death was a great shock to his many friends in New York, where he was widely known and highly appreciated. Mr. Duncan had been continuously associated with Sanderson & Porter, engineers and contractors, since February, 1901, and had acted as managing engineer of their New York office for the last six years. He was 39 years old. Mr. Duncan was a member of the American Society of Mechanical Engineers, the American Institute of Electrical Engineers, the Brooklyn Engineers Club, the Engineers Club of New York, the Machinery Club of the City of New York, the Michigan Club of New York, the University of Michigan Club, and the Cornell University Club.

SOCIETY NOTES

Rhode Island No. 2 National Association of Stationary Engineers will hold its fifteenth anniversary reunion, field day, shore dinner and entertainment at Palace gardens, Narragansett bay, on Sunday, July 31, 1910.

A genuine old-fashioned Rhode Island shore dinner will be served and everything which the judgment and foresight of the committee can devise for the entertainment of guests will be done. This anniversary is the New England event in National Association of Stationary Engineers' social functions and is sure to be largely attended as members and friends of the order are always guaranteed their "money's worth."

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

FLUID MOTOR. Archibald Kelso, Pittsburg, Penn. 963,581.

FURNACE. Albert A. E. Sterzing, New York, N. Y. 963,088.

MECHANICAL STOKER. William M. Duncan, Alton, Ill. 963,532.

SMOKE PREVENTER. Latimer H. Long, Louisville, Ky. 963,602.

FIRE BOX FOR BOILERS. James M. McClellon, Everett, Mass. 963,627.

GAS ENGINE. Frederick O. Kilgore, Minneapolis, Minn. 962,437.

FLUID PRESSURE IMPACT ENGINE. John Kincaid, Vancouver, British Columbia, Canada. 962,511.

HYDRAULIC MOTOR. Peter T. Coffield, Dayton, Ohio. 962,623.

MOTOR. Nathaniel Lombard, Winthrop, Mass. 962,711.

ROTARY ENGINE. Russell Clinton Leedham, Trinidad, Colo. 962,850.

STEAM ENGINE. John March, Dillon, Mont. 962,907.

ROTARY STEAM ENGINE. John Schuller and Andrew Niland, Hoquiam, Wash. 962,984.

GAS ENGINE. Samuel J. Webb, Minden, La. 962,995.

POWER PLANT AUXILIARIES AND APPLIANCES

BOILER ALARM. Alexander J. Aderhold, Birmingham, Ala. 963,028.

GOVERNING MECHANISM FOR TURBINES. Frederick Samuelson, Rugby, England, assignor to General Electric Company, a Corporation of New York. 963,074.

VAPORIZER OR CARBURETER FOR GAS ENGINES. Charles D. Shain, Rockaway Park, N. Y. 963,081.

GOVERNING MECHANISM FOR TURBINES. Charles H. Worsey, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 963,107.

VALVE. Charles W. Jones and James A. Gibson, Urbana, Ill. 963,152.

SYSTEM OF LUBRICATION FOR ELASTIC-FLUID TURBINES. Oscar Junggren, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 963,154.

CARBUREER. Frederick W. Tuerk, Chicago, Ill., assignor to the Tuerk Novelty Company, Chicago, Ill., a Corporation of South Dakota. 963,187.

SMOKE STACK. Annah Rockhill Clark, Spokane, Wash., executrix of Edward Thomas Jenkins, deceased. 963,208.

VALVE. Lewis Watson Eggleston, Chicago, Ill. 963,215.

MEANS FOR CONTROLLING THE VALVE MOVEMENT OF GATE VALVES. Dorville Libby, Jr., San Francisco, Cal., assignor to the Pelton Water Wheel Company, San Francisco, Cal., a Corporation of California. 963,230.

PIPE FITTING. Daniel S. Ramelli, New Orleans, La. 963,248.

GOVERNOR. Homer N. Motsinger, Pendleton, Ind., assignor to Motsinger Device Mfg. Co., Pendleton, Ind., a Corporation of Indiana. 963,309.

VALVE. Johnston Nolan, Philadelphia, Penn. 963,314.

STARTING VALVE FOR COMPOUND ENGINES. Wladimir Ivanovich Kovalsky, Stanzia Zdolbunoff R. R. Colony, Russia, assignor to Feodor A. Evdokimoff, New York, N. Y. 963,375.

PRESSURE REGULATOR. William Leuckert, New York, N. Y., assignor of one-half to Charles S. Hoffman, New York, N. Y 963,376.

VALVE. Olaf E. Oleson, Chicago, Ill. 963,385.

AUTOMATIC SPRING PACKING. Alfred P. Sundt, Stoughton, Wis. 963,406.

HAND-HOLE SCRAPER. Henry B. Cockrell and Heyward B. Miller, Washington, D. C. 963,421.

LUBRICATOR. Linsey Vestal Hood, Gail, Tex. 963,441.

SUCTION VALVE FOR GAS ENGINES. Herman F. Mueller, Minneapolis, Minn. 963,463.

PIPE CONNECTION. George R. Bard, Chicago, Ill., assignor by mesne assignments, to the Waterworks Specialty Company, Chicago, Ill., a Corporation of South Dakota. 963,498.

PUMP. Fairleigh S. Dickinson, Rutherford, N. J. 963,528.

LUBRICATOR. Otto Hajek, Vienna, Austria-Hungary, assignor to the firm of Alex. Friedmann, Vienna, Austria-Hungary. 963,547.

VALVE OF INTERNAL COMBUSTION ENGINES. D'Arcy Victor Thomas Andrews Locke, Southbourne, England, assignor to Edwin James Castiglione, Carlisle, England, and Francis Cheney Bolton, Wandales, Eetheral, England. 963,600.

STEAM METER. Frank E. Pendleton, Montclair, N. J., assignor to Gamaliel C. St. John, Montclair, N. J. 963,643.

PACKING FOR PISTON RODS. John S. Pendleton, Lincoln, Ill. 963,645.

ROTARY PUMP. Frank J. Curtis, Ben Avon, Penn. 963,690.

STEAM SUPERHEATER. Ernest H. Foster, New York, N. Y. 963,732.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC BOND. Thomas J. Cope, Philadelphia, Penn. 963,035.

SYSTEM OF ELECTRIC METERING. Matthew O. Troy, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 963,096.

MOTOR-CONTROL SYSTEM. Henry B. Emerson, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 963,127.

STARTING DEVICE FOR ELECTRIC MOTORS. Harry H. Lee and Arnold E. Button, Schenectady, N. Y., assignors to General Electric Company, a Corporation of New York. 963,160.

LIQUID RHEOSTAT. Frederick Mackintosh, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 963,163.

LIGHTNING ARRESTER. Bertram H. Mann, Webster Groves, Mo. 963,233.

STORAGE-BATTERY SEPARATOR. Ervin M. Fitz, Columbus, Ohio. 963,284.

SPARKING DEVICE. Charles H. Stonebridge, New York, N. Y. 963,334.

ELECTRICAL SYSTEM OF DISTRIBUTION. William A. Turbayne, Lancaster, N. Y., assignor to oGuld Storage Battery Company, a Corporation of New York. 963,339.

RELAY. Isidor Kitsee, Philadelphia, Penn. 963,374.

SLACK SPOOL FOR ELECTRIC WIRING. Henry Van Altena, New Brunswick, N. J. 963,966.

ARRANGEMENT FOR STARTING AND REGULATING THE SPEED OF ALTERNATING-CURRENT MOTORS. Engelbert Arnold, Karlsruhe, Germany, and Jens Lassen LaCour, Edinburgh, Scotland. 963,678.

POWER PLANT TOOLS

LIFTING JACK. John H. Burkholder, Ashland, Ohio, assignor to the Elite Manufacturing Company, Ashland, Ohio, a Corporation. 963,206.

WRENCH. Benedict Goodman, Wallace, Kan. 963,287.

WRENCH. Robert Earl Harshell, Georges Station, Penn. 963,550.

WRENCH. William E. Elliott, Sault Ste. Marie, Mich. 963,700.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS.

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

BUSINESS ITEMS

George H. Kendrick, formerly with the H. W. Johns-Manville Company, has been appointed New England sales manager of the Crandall Packing Company, with offices at 19 High street, Boston, Mass.

The Peerless Motor Car Company, Cleveland, Ohio, and the Buick Motor Car Company, Flint, Mich., have recently purchased Swartwout cast-iron exhaust heads and centrifugal separators from the Ohio Blower Company, Cleveland, Ohio.

A. K. Waycott, formerly with the Crandall Packing Company, has been elected president of the Argo Supply Company, with offices at 136 Liberty street, New York. This company handles engineer and mill supplies, and steam-engineering specialties.

The Jefferson Union Company, of Lexington, Mass., has recently prepared a catalog on its style F male and female union. This is ready for distribution and will be sent on request. It is also preparing a catalog on its new swing union, which is almost a universal union. This will be ready for distribution in the very near future.

The Gleason Works, Rochester, N. Y., has awarded contract to the C. & G. Cooper Company, Mt. Vernon, Ohio, for a 375-horsepower simple Corliss engine to be installed in the new power house. This engine will drive a direct-current 250-kilowatt generator and furnish power for the Gleason Works, including the extensive additions now being made.

The Precision Instrument Company has been incorporated at Detroit, Mich., to manufacture CO_2 recorders and other instruments under the patents of Alexander Wright & Co., of England. The officers of the company are president, Andrew H. Green, Jr.; vice-president, V. F. Dewey; secretary and treasurer, Abram T. Baldwin; general manager, M. Woolsey Campau.

At the recent convention of the International Association for the Prevention of Smoke, at Minneapolis, Minn., R. S. Riley, president of the American Ship Windlass Company, Providence, R. I., read an instructive paper on "Taylor Stokers and Steam Boiler Efficiency," in which he showed by numerous illustrations the principles of construction and operation in their relation to increased efficiency.

The Mine and Smelter Supply Company, who maintain large offices and warehouses at Denver, El Paso and Mexico City, with executive offices at 42 Broadway, New York City, announce it has taken over the agency for the entire western territory of the United States of the line of pumps made by the Epping-Carpenter Company, Pittsburg, and the line of compressors made by the Bury Compressor Company, Erie, Penn.

The De La Vergne Machine Company has opened a new office at 380 Bourse building, Philadelphia, Penn., and has appointed Hy. J. Engel to take charge of it. He will represent them in Pennsylvania and some of the surrounding States. Mr. Engel has been connected with the company for a good many years, is a mechanical engineer and has a long and varied experience in the refrigerating line. The company has also opened an office at 1124 Chandler building, Atlanta, Ga., which will be in charge of Wm. Hargreaves, who will represent it in the southern district, which territory he is very familiar with on account of having acted as its agent there up to two years ago when it closed its Atlanta agency.

The Atlas Water Tube Boiler Company has opened general offices in the Franklin Bank building, Philadelphia, Penn. The officers of the company are B. J. Morrison, president; H. A. Scattergood, secretary and treasurer; George R. Harvey, general manager. This company has taken over all rights and patents of the Atlas water-tube boiler, formerly manufactured by the Atlas Engine Works, of Indianapolis, Ind. The new company will devote its energies exclusively to the manufacture of this boiler, of which there are now installed in this country something over 250,000 horsepower. It will be recalled that the Government power plant, at Washington, D. C., just recently turned over, is equipped with 16 (550 horsepower each) Atlas boilers, which is said to have been the largest single order foi boilers ever placed by the Government.

The International Acheson Graphite Company, of Niagara Falls, N. Y., has arranged to erect a very substantial and large addition to its works on the lands of the Niagara Falls Power Company in that city. The addition will be in the nature of a furnace room, which will afford capacity for 10 or 12 new furnaces. The building will be of brick and steel construction and erected to the west of its present works, where there is ample room for still further enlargement. The additional furnaces will make possible a very large increase in the output of graphite, and is indicative of the constantly increasing recognition of the value of the products of the company. These products are electrodes for electrochemical, metallurgical and electrical purposes, and powdered graphite for dry-cell filler, paint pigment, powder glazing, lead pencils, firearms lubrication, general lubrication, electrotypers' molding and polishing leads, etc.

The Berlin Construction Company has received the contract for rebuilding the fan shop of the Green Fuel Economizer Company, at Matteawan, N. Y., which was almost totally destroyed by fire some weeks ago. It is expected to have the new building erected and complete, ready for occupancy, about the middle of August. The new shop will have about 25 per cent. more floor space than the old shop, including a test room fully equipped with chambers, gages, etc., suitable for making exhaustive tests on the capacity, pressure and efficiency of the improved fans built by this concern for heating and ventilating, mechanical draft and other services. The new building will be of steel construction throughout and will be heated by the hot-blast system, with frequent outlets near the floor to secure uniform distribution of the heat. This system has been applied by the Green Fuel Economizer Company with great success in several buildings that were peculiarly difficult to heat, such, for instance, as the new foundry of the Waterbury Castings Company, where the wall surface is almost entirely of glass.

NEW EQUIPMENT

The city of Decatur, Neb., is contemplating installing waterworks.

The city of Yukon, Okla., voted $30,000 bonds for waterworks.

Hotel Rudolf, Atlantic City, N. J., will install two more boilers.

The city of Belleville, Ont., will equip a new electric-light station.

The Granite (Colo.) Tunnel Company will build an electric power plant.

The Cudahy Packing Company is erecting a new building at Portland, Ore.

The Snyder (Tex.) Ice, Light and Power Company will build a new power house.

The town of Munday, Tex., has voted the issuance of $10,000 bonds for waterworks.

The Peoples Power Company, Moline, Ill., is erecting a $500,000 gas- and electric-light plant.

The Wheeling (W. Va.) Traction Company will shortly commence work on a new power house.

Hartselle, Ala., will vote on issuing $28,000 bonds for an electric-light plant and waterworks.

The city council, of Madisonville, Ky., voted in favor of installing a waterworks system.

The city of Mansfield, Okla., will vote on the question of issuing $20,000 bonds for waterworks.

The Providence (R. I.) Dyeing, Bleaching and Calendering Company will enlarge its boiler house.

The Dubuque (Iowa) Packing Company will build a new plant to be operated by electric power.

The city of Oceanside, Cal., will vote on the issuance of $15,000 for improvements to its waterworks.

The Northern Illinois Electric Railyway Company will erect a large power house at Lee Center, Ill.

The citizens of Belgrade, Neb., voted to issue $18,000 bonds for an electric-light plant and waterworks.

Plans for a waterworks plant at Conway, Ark., have been accepted and bids will be advertised for soon.

Lake Arthur, La., will install a small system of waterworks. Address J. B. Streator, chief, fire department.

Two hundred thousand dollars will be spent on improvements in the water, light and sewer system at Altus, Okla.

The Alabama Canning Company, Mobile, Ala., is said to be contemplating the installation of an electric-light plant.

The Kooskia (Idaho) Land and Power Company will build a hydroelectric plant of about 5000 kilowatts capacity.

The citizens of Roseville, Cal., voted to issue $90,000 bonds for electric-light and power plant and sewer system.

Bids will be received August 4, at the office of the city clerk, Gadsden, Ala., for building new waterworks plant, for which $60,000 bonds were voted recently.

NEW CATALOGS

The Ohio Brass Company, Mansfield, Ohio. Pamphlet. Ohio radiator valves Illustrated, 8 pages, 5x6 inches.

The Canton-Hughes Company, Canton, O. Catalog No. 25. Steam pumps and hydraulic machinery. Illustrated, 120 pages, 5½x7 inches.

L. J. Wing Manufacturing Company, 90 West street, New York. Bulletin No. 7. Typhoon turbine blower. Illustrated, 20 pages, 6x9 inches.

Pennsylvania Flexible Metallic Tubing Company, 1305 Arch street, Philadelphia, Penn. Booklet. Interlocking metal hose. Illustrated, 22 pages, 3½x6 inches.

C. W. Hunt Company, West New Brighton, N. Y. General Catalog No. 102. Coal-handling and hoisting machinery, conveyors, electric and steam hoists, etc. Illustrated, 112 pages, 4x6½ inches.

Sprague Electric Company, 527 West Thirty-fourth street, New York. Bulletin No. 600. Single and polyphase induction motors. Illustrated, 40 pages, 8x10½ inches. Form No. 517. Flexible steel-armored hose. Illustrated, 10 pages, 3½x6 inches. Form No. 322-A. Electric fans. Illustrated, 20 pages, 3½x6 inches.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

How often do you take advantage of the opportunities offered in the advertising sections of Power of getting into close and complete touch with fresh and vigorous ideas, with the newest developments in power plant equipment, by sending for the catalogs, booklets, souvenirs and samples which the advertisers invite you to have at their expense?

There can be no doubt but that the great bulk of engineers and other power plant operators appreciate the great advantage of buying their equipment through good advertising mediums. Returns from the advertising pages of Power prove that conclusively.

But it is not so clear that there is a full realization of *all* the opportunities of keeping abreast of the times through the free matter offered by advertisers.

There are people who are afraid of anything that is free, who argue that the only things worth having—especially in business—are those paid for in good hard cash, and that everything else had best be avoided. Hence, they don't send for the free catalogs, etc.

Then there are also men whose very life in business depends upon their up-to-dateness in their special field, who fail to take advantage of the opportunities offered in the invitations to get "full details," simply because they do not realize the great advance made in printed salesmanship during recent years.

Advertisers pay good money for space in Power in order to put the *main* facts about their product in your hands, because they have confidence that their product is something that you *really* need.

That same confidence of genuine merit is also the thing that prompts the advertiser's desire to go to the additional expense of putting the *full and complete* facts in your possession by means of free catalogs, booklets, souvenirs and samples.

And, furthermore, it is your best evidence that the something which he wants to send you, without charge, is *worth having*.

Catalogs and booklets on power plant equipment have undergone much the same evolution, in recent years, as has advertising copy on the same class of products.

Examine the ads. in any number of Power. Instead of mere publicity notices—names, addresses and a picture—the Power ad. sections contain page after page of meaty information, just the matter that the Power plant man *needs* for wise buying.

The same thing has come to be true of the printed salesmanship that follows the ad.—if you will let it. Not price lists. Not bald claims with no evidence. But clear, full statements of *ideas, facts, reason why*.

Good money and good brains go into them. They offer—with the advertising copy—the best method of getting the full story of what is being done by progressive men in your line.

That they are free is no reason why you should be skeptical of them. It is simply the strongest proof of the *real* merit of the matter which they contain.

Free souvenirs, engineering handbooks, samples and the like, have also undergone an evolution. They, too, now have their story to tell.

The leading features of many a product can often be strikingly displayed in some handy little article.

Again, if the advertiser has the confidence in the *real* worth of his product to go to the still further expense of adopting this pleasant method of presenting his case, why not take him up? You have everything to gain and nothing to lose!

Nor does the advertiser lose. He realizes that the circulation of his advertising matter among power plant men means a practically "wasteless" circulation.

The whole matter comes right down to this point: You *must* take *full* advantage of progressive printed salesmanship if you are to win your full measure of business success.

And the best way to do it is to read the Power ads. carefully and then get the *full* story from the advertisers who want to give it to you.

Power
and The Engineer

Vol. 32 NEW YORK, AUGUST 2, 1910 No. 31

THERE is a right and a wrong way to do everything. This applies even to reading your POWER. In fact, there are numerous *wrong* ways in which this can be done.

What is one man's meat is another man's poison. In a like manner that which is the best way for one man to read his technical magazine may be altogether wrong for another.

No hard and fast rules can be laid down for all to follow.

One general suggestion can be offered, however: *Get* what you go after.

A technical periodical which is rightly balanced contains in every issue something for the men of each level of that branch of engineering to which the paper is devoted.

Now, most men have not the time to read from 40 to 45 pages each week and *get* all that they contain.

By "get" we mean read so carefully that all of the facts will become fixed in his memory.

It thus becomes necessary to make a careful selection of the articles printed in each number. The chief engineer of some big electric railway or lighting system would find it unprofitable to read articles such as "Device for Inserting a Valve Stem" and "A Homemade Planimeter" to the neglect of some of more importance to him. In the first place, he may never need to use the information they contain; in the second, he can use the time to better advantage by devoting it to articles dealing with subjects more closely connected with his position.

By the same token, it is unprofitable for the man in charge of a 100-horsepower plant to pore over articles dealing with the construction or design of some big central station.

By this we do not mean that he should not read any but the articles which vitally concern him. On the contrary, the more he reads the better, providing, of course, that he does not do the extra reading at the expense of more important things.

Read your scientific paper in a *scientific* manner. Its chief purpose is to help and instruct.

Get the help and instruction first. By so doing you avoid the possibility of losing out through lack of time to read what you should read for your own improvement and profit.

There are some subjects which are of importance to you. When an article on one of them appears, go after *all* that it contains. Don't glance through it and then turn the page, with the remark, "That's a good article. I'll read that carefully,—tomorrow."

Tomorrow never comes, no matter how good your intentions are.

If you are wise, you will "anchor" right at that article and "dredge" for all that there is in it. You will check the facts as far as you are able.

By so doing you will not only establish their reliability, but will also fix them more firmly in your mind.

After you have "*got*" the articles designed for you, if you have any time to spare, pass on to the pages not so closely related to your particular work.

But *get yours first*, and get it *right*.

Economical Fire Room Methods

By F. R. Low

In the plant under discussion it is realized that the boiler room is the place to save fuel. Return tubular boilers are hand fired by intelligent men at regular intervals with uniform quantities of coal. Fans furnish the draft and an economizer reduces the temperature of the escaping gases to 200 degrees and raises the temperature of the hotwell water from 105 to 185 degrees. Careful firing gives a high percentage of CO_2, and but little excess air. Some remarkable results are obtained in the plant.

I had the privilege and pleasure of spending a day recently with George H. Diman, of Lawrence, Mass. We had made the rounds of the power plants of the American Woolen Company, which are run under Mr. Diman's direction, and wound up at the Wood mill, which is Mr. Diman's particular favorite, being his own creation, erected under his direction and from his own designs. "Come down to the boiler room," he said, "I want to show you my Hibernian Automatic Stokers."

"The Hibernian stoker," I said, "Never heard of it."

"You can't beat them," he said, as he pointed to a line of firemen operating the boilers by hand, and drew up some chairs where we could watch operations.

The boiler house is light and roomy, behind the boilers as well as in front, and spick-and-span all over. It contains forty-two 72-inch horizontal return-tubular boilers with 20-foot tubes, and everything goes on with the regularity of clock work.

"How do you keep it so clean," I asked.

"By never letting it get dirty," was the laconic reply.

The captain introduced me to the head fireman and told him to bring him "some of those records." He produced a pack of recording-temperature gage charts and Mr. Diman passed them to me with the question, "Did you ever see a boiler plant with a lower stack temperature than that?"

I looked the records over. Fig. 1 is an average sample. The inner diagram shows the variation of the temperature at which the gases go to the stack.

It hangs close to 200 degrees.

The larger diagram shows the temperature of the gases as they leave the boilers—never up to 500 degrees, and part of the time less than 400.

Economizers reduce them to the low final temperature, and were raising the feed-water temperature on the day of my visit from that of the hotwell, about 105, to 185 degrees. Four 15-foot Sturtevant fans on the cool side of the economizer supplied the draft.

I was obliged to own that I had never seen gases sent to the stack so cool before, and asked, "How much excess air is there in them?"

"Look at that gas analysis I gave you," advised the captain.

I pulled it out, and it was as follows:

Lawrence, Mass, April 29, 1910.

G. H. Diman, Esq., Consulting Engineer,

WOOD WORSTED MILLS.

Dear Sir:

Record of gas analyses made April 29, 1910. Samples taken from uptake just back of the front doors:

Time.	Carbon Dioxide.	Oxygen.	Carbon Monoxide.
Boiler No. 6.			
8:45 a.m......	14.1%	3.5%	0.0%
8:55 a.m......	14.4%	4.6%	0.0%
Boiler No. 9.			
9:05 a m......	15.4%	0.9%	0.5%
Boiler No. 15.			
9:20 a.m......	15.0%	**3.6%**	0.0%
9:30 a.m......	12.8%	**6.7%**	0.0%
Boiler No. 18.			
9:40 a.m......	12.7%	6.4%	0.0%
Average....	14.1%	4.5%	0.1%

WOOD WORSTED MILLS.

Dear Sir:

Record of gas analyses made May 13, 1910. Samples from boilers No. 33 and No. 35 were taken from the center of the roof of boilers. Other samples were taken from the uptake, as previously.

Samples from boiler No. 33 show what value the flue-gas analyses may have in locating trouble with the fire.

It was found by the man in charge after the 9:30 a.m. sample was taken that the fire had burned a hole in the side. This was remedied, and the 9:40 sample shows a much better result:

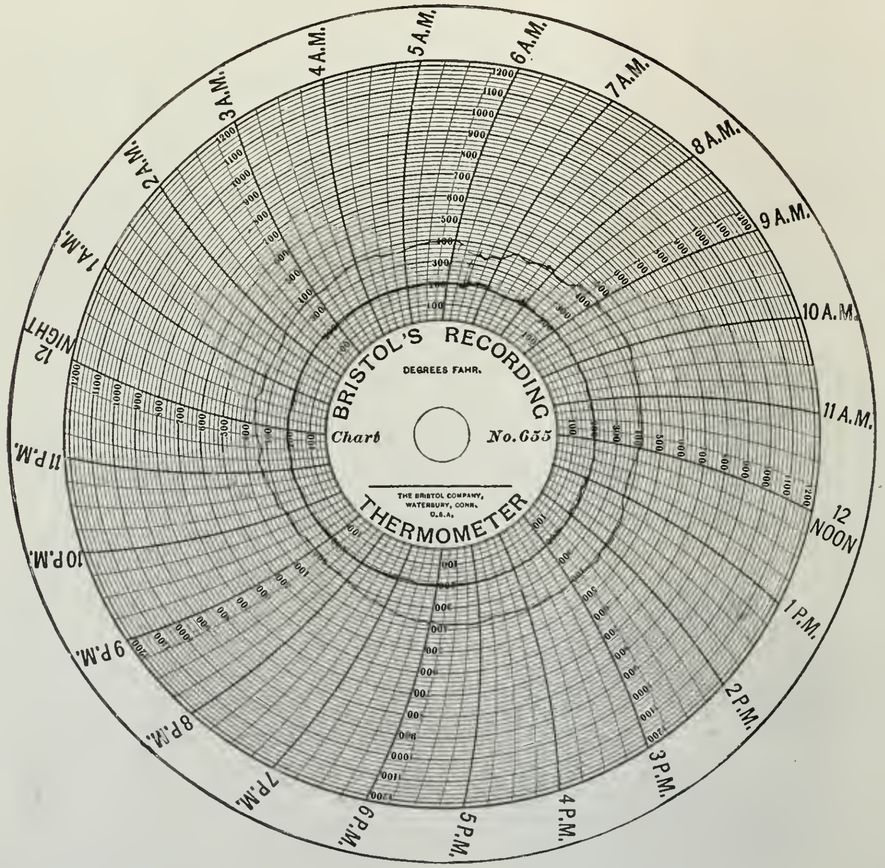


Fig. 1. Typical Record of Uptake and Stack Temperatures

Time.	Carbon Dioxide.	Oxygen.	Carbon Monoxide.
Boiler No. 33.			
8:45 a.m.	13.1%	4.8%	0.0%
8:55 a.m.	11.8%	6.6%	0.0%
9:05 a.m.	11.6%	7.1%	0.0%
9:15 a.m.	10.8%	7.7%	0.0%
9:30 a.m.	8.4%	11.5%	0.0%
9:40 a.m.	13.7%	4.8%	0.3%
Boiler No. 35.			
9:50 a.m.	13.5%	6.0%	0.0%

Time.	Carbon Dioxide.	Oxygen.	Carbon Monoxide.
Boiler No. 28.			
10:05 a.m.	13.3%	5.2%	0.1%
Boiler No. 29.			
10:15 a.m.	14.0%	5.3%	0.2%
Boiler No. 25.			
10:25 a.m.	11.8%	7.5%	0.1%
10:35 a.m.	10.4%	9.7%	0.0%
Average	12.0%	6.9%	0.1%

Lawrence, Mass., May 13, 1910.
G. H. Diman, Esq., Consulting Engineer,
Yours truly,
P. F. Ripling,
(P. F. R.) Chemist.

"We have our own chemist come in and make these analyses every week. He comes unexpectedly, picks his own boilers and fires, and his reports will all run about like that. Not much free air when you are getting that kind of analysis, is there?"

There certainly could not be, and really such excess air as there was represented a considerably less loss at this low temperature than at the usual temperature of around 500 degrees. If, as Mr. Diman pointed out, he cooled the gases down to the temperature at which the air came in, it would not make any difference how much excess air there was, for none of it would go away hotter than it came, and so would not carry out any heat units.

The boilers are hand fired on ordinary Foster flat shaking grates. "How do you manage your fires," I asked, "to keep up any such percentage of CO_2 as that?"

"Watch them," he said, "get out your watch and count the scoopfuls."

The plant is divided, so far as organization and operation are concerned, into four groups of ten boilers each. Each group has twenty fire doors and four stokers, giving five fire doors to a man. The man at the near end of the group would fire the first, third and fifth doors. The second man would take it up and fire the seventh and ninth; then the next man, and so on. Each man put on just nine scoopfuls of coal, no more and no less. When the first man had finished coaling door No. 3 he took his hoe and trimmed Nos. 2 and 4, then he went and sat down. Each fireman has a stout, comfortable chair, and it is for his especial use. Fireman No. 2 was soon sitting beside him, having fired his two doors and trimmed his three. After an interval of eight or ten minutes from the time he fired No. 1 door, fireman No. 1 sees fireman No. 4 finish his turn, gets up, coals furnaces Nos. 2 and 4, trims Nos. 1, 3 and 5, and sits down again. Fireman No. 2 commences to throw coal into No. 6 as soon as fireman No. 1 closes the door of No. 4, and as soon as he has fired 6, 8 and 10, and trimmed 7 and 9, he sits down by his partner.

And so it goes on like clockwork. Nobody has to be told when it is his turn, there is no bossing, nagging, nor bawling, the men work easily and systematically, there are never more than two doors open in the group at a time, that into which the man is shoveling and that of the furnace which is being trimmed. The firing of alternate doors at considerable intervals insures that one side of each furnace shall be full of seething, white-hot, well coked coal to burn the volatiles from the fresh fuel fired to the other side, and the fire is carried some 14 inches in thickness all over the grates, giving no chance for air to get in without losing its oxygen to the white-hot carbon. If the load is light and coal is not being burned so fast but that the fires thicken up under this treatment above 16 inches or so, the number of scoopfuls per firing is reduced; but they can get around fast enough, so that nine scoopfuls at a time will supply all the coal the boilers will use at the greatest rate that they are ever driven.

"What about slicing?" I asked.

"Only when the whistle blows," was the reply.

"And the whistle blows——?"

"When the pressure falls 15 pounds," said the captain, smiling as he saw my puzzled look.

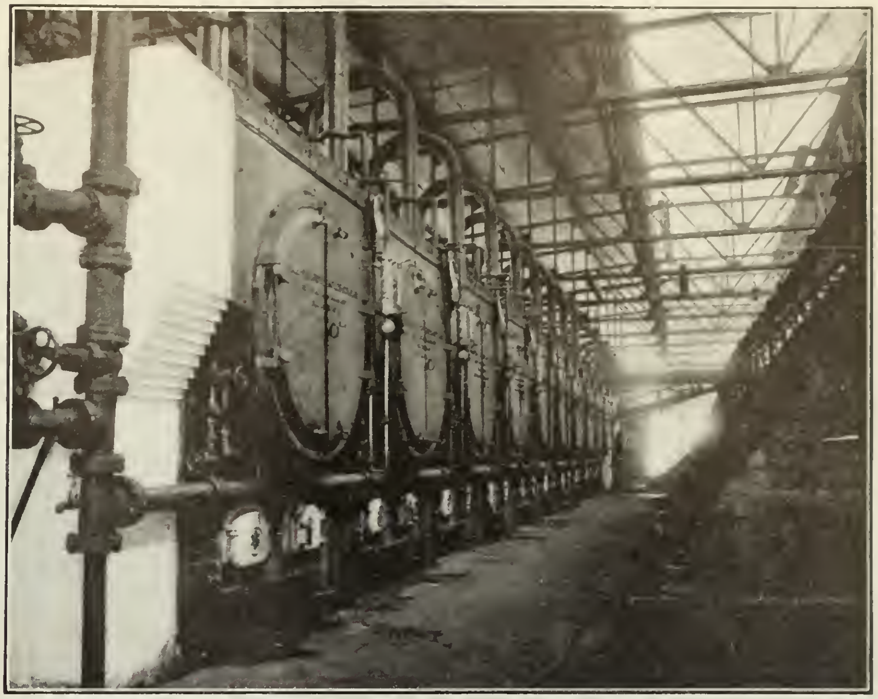


Fig. 2. One Side of the Wood Mill Boiler Room

"There is another fallacy," he explained, "that we have all been trying to live up to. We've all been trying and educating our men to try to keep a perfectly even boiler pressure. We put on a damper regulator, and it held the steam pressure within a pound, and the recording-pressure gage made a straight line, and we thought it was great. Well how did it do it? When the pressure got a little bit too high the damper went shut. You cut off your air supply and your furnaces went to making carbon monoxide; 4450 B.t.u. to the pound of carbon, instead of 14,500, sending off the same kind of gas that they use to run gas engines up the stack, instead of burning it in the furnace. Figure out some time how little you lose by dropping your steam pressure from 160 to 145, and how much you lose by dropping your percentage of CO_2 from 14 down to 8 by imperfect combustion."

"Now instead of monkeying with my dampers to try to keep an absolutely straight steam pressure, I try to keep the air supply so that it will burn the coal. The steam pressure can vary the speed of the fans so that the draft over the fire runs from about 0.3 of an inch when the boiler pressure is highest, to about 0.6 when the pressure is down.

FIG. 3. THE "HIBERNIAN AUTOMATIC STOKERS" AT THE WOOD WORSTED MILL

This lowering of the pressure is not like shutting a damper off tight. Enough air comes through at the lowest pressure to burn the carbon to CO_2 and there cannot so much get through that thick fire, even at the highest speed of the fans, as to give an excess."

"How do you keep the fans from going over or under that speed?" I asked.

"We have stops on the valves that control them, and when the fires are in goo condition that amount of come-and-go wi keep the pressure fairly regular."

"Suppose you made so much stea

FIG. 4. THE WOOD WORSTED MILL STOKERS OFF DUTY

with 0.3 of an inch pressure that the safeties blow," I asked.

"In that case we should lay off boilers. And say!" he continued, "there goes another fallacy. We used to think that the more boilers we had and the slower we ran them the better off we were. Lots of heating surface to soak up the heat. The consequence was that you couldn't run your fires hot—they simply mulled and stewed off a lot of gas that went off through the top tubes, and you were paying interest and upkeep on a lot of boilers that were not earning their salt. Now we run the furnaces so as to get all there is out of the coal, get the gas to the boiler good and hot, and then take out what the boiler does not get with an economizer."

"And what do you do when the pressure runs down?"

"We keep enough boilers going to make all the steam we want with less than 0.6 of an inch draft ordinarily. If there should come a sudden temporary pull, it is better to let them draw the steam down a little than to force the fans and blow a lot of excess air through the fires. When the fires have been running for some time they will cake over so badly that 0.6 of an inch draft will not force air enough through them. Then the pressure falls off, and when it gets 15 pounds below normal the whistle blows automatically and everybody goes to slicing."

Reverting to his remarks about slow combustion, I told him that one of our editors had just worked out an easy rule for boiler horsepower. He got it so that it would fit the ratings of a well known builder of horizontal return tubulars nicely. The next time the mechanical engineer of the company was in he showed it to him, and he said that it was all right for the old catalog, but "we rate our boilers on 10 square feet now."

"Well, you can see what we get out of ours," Mr. Diman said, leading me to a framed blueprint report of tests hanging against the brick wall of one of the economizers. I begged the copy of it which is reproduced here.

"The boilers have a little over 2300 square feet of heating surface apiece, and we bought them for 200 horsepower. You can see that in one test (that of October 1, in line 54) we got 316 horsepower out of one of them, and that in none of the eighteen tests did we get less than 200."

"When you burn your coal as completely as those flue-gas analyses show, you are bound to get your heat out of it, aren't you?" he continued. "Well, if you get your heat developed in the furnace there are only three places it can go to—into the steam, up the stack, or out into the room. You don't find much radiation here, do you? A man can eat his lunch comfortably on top of those boilers. There can't be much heat going up the stack with the gases at 200 degrees and that excess of air, can there? How are you going to beat it?"

It did look hard to beat. The stack temperatures were lower on the day that I was there than the values given in the table, but I saw the records being made, in fact Mr. Diman had new charts put on, and the record of the balance of the day's run made so that I could compare directly the flue temperatures with the temperatures of the gases leaving the economizers for the same period of time.

Geo H Diman

"There's another fallacy that they're going to get over some time," said the captain, "and that's building great, tall chimneys to pump off the flue gases by the most wasteful method of pumping that you could conceive. What runs a chimney? It is the heat that you put into it, isn't it? If the gas inside of the chimney were no hotter than the air you wouldn't get any draft, would you? And to get draft enough to burn coal you have got to have a good high chimney and keep it full of good hot gas. Instead of putting the flue gases into the chimney at 450 or 500 degrees, I am putting them in at 200 and am heating the water 80 degrees with them. That means over 7 per cent. saving, doesn't it? Well, don't you suppose that with 7 per cent. of the coal that I turn here I can a good deal more than run those fans? And having them I can have any amount of draft that I want any time that I want it, and am entirely independent of wind, weather, barometer, or anything else, so long as I have steam pressure?"

Mr. Diman was one of the first to realize that the boiler room is the place to save fuel, and this is particularly true in a plant in which, like those of which he is in charge, the demand for exhaust steam is so great as to discourage any overreaching for economy on the engine-room end. The thing that struck me particularly about it was the quiet, orderly effectiveness of it all. The men fired not as though they were there to dig a hole in the coal pile, but as though they were intelligent units in an important industrial system, and as though they realized that their present occupation was but the initial step to a creditable and remunerative career. Mr. Diman knows the characteristics and the capabilities of his men;

EVAPORATIVE TESTS MADE AT WOOD WORSTED MILLS, LAWRENCE, MASS.

By Arthur D. Little, Chemical Expert and Engineer, Boston, Mass. Coal Department, E. G. Bailey, Chief.

Average Pressures:													
8. Steam, lb. per sq.in.	153	153	151	151	152	153	152	151	152	153	151	150	153
9. Draft leaving boiler, in. of water	0.99	1.05	1.74	1.14	1.22	1.18	1.25	1.15	1.31	1.36	1.27	1.21	2.1[illegible]
10. Draft over fire, in. of water	0.58	0.50	0.65	0.57	0.69	0.66	0.60	0.52	0.67	0.76	0.73	0.66	0.86
Average Temperatures:													
11. Fire room, deg. F.	87	85	67	76	76	77	82	62	73	79	64	65	63
12. Products of combustion leaving boiler, deg. F.	496	479	478	478	478	473	489	490	489	477	470	491	520
13. Products of combustion entering economizer, deg. F.	443	442	437	440	440	438	440	435	436	435	442	448	410
14. Products of combustion leaving economizer, deg. F.	236	237	238	242	242	239	243	239	243	240	241	244	240
15. Feed water entering economizer, deg. F.	110	112	113	112	112	113	108	114	109	109	107	109	106
16. Feed water entering boiler, deg. F.	181.6	176	178	181	179	180	180	181	178	178	173	177	178
Fuel and Furnace:													
18. Weight of wood for kindling, lb.	345	351	355	538	561	369	344	332	317	493	407	387	480
19. Total weight of coal fired, lb.	9,600	10,200	11,200	11,200	11,200	10,800	11,200	11,600	11,600	11,800	10,400	14,000	11,000
20. Total weight of coal and equivalent wood fired during test, lb.	9,738	10,320	11,342	11,416	11,424	10,947	11,337	11,733	11,727	11,997	10,563	14,155	14,192
21. Refuse from grate and cleaning, lb.	536	416	546	370	648	626	468	587	568	1,107	540	770	533
22. Refuse from ashpit, lb.	566	608	656	674	776	599	578	422	378	530	565	454	500
23. Total refuse, lb.	1,102	1,024	1,202	1,044	1,424	1,225	1,046	1,009	946	1,637	1,105	1,224	1,033
24. Weight of clinker from total coal, lb.	231	219	231	130	309	263	321	351	278	563	104	240	307
25. Per cent. total refuse to total coal	11.5	10.0	10.9	9.3	12.7	11.3	9.3	8.7	8.2	13.4	10.6	8.7	7.4
26. Per cent. clinker to total coal	2.4	2.2	2.1	1.2	2.8	2.4	2.9	3.0	2.4	4.8	1.0	1.7	2.2
27. Per cent. clinker to total refuse	21.0	21.4	19.2	12.4	21.6	21.4	30.8	34.8	29.4	34.4	9.4	19.6	29.8
28. Per hour, weight of coal fired, lb.	800	827	933	933	947	913	927	1,016	967	983	873	1,183	1,167
29. Per hour, weight of coal fired per sq ft. grate, lb.	20.5	21.2	23.9	23.9	24.3	23.4	23.8	26.1	24.8	25.2	22.4	30.3	29.9
30. Per hr., weight of coal fired per sq.ft. heat'g sur., lb.	0.348	0.359	0.405	0.405	0.411	0.397	0.402	0.441	0.420	0.426	0.379	0.514	0.506
Analyses (Proximate Analysis):													
31. Moisture	0.91	1.00	1.15	1.24	1.54	1.26	1.24	1.33	1.17	1.80	1.58	3.26	1.02
32. Volatile	15.51	15.01	16.82	16.77	14.86	15.64	19.42	19.70	17.95	30.85	18.18	22.16	20.87
33. Fixed carbon	73.13	72.45	72.06	72.16	71.43	71.72	72.18	70.42	73.82	54.98	69.64	66.53	71.10
34. Ash	10.45	11.54	9.97	9.83	12.17	11.38	7.16	8.55	7.06	12.37	10.60	8.05	7.01
Ultimate Analysis:													
35. Carbon	79.63	78.37	79.36	79.30	76.80	77.81	81.94	80.12	82.54	71.15	78.29	77.08	82.18
36. Hydrogen	4.13	4.08	4.16	4.17	4.05	4.07	4.45	4.36	4.23	5.04	4.29	4.26	4.40
37. Oxygen	2.68	2.73	3.09	3.17	3.87	3.65	3.85	3.87	3.05	7.10	3.84	7.75	4.11
38. Nitrogen	1.47	1.45	1.51	1.51	1.44	1.46	1.39	1.36	1.32	1.50	1.55	1.49	1.45
39. Sulphur	1.64	1.83	1.91	2.02	1.67	1.63	1.21	1.74	1.80	2.82	1.43	1.37	0.85
40. British thermal units	14,046	13,784	13,921	13,966	13,478	13,620	14,433	14,142	14,426	13,038	13,747	12,937	14,402
Refuse:													
41. Combustible in refuse from ashpit	23.8	20.7	29.7	39.7	22.9	18.1	30.0	38.7	32.9	24.5	20.7	28.3	27.7
42. Combustible in refuse from grate	7.3	10.7	7.5	18.6	9.9	9.4	3.6	10.7	4.2	8.9	12.5	14.9	6.0
43. Combustible in total refuse	15.8	16.6	19.6	35.4	16.9	13.6	18.2	22.4	15.7	13.9	16.7	19.8	16.4
Water:													
44. Per cent. moisture in steam											0.37	0.53	0.68
45. Factor of correction for quality											0.9970	0.9958	0.9946
46. Gallons of water fed to boiler by meter	13,185	13,400	14,350	14,815	14,230	13,815	14,640	15,260	15,610	13,635	13,875	17,070	17,235
47. Gallons of water fed to boiler by meter corrected, factor 0.864	11,392	11,578	12,398	12,800	12,295	11,936	12,649	13,185	13,487	11,781	11,987	14,748	14,890
48. Weight per gallon of water, lb.	8.085	8.107	8.100	8.092	8.098	8.095	8.095	8.092	8.100	8.100	8.115	8.104	8.100
49. Equivalent weight of water evaporated, lb.	92,104	93,663	100,424	103,578	99,563	96,621	102,392	106,692	109,243	95,427	97,273	119,517	120,615
50. Factor of evaporation for boiler	1.0809	1.0867	1.0842	1.0811	1.0833	1.0824	1.0823	1.0811	1.0843	1.0844	1.0895	1.0852	1.0825
51. Equivalent water evaporated from and at 212° F., lb.	99,555	101,783	108,880	111,716	107,857	104,583	110,818	115,344	118,453	103,481	105,660	129,155	129,681
52. Equival't water evap. from and at 212° F. per hr., lb.	8,296	8,254	9,073	9,310	9,118	8,840	9,174	10,100	9,871	8,623	8,864	10,918	10,807
53. Equivalent water evaporated from and at 212° F. per hr. per sq.ft. heating surface, lb.	3.60	3.58	3.94	4.04	3.96	3.84	3.98	4.38	4.28	3.74	3.85	4.74	4.69
54. Horsepower developed by boiler	243	239	263	270	264	256	266	293	286	250	257	316	313
56. H.P. developed above rated per cent.	22	20	31	35	32	28	33	46	43	25	29	58	57
57. Sq.ft. heating surface per H.P. developed	9.48	9.64	8.76	8.53	8.73	9.00	8.66	8.76	8.06	9.22	8.97	7.29	7.36
Evaporation:													
58. Pounds of water apparently evaporated per lb. coal.	9.35	9.08	8.85	9.07	8.72	8.83	9.03	9.09	9.32	7.95	9.21	8.44	8.50
59. Equivalent evap'tion from and at 212° F. per lb. coal	10.22	9.86	9.60	9.79	9.44	9.55	9.78	9.83	10.10	8.63	10.00	9.12	9.14
60. Equivalent evaporation from and at 212° F. per lb. combustible	11.67	11.08	10.89	10.94	11.08	10.91	10.93	10.93	11.14	10.18	11.39	10.36	9.98
61. Efficiency of boiler and furnace, per cent.	70.2	69.1	66.6	67.7	67.6	67.7	65.5	67.1	67.6	63.9	70.3	68.1	61.3
Flue Gas by Volume:													
62. Carbon dioxide, CO_2	9.0	9.8	8.9	9.3	9.0	9.0	7.8	8.3	8.9	8.8	8.6	10.1	9.3
63. Oxygen, O	11.4	10.5	10.9	10.4	11.2	11.1	12.4	12.1	11.4	11.3	11.6	10.2	10.8
64. Carbon monoxide, CO	0.0	0.0	0.1	0.1	0.0	0.0	0.2	0.1	0.1	0.1	0.2	0.1	0.1
65. Nitrogen, N	79.6	79.7	80.1	80.2	79.8	79.9	79.6	79.5	79.6	79.8	79.6	79.6	79.8
66. Air excess	121	102	109	98	116	114	148	140	122	118	126	96	107
Heat Balance:	%	%	%	%	%	%	%	%	%	%	%	%	%
67. Heat used in evaporation, B.t.u. per lb. coal	9,870 70.2	9,522 69.1	9,271 66.6	9,454 61.7	9,117 67.6	9,223 67.7	9,445 65.5	9,493 67.1	9,754 67.6	8,334 63.9	9,658 70.3	8,808 68.1	8,826 61.3
68. Loss due to latent heat of water formed	360 2.6	360 2.6	360 2.6	360 2.6	350 2.6	350 2.6	385 2.6	380 2.7	360 2.5	435 3.3	370 2.7	365 2.8	380 2.6
69. Loss due to products of combustion	1,195 8.5	1,105 8.0	1,195 8.6	1,145 8.2	1,120 8.3	1,120 8.3	1,205 8.4	1,215 8.6	1,225 8.5	1,072 8.2	1,160 8.4	1,171 9.1	1,338 9.3
70. Loss due to air excess	1,260 9.0	985 7.1	1,160 8.3	975 7.0	1,150 8.6	1,020 7.6	1,560 10.8	1,500 10.6	1,330 9.2	1,100 8.5	1,260 9.2	970 7.5	1,260 8.7
71. Loss due to CO and unburned gases			78 0.5	80 0.5			204 1.4	79 0.5	91 0.6	77 0.6	178 1.3	76 0.5	82 0.6
72. Loss due to unburned coal	285 2.0	330 2.4	345 2.5	770 5.5	355 2.6	255 1.8	230 1.6	355 2.5	185 1.3	285 2.2	300 2.2	285 2.2	200 1.4
73. Loss due to radiation, absorption, etc.	1,076 7.7	1,482 10.8	1,512 10.9	1,182 8.5	1,396 10.3	1,652 12.0	1,404 9.7	1,120 7.9	1,481 10.3	1,735 13.3	821 6.0	1,262 9.8	2,316 16.1
74. Total	14,046 100.0	13,784 100.0	13,921 100.0	13,966 100.0	13,478 100.0	13,620 100.0	14,433 100.0	14,142 100.0	14,426 100.0	13,038 100.0	13,747 100.0	12,937 100.0	14,402 100.0

he never hires a man from the outside when he has a man inside who can fill a better job; he has an interest in them as individuals, is anxious to teach them all they will absorb and to help them to profit by what they have learned. I found out that a number of boys who had come to him as firemen and helpers owed their ability to read and write to his teaching, and it is characteristic both of his disposition to advance his own men and of his high appreciation of the importance of the fire room that he has selected the head fireman of the Washington mill to be the chief engineer of the new Ayer mill, an uptodate turbine plant of 5000 kilowatts with all of the modern developments.

Cleanliness in Power Plants

In the Public Service Journal Thomas J. Walsh briefly points out to what extent the cost entering into the production of power may be affected by the proper attention to cleanliness about the power station. His suggestions are well worth careful attention.

The added exertion necessary to keep equipment and building tidy and free from accumulation of dirt, with all its attendant influences on the deterioration of the apparatus, is so slight, while the benefits are so marked, that there can be no question as to its expediency. No one will deny that the replacing of tools and supplies in their proper places, the removal of foreign matter from any part, the gathering up of waste scraps for easy disposal, the tightening of a loose joint or the replacement of a fractured pipe or fitting in the oil piping, require but the minimum of attention at the appropriate time. Few things are more disheartening to an engineer than the wail of a "dying" turbine, or the laying down of an engine under load, especially with the tools scattered about the plant. Then, too, when the cleanliness of engines and boilers is neglected, the increase in coal consumption becomes at once noticeable; and where there is leaky oil piping, the oil bills must increase as oil is thrown about the engine-room floor and into flywheel pits, or into generator windings and commutator with destructive effects, especially when the cleaning is neglected.

No stronger proof could be brought forward as to the efficacy of the attentions as urged here in this article than the consideration of the power plants of Holland. The wonderful economic results obtained from the obsolete equipment in those stations was remarked some time ago by the editor of the *Street Railway Journal,* and together with these economic results was mentioned the remarkable cleanliness of everything about the plant, not only superficially, but of the invisible parts as well.

And in our own country it is only necessary to observe those stations where economy in the production of power is the aim of the entire personnel, to obtain added proof as to the advantages of cleanliness. One need not look for highly polished brass railings or nickel-plated tools, but one does find no waste of oil, every tool in its place, an orderly supply cabinet or locker, convenient receptacles for clean and dirty waste, drip pans under the auxiliaries, neat piles of coal, feed pumps and lubricators free from accumulation of dust. These will be the most marked, but it is safe to predict that these visible signs are faithful indications of an excellence in the maintenance of the internal parts of the equipment.

As affecting the costs entering into the production of the output, the boiler room contains great possibilities of economic results where cleanliness is properly attended to. The confining of coal in neat piles on the firing floor, or in the containing cars, is not only pleasing to the eye, but firing and the cleaning of fires are simplified by the freedom afforded the firemen in moving about, and furthermore the coal does not become mixed with the ashes. Clean fires and clean ashpits mean efficient combustion as well as increased capacity. Clean drums and tubes, internally and externally, are likewise necessary for efficiency, an instance being known where an ordinarily well operated station increased its rate of coal consumption 2½ per cent. by neglecting the daily blowing or dusting of tubes for a week. And aside from the reduced coal consumption this cleanliness means longer lived grates and tubes, with the resulting decrease in maintenance charges. Clean feed pumps, plungers or rods, protected or kept free from grit, insure a longer life to those parts, as well as a reduction in charges for packing; while lubricators similarly attended to provide clean oil at all times to still further reduce the friction to be overcome. And increased friction means increased coal consumption.

In the engine room a systematic overhaul and cleaning of all parts, internal as well as external, reduce maintenance charges to a minimum, the consequence being increased useful life to the machine. Inspection of engine cylinders and turbine rotors discloses accumulation of carbon, scale and other foreign matter that may easily counteract much of the care devoted to the boiler-room equipment previously mentioned. Carbonized oil, due to an excess of lard or tallow, or metal flake carried over from the boilers with the steam, while blocking up steam ports of valves, is also sure to cause scored cylinders; and under certain conditions blades of turbines will accumulate scale to a degree hardly imaginable, but which is only disclosed by inspection, such a condition being obviously destructive of all the refinements accorded to the previous stages in the chain of conversions. Bearings, pins and boxes likewise need only to be examined to bear out this discussion, indicating the necessity for frequent cleaning of oil grooves if lubrication is to be effective.

This argument extends to the other machines making up the engine-room equipment.

Care should be given to offices and workshop in the same way; the effect of such a general scheme of cleanliness could not but be beneficial to each member of the operating crew.

As affecting the fixed charges, the surplus capacity in a plant could reasonably be lower where operating engineers of a high grade were known to be in charge, men who would insist upon the maintenance of all equipment in first-class operating condition. In this way the investment charges might be materially reduced. As to the depreciation charges, the assumption of a continued régime of cleanliness would hardly justify a reduction; yet to find at the end of its useful life in a particular station—or when it shall have been superseded—that the depreciation on a machine has been overestimated, is a much more satisfactory outlook than would be the reverse.

So, too, oily waste thrown around and oil-saturated floors—particularly in wooden structures—increase the fire risk, and this must be provided for by a higher insurance rate.

Granting then the improved efficiency of the equipment, as well as of the men, and the possible reduction in investment, depreciation and other fixed charges, it becomes obvious that for the successful operation of a power station the first essential is cleanliness.

A systematic and careful cleaning of every piece of apparatus in use in the station will in the beginning disclose weaknesses to be attended to before becoming too serious. The application of a wrench to a loose nut or bolt, or the careful cleaning of electrical apparatus from oil, might easily prevent highly destructive consequences. This attention should be given not only to the main units but also to pumps, heaters, condensers, all piping, traps, oil tanks, filters, water-measuring devices, scales, motors, etc.

Measurement of High Temperatures

By A. R. Maujer

With appreciation of the value of exact information relative to boiler operation comes a demand for instruments to measure the temperatures at various points in the boiler and flue. The six principal types of pyrometers available for boiler-plant use are described.

Pyrometry, or the measurement of temperatures above the range of the ordinary thermometer, has but recently become available for industrial uses. In many manufacturing processes the desirability of accurate temperature observations was recognized long before any pyrometer of practical value had been developed. At the present time, however, there are several types of pyrometers which are very satisfactory for commercial as well as scientific uses, and in some branches of industry the pyrometer has become indispensable. New uses for it are constantly being found. Its value in economical boiler operation is being more widely recognized. Indeed, it requires no great stretch of imagination to foresee the time when the pyrometer will be a device as familiar to the fireman as is the steam-pressure gage today.

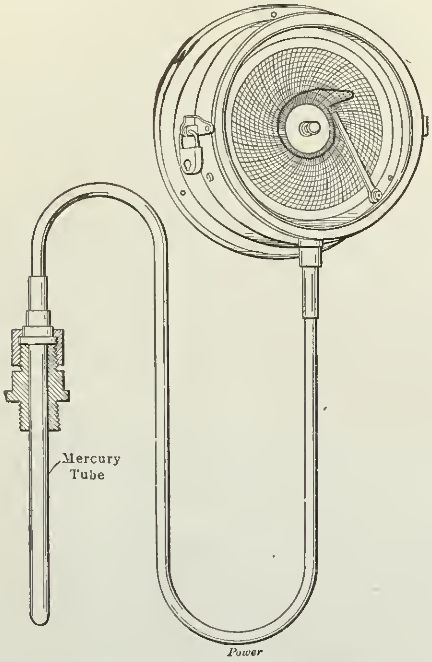


Fig. 1. Mercurial Pyrometer with Recording Gage

Six principal types of pyrometers available for boiler-plant work will be discussed, to wit:

Mercurial.
Expansion.
Calorimetric.
Electric.
Pneumatic.
Optical.

Mercurial Pyrometer

In the mercurial pyrometer a portion of a tube is filled with mercury and the remaining space with nitrogen gas. At atmospheric pressure, mercury will boil at 675 degrees Fahrenheit. As the tube is heated the mercury expands and compresses the nitrogen, the resultant increase in pressure on the mercury raises its boiling point. These pyrometers are made in many styles of both portable and stationary form. The end which contains the nitrogen may terminate in an ordinary pressure gage calibrated in degrees of temperature; such an instrument is more easily read than one which has only the ordinary thermometer type of scale. In place of the indicating pressure gage a recording gage may be used from which a chart can be taken showing the temperature changes throughout each twenty-four hours. In Fig. 1 is shown a typical mercurial pyrometer with recording gage and flexible connection tube.

Mercurial pyrometers are accurate, durable and cheap. They are excellently adapted to measuring uptake and chimney temperatures. They are constructed to read as high as 1000 degrees Fahrenheit.

Expansion Pyrometers

The difference in the rate of lineal expansion of two metals is the principle upon which expansion pyrometers are based. For the same rise in temperature, copper expands 50 per cent. more than iron. In the expansion pyrometer a tube of copper is inclosed in a tube of iron. At the end to be heated the tubes are securely fastened to each other. At the other end they are attached to a set of multiplying gears which actuates a needle pointer over the face of a properly calibrated dial. It is necessary to expose the entire length of the expansion tubes to the full effect of the heat to be measured; if this is not accomplished, error will result as the proper amount of elongation has not been obtained. When the pyrometer is first inserted, the pointer will act rapidly in one direction or the other and give an untrue reading temporarily. This is caused by the outer tube heating and expanding more rapidly than the inner one. As soon as the inner tube heats up and expands proportionately, the pointer will correctly indicate the temperature. When an expansion pyrometer has been used repeatedly for temperatures near its limit, the indicator will no longer return to the position indicating the temperature of the atmosphere. A permanent change has taken place in the length of one of the tubes. By loosening a set screw the dial may be shifted to correct the variation.

A standard type of expansion pyrometer is shown in Fig. 2. In spite of the fact that these pyrometers get out of calibration rather easily they are capable of giving close results if understood and carefully handled. They will indicate temperatures as high as 1500 degrees Fahrenheit.

Calorimetric Pyrometers

The method of indicating temperature by the calorimetric pyrometer is the same as that of indicating specific heats by the water calorimeter. In one case, from the known temperature the specific heat is calculated; in the other case, from the known specific heat the temperature is calculated.

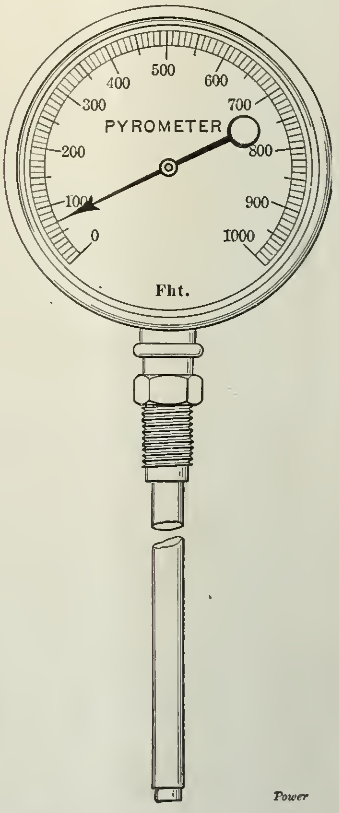


Fig. 2. Expansion Pyrometer

A given weight of some metal, the specific heat of which is known, is heated to the temperature to be measured and then instantly plunged into a known weight of water. The rise in temperature of the water is noted. The formula for finding the temperature is:

$$X = T + \frac{W\ t}{w\ s}$$

in which,

X = Temperature to be measured in degrees Fahrenheit;
T = Final temperature of the water;
W = Weight of the water in pounds;
t = Rise in temperature of the water;
w = Weight of the metal, in pounds;
s = Specific heat of the metal.

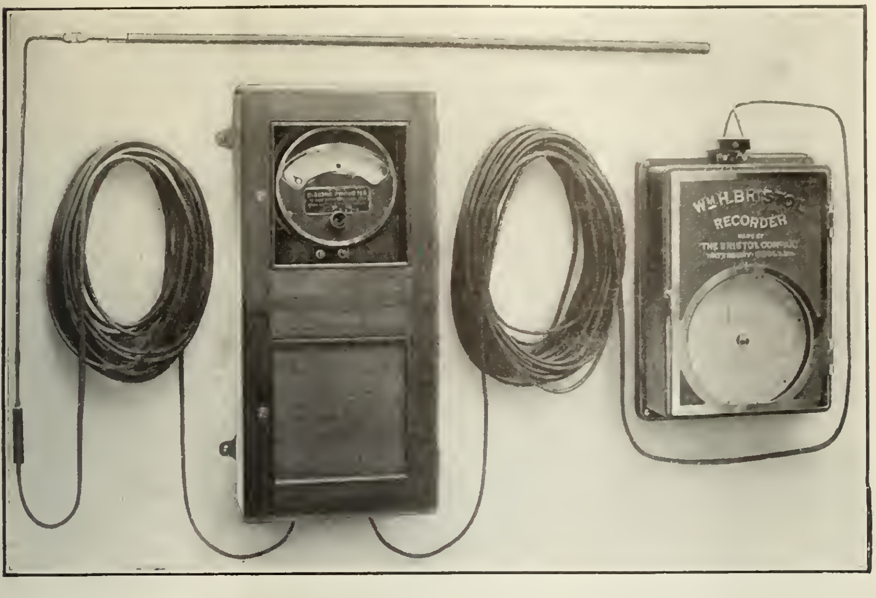


Fig. 3. Electric Pyrometer with Indicating and Recording Gages

A calorimetric pyrometer is an apparatus by means of which the temperature may be found without using each time the formula previously given. It usually consists of a copper cup which is insulated to prevent loss from radiation and which has gage marks upon it for a definite amount of water. Fastened to the cup and so arranged as to be properly immersed, is a small thermometer. A sliding scale is attached to the thermometer. A copper or platinum ball completes the outfit. The scale on the thermometer is calibrated with the quantity of water which the instrument holds and with the metal ball so that the rise in temperature of the water causes the thermometer to indicate correctly the temperature being measured.

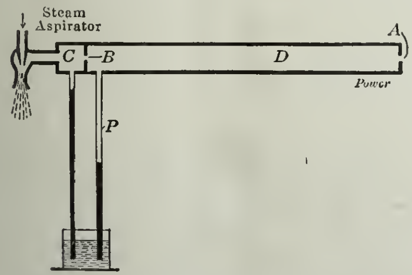


Fig. 4. Diagrammatic Sketch of Pneumatic Pyrometer

This pyrometer is very liable to inaccuracy. The results which it gives vary with the skill and care used in its manipulation. Its cheapness and portability, however, recommend it where approximate results are satisfactory. It may be used for temperatures up to 3000 degrees Fahrenheit.

Electric Pyrometers

When rods or wires of two dissimilar metals are joined at one end, they compose a thermoelectric "couple" or "element." When the junction is heated, a difference of electrical pressure is established between the cool ends. If a circuit is made by joining the cool ends together, either directly or by means of a conductor, a current will flow. The strength of the current depends upon the nature of the "element" and the difference in temperature between the hot junction and the cool ends or cold junction. The deflection of a galvanometer placed in the circuit may be calibrated, then, to indicate the temperature at the hot junction.

An electric pyrometer consists of the thermoelectric couple, the galvanometer and a device for compensating for the changes in temperature at the cold junction. In most makes for commercial use, this change is taken care of by shifting the scale on the galvanometer a few degrees one way or the other when the temperature at the cold junction is observed to change appreciably. Fig. 3 shows an electric-pyrometer outfit having both an indicating and a recording gage.

Electric pyrometers are adaptable to many uses. With proper handling they will give accurate results. Their great advantages are convenience and simplicity. A disadvantage is delicateness. Where high temperatures are measured and continuous service required, the cost of upkeep may be considered another disadvantage. Depending on the composition of the thermoelectric element, electric pyrometers will measure temperatures up to 3000 degrees Fahrenheit.

For temperatures up to 1100 degrees, the element is made of nickel and constantan; for 2300 degrees, nickel and a special carbon are used, and for 3000 degrees, platinum and platinum-rhodium make up the element.

Pneumatic Pyrometer

Under constant pressure the volume of air varies with the temperature. The pneumatic pyrometer makes use of this principle. Fig. 4 diagrammatically illustrates its operation. Due to the resistance of the small apertures A and B, the steam aspirator, working at a uniform rate, causes a partial vacuum in chambers C and D. If A and B are of the same size and if the air remains at a fixed temperature, the same amount of air will pass through each aperture during a given length of time and the suction in chamber C will be practically twice that in chamber D. If, however, the air is heated when it passes through A, but again cooled to a lower fixed temperature before it passes through B, more air will pass through B than through A during the same time. The effect will be to increase the suction in chamber D. In the same way, every change in temperature of the air flowing through A will have its influence on the amount of vac-

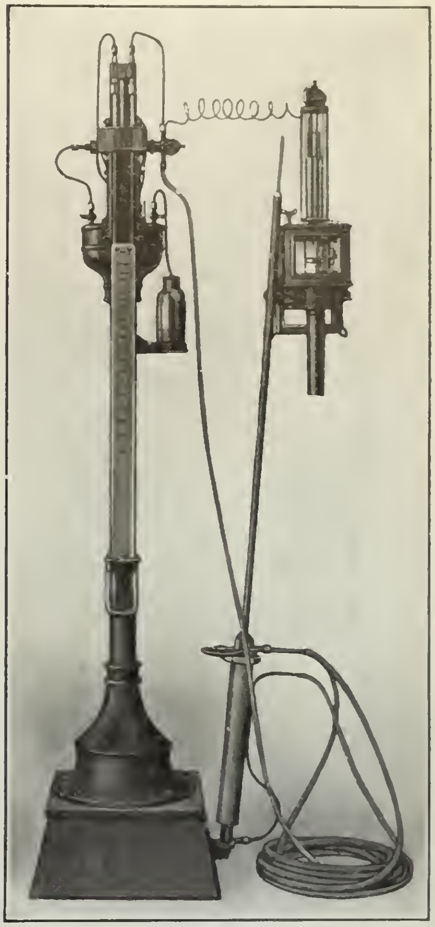


Fig. 5. The Pneumatic Pyrometer with Recording Gage

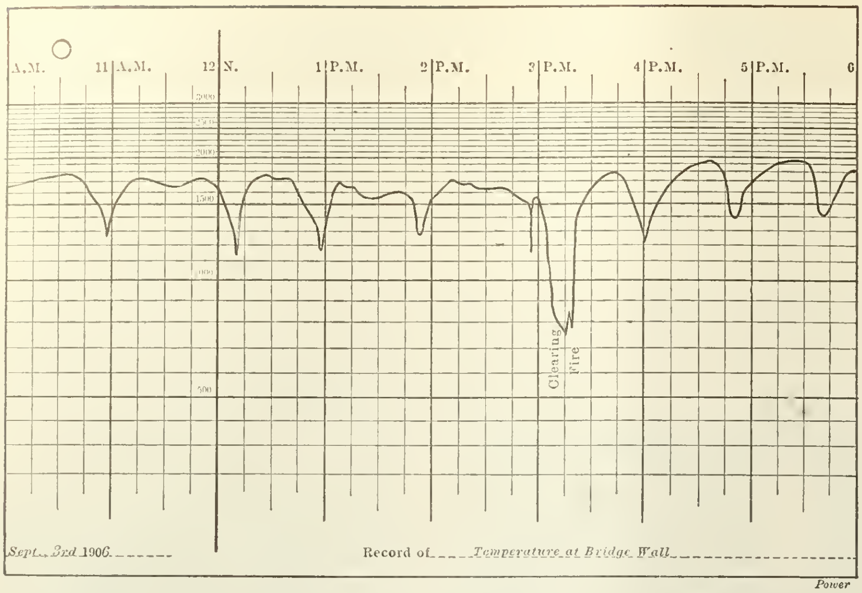


FIG. 6. EIGHT-HOUR TEMPERATURE RECORD FROM PNEUMATIC PYROMETER

uum in chamber *D*. Thus the manometer tube *P* may be calibrated to indicate the temperature of the air passing through *A*.

Fig. 5 shows the pneumatic pyrometer. The recording gage is simply a light-vacuum-recording gage which makes a record on a long paper tape carried on rollers. A record can be taken to cover a period of any length of time. Fig. 6 shows an eight-hour record of the temperature at the bridgewall of a hand-fired return-tubular boiler furnace burning a small size of anthracite.

That it is not portable is the most serious disadvantage of the pneumatic pyrometer. In its favor are accuracy and durability. It has a temperature range up to 3000 degrees Fahrenheit.

OPTICAL PYROMETERS

Estimating temperatures by the intensity of light radiated from a heated body is probably the oldest effort at pyrometry. Until recently, the only apparatus was the human eye "calibrated" by long practice. Study of the laws of radiation from incandescent bodies has led to the development of the optical pyrometer, an instrument which eliminates practically all error to which the eye is liable.

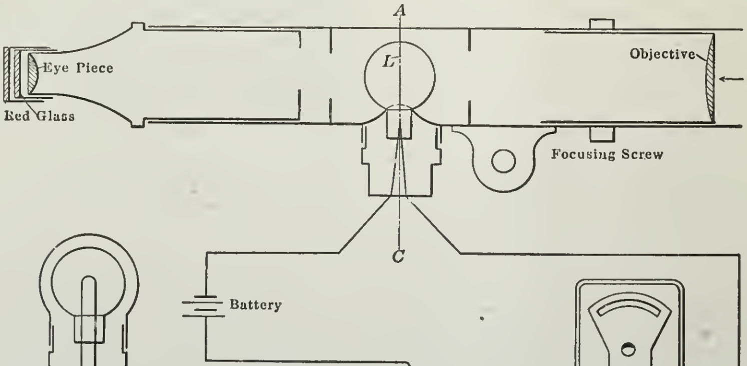


FIG. 7. HOLBORN-KURLBAUM OPTICAL PYROMETER

WANNER PYROMETER

In the Wanner optical pyrometer, by means of a series of lenses and prisms, the red rays of the light from the source of heat to be measured are compared with the red rays of the light from an incandescent lamp which is carefully maintained at a given brilliancy. The rays from each source are polarized in planes at right angles and viewed through a nicol analyzer. The analyzer is rotated until both sets of rays are of equal brightness. The angle of deviation from the normal position is calibrated in degrees of temperature.

LE CHATELIER PYROMETER

In the Le Chatelier pyrometer, instead of adjusting the two sets of rays to equal brightness by means of a polarizing device the rays from the source to be measured are intercepted by a cat's-eye diaphragm which is adjusted until the two sets of rays correspond in brilliancy. The size of the opening in the diaphragm is calibrated to indicate degrees of temperature.

HOLBORN-KURLBAUM PYROMETER

When a circuit of gradually increasing strength is passed through an incandescent lamp, the filament first glows red, then changes successively to orange, yellow and white. When the color of some part of the filament matches the radiated light the current passing through the lamp at that time is a measure of the temperature of the source of radiation. In the Holborn-Kurlbaum pyrometer, Fig. 7, a 4-volt lamp with a simple U-shaped filament is mounted at *L* in the focal plane of the objective and eye piece of a telescope. The radiating light is focused in the plane *A C*. The current through the lamp is varied until the tip of the filament seems to disappear against the bright background. The milliammeter is calibrated to read directly in degrees of temperature. The red screens over the eye piece are used when the temperatures to be measured are so high as to produce a brilliancy uncomfortable to the eye.

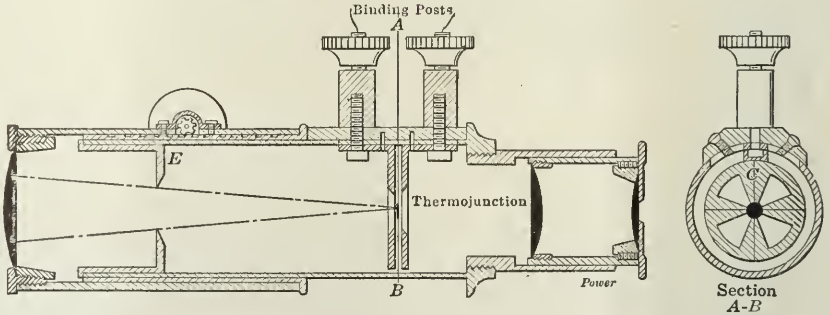


FIG. 8. FÉRY THERMOELECTRIC OPTICAL PYROMETER

FÉRY PYROMETER

In Féry's pyrometer the total radiation from the incandescent source is focused on a small thermocouple which is connected to a sensitive galvanometer and acts in the same way as the ordinary direct-contact electric pyrometer. The Féry pyrometer is shown in Fig. 8.

In general, it may be said that all optical pyrometers, being very delicate instruments, depend for their usefulness upon great care and intelligence in handling. They must frequently be calibrated and in most cases this cannot be done outside of the laboratory. For these reasons they are not well suited to ordinary use in the boiler room.

Development of Chain Grate Stokers

By J. F. Springer

The original chain-grate stoker was brought out in England by Juckes in 1841. The basic principles of which it made use are employed in the present-day types. An important feature of the modern chain-grate stoker is the flat combustion arch.

As far back as 1841, the British government granted to Juckes a patent for a traveling chain-grate stoker. About thirty years later, September 5, 1871, an American patent was issued to Royal F. Weller, of Albany, N. Y., for a similar device. This was, apparently, the first American stoker of this type, and for that reason it demands our attention, for chain-grate stokers have come into very extensive use indeed. Details of Weller's device are shown in Fig. 1. It will be noticed that this stoker differs from that designed by Juckes in the arrangement of the bars which go to make up the grate. Juckes arranged his longitudinally, while Weller placed his transversely. This effects a complete change in the method of connecting them. In Juckes' machine they were strung upon bars. Weller had them connected (see longitudinal view) by links working on studs projecting from the bars which, it will be observed, were cast in pairs. A little reflection will convince one that these studs, exposed as they would be to very severe temperatures, would be liable to severe damage. However, Weller provided that they should be of wrought iron, the grate bars being, in fact, cast around them. The motion of the ribbon of bars is accomplished by rotating one of the drums *J*, upon which it is arranged. Cogs or teeth project radially from suitable points around the ends of the drum body and engage the space between the members of a pair of bars. Two inclined grate surfaces *M* and *O* were arranged at either end of the upper surface of the ribbon. A hopper *Q* was placed at the front and above one of the inclined grates. If we recall James Watt's device of 1785, which was not a stoker but a furnace, we shall see that the hopper *Q* corresponds to the fuel chamber, while the grate *M* has its counterpart in the front opening in Watt's furnace. Here the similarity ceases, for the American's chain-grate device did not permit complete combustion to take place here, but carried the coal on toward the rear. It will be seen, however, that both devices operated to coke the fuel, before igniting it. The rear plate is hinged and may be raised to permit the moving ribbon to carry off any unconsumed fuel.

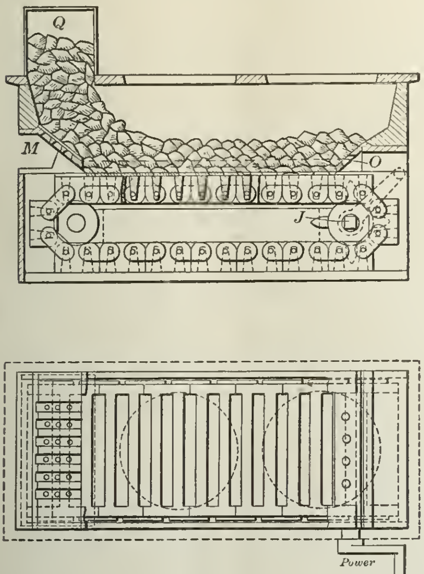


FIG. 1. ROYAL F. WELLER'S CHAIN-GRATE STOKER

This cross-bar method of fabricating the grate dispensed, it is true, with the stringing of a considerable number of bars on a single rod. This is, no doubt, a gain from one point of view. Still the operation of the longitudinal, interlaced bars of Juckes' machine had the advantage of stripping off clinkers as they passed over the rear drum. In patents issued to Simon Regan, of England, in Great Britain in 1874 and in the United States in 1877, the advantages of both methods of construction were obtained. This will be understood upon consulting Fig. 2, where the grate surface is shown. Each bar consists of a transverse rod having double arms arranged at intervals longitudinally. This device had the merit of breaking the fabric up into a number of individual units which could be detached with some approach to ease, and which yet effected the clinker-stripping movement in passing around the rear drum.

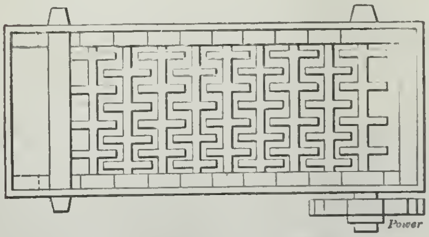


FIG. 2. REGAN'S CHAIN-GRATE STOKER

Eckley B. Coxe, of Drifton, Penn., was a rather prolific inventor and gave attention, among other things, to the question of the burning of coal in furnaces. With him, the coal in mind was particularly the smaller grades of anthracite. On June 20 and December 12, 1893, Mr. Coxe had issued to him as sole inventor no less than 26 patents upon distinct inventions relating to the question of fuel consumption. These patents were followed by still others up to the time of his death in about 1896. The traveling grate was the type upon which he spent his time. In what are, apparently, the two earliest of his patents (both June 20, 1893), Coxe provided an endless grate ribbon formed of sections composed of several transverse bars linked together at the ends. This ribbon was driven by wheels provided with cogs which coöperated with notches in the under side of the chain links. A striking characteristic of this stoker is the system of air supply. By referring to Fig. 3 a series of transverse air compartments *A, B, C, D*, located immediately below the grate surface, may be seen. These compartments were supplied by separate pipes, *E, F, G, H*, which could be controlled independently of each other. The object of this arrangement was to vary the pressure of air from point to point as the fuel receded into the furnace, the idea being that combustion would thus be carried on more perfectly, precisely that amount of air being supplied that experience and observation showed to be most effective at the various stages through which the fuel passed. Observation doors *I* and *J* were provided for the purpose of enabling this control to be carried out.

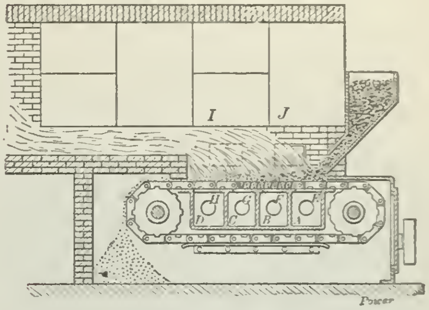


FIG. 3. COXES' CHAIN-GRATE STOKER

Coxe seems to have thought quite a variety of forms of grate to be possible. One form is an elaborate system of overlapping perforated plates, invented jointly by him and another. These plates were so supported and arranged as to separate and discharge from one to another as they passed around the rear wheel. It was thought that by this means the ashes and cinders would be discharged without clogging the mechanism.

As Juckes' British patent of 1841 was really the first disclosure of the chain grate, this inventor was the originator of a very important permanent idea in mechanical stoking. Without taking into account the sales of other chain-grate stokers, the importance of this type is shown by the fact that a single concern has already equipped a total boiler capacity of over 1,000,000 horsepower. The fundamental idea, which is at least 68 years old, is thus seen to have been commercially correct.

But, while Juckes, if he were still alive, could find a chain ribbon mechanically operated to carry the fuel through the furnace, he would see that the latest models differ from his in many important details. He would conclude that his idea of longitudinal interlocking grate bars must have been a good one, seeing that this style of bar has been returned to, although intervening history discloses attempts to depart from it. Upon examining the fabric of the modern chain grate, he would discover that his method of semi-permanently stringing the bars had been entirely abandoned for a system permitting the quick removal of any individual bar without the withdrawal of any cross bars. Remembering that ordinarily it was necessary with his machine to withdraw, partially at least, two rods, he would, no doubt, be impressed with this feature.

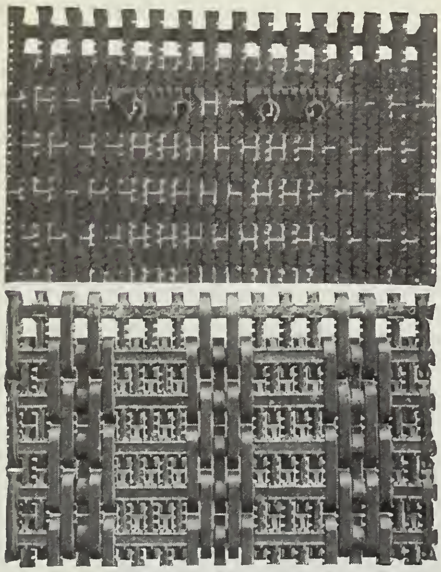


Fig. 4. Upper and Under Grate Surface

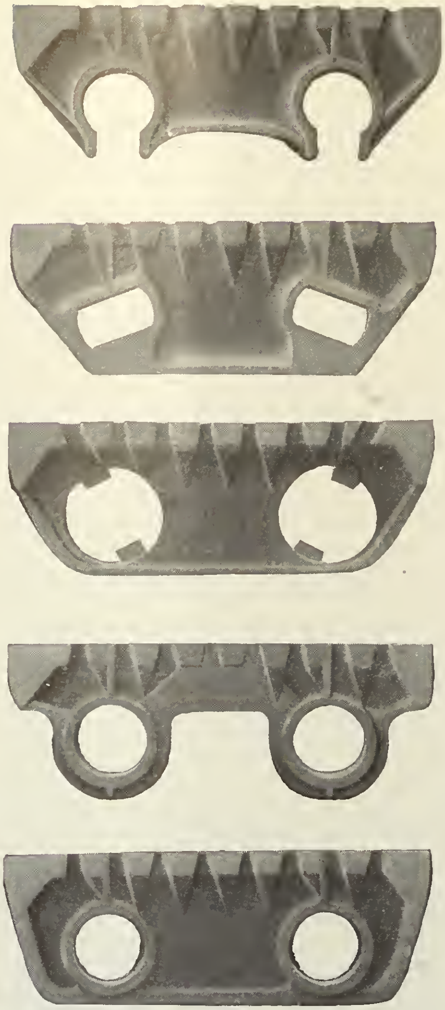


Fig. 5. Various Forms of Link

In order to understand just how the idea of individual removability is carried out in practice, reference should be made to Figs. 4 and 5. In Fig. 4 are shown the upper and under surfaces of a portion of the grate fabric. The vertical direction in the figure corresponds to the longitudinal direction in the furnace. By looking closely at the upper view, two loose links may be seen. The form of these is exhibited more clearly by the uppermost link shown in Fig. 5. All of the links in the fabric are of this style except those on the outer edges. These are of the style of the second and third links from the top in Fig. 5. The cross bars which bind the links into a whole are somewhat flattened in form. These bars may be seen in the lower view of Fig. 4. It will be seen that by turning the bars on their edges, the inner links may be individually removed or inserted. The outer links present no difficulty. These outer links, it may be remarked, must be removed before attempting to turn the bars. When these links are in place together with the cotter pins, the whole fabric has its integrity maintained. When making the rounds at the ends, the external links and the internal ones abreast of them, all remain fixed relatively to their cross bars. But the transverse row of connecting links turns somewhat on the rods. These connecting links must, therefore, all be interior ones with their more or less circular slots. That the connecting links are all interior ones may be seen by consulting Fig. 4. An inspection of the lower view in this figure shows that some of the interior links and all of the exterior ones have a straight under surface. This is to facilitate the movement of the whole ribbon on the dead rollers which support it between the sprocket wheels at either end. In the stoker in which this grate is used, if the width of the grate surface exceeds 7½ feet, two ribbons are used side by side. In this case, the links which form the adjacent edges of these ribbons are made of the special forms shown in the case of the two links at the bottom of Fig. 5. This is to permit the turning of the cross bars without their removal. It will be observed that there may be a relative movement of the links among themselves as they round the ends of their course. This is effective, just as with the old Juckes' type, in removing clinkers.

As the whole grate moves slowly round in its circuit, the links are completely overturned, so that their under surfaces are uppermost as they pass from the rear to the front. Right here is where attention to detail is necessary. The material sifting through from the upper grate surface will, some of it, lodge on the lower portion and perhaps tend to clog the under surfaces of the links. This would occasion difficulty at the sprocket wheels. Consequently, these latter are

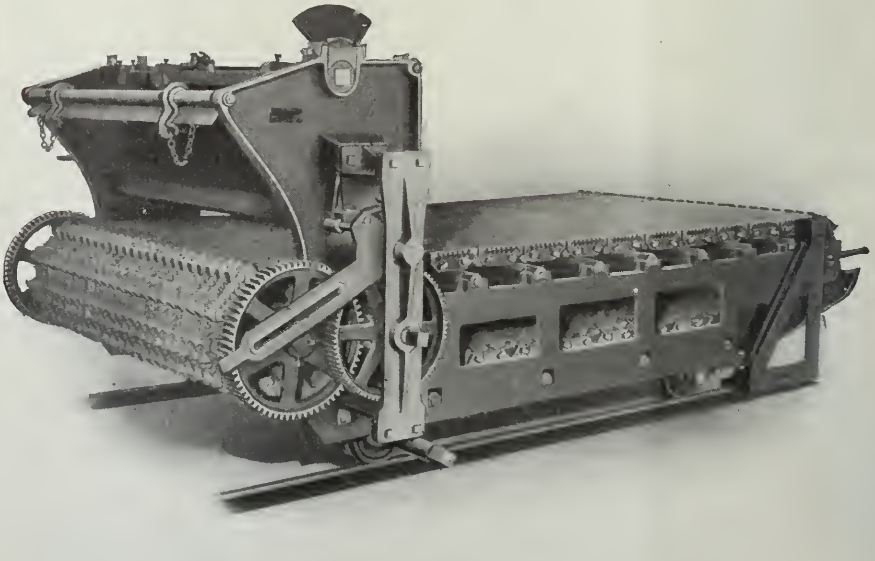


Fig. 6. The Green Chain-grate Stoker

hollowed out between teeth to insure a correct seat for the driving links.

By referring to Fig. 6 the complete stoker may be seen. It will at once be observed that the frame has its upper edges much below the grate surface. This is very important, as it is thus that protection from the excessive temperatures of the fire is obtained. It will be noticed that by means of the track the entire machine may be run into position. The usual length of the grate surface is 9 feet from the point where ignition occurs to a point above the rear sprocket shaft. For special conditions, longer surfaces are employed up to 12 feet. The feed gate, which is arranged to operate vertically at the rear of the hopper for the purpose of regulating the amount of fuel passing into the furnace, is protected from the effect of the heat of the fire by an inner facing of tiles. Trouble has, in the past, been found to arise from the fire penetrating forward into the hopper. The skeleton device for the support of the tiles and the tiles themselves are so arranged mutually that the latter may be removed or inserted individually and the bracket may be effectually protected.

A detail of considerable importance relates to the ignition arch. Under the older systems, this was curved. A very serious objection was the variable distance between arch and grate. Indeed, this difficulty was so serious as practically to limit the width of chain-grate stokers. And this in turn tended to restrict efficiency. A proper ratio between grate surface and heating surface is necessary to secure the best results. The modern chain-grate stokers may be installed with perfectly flat arches. The construction of the Green arch will be understood from Fig. 7. Two channels are placed transversely across the fore part of the furnace roof. A series of small I-beams are arranged connecting the two channels and depending from them. Specially formed bricks of refractory material are slipped in between the I-beams and supported by them. By means of this construction, the stoker may be installed having any width of grate surface without involving variations in the distance to the ignition arch. Thus, an evenness of conditions of combustion is brought about at the very start. Further, this style of arch is said to be more economical to maintain. It will be noticed that each brick sustains nothing but its own weight. There is no stress or strain induced by contraction or expansion. The flat arch is a notable step in the development of the chain-grate stoker.

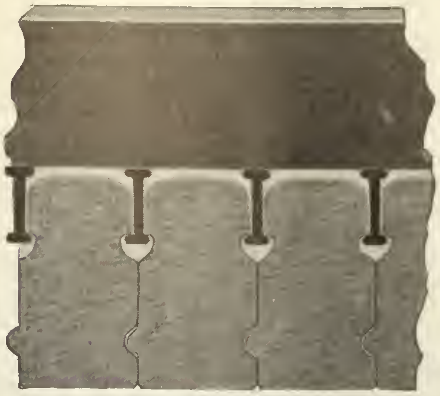


FIG. 7. CONSTRUCTION OF THE FLAT COMBUSTION ARCH

Cold and Its Production

By F. E. Matthews

In this article the subject of heat is discussed, its effects, as in producing change of temperature as well as change of state of matter, units by which it is measured, its absorption in the process of evaporation of liquids, laws which govern its flow from one body to another, and their application to both natural and artificial processes of refrigeration.

In its broadest sense, refrigeration may be defined as the process of cooling; or, since cold is simply the absence of heat, as darkness is the absence of light and dryness the absence of moisture, in which cases the real entities are heat, light and moisture respectively, refrigeration may be more accurately defined as the process of extracting heat.

The study of refrigeration, therefore, necessitates the study of heat. The above definition is, however, somewhat inadequate in that it conveys the impression that heat is a passive element, while on the contrary, it is inherently an active one. While heat may be generated by the performance of work, as when we compress gases or even when we strike a piece of metal a few sharp blows with a hammer, its appearance being incident to the disappearance of an equivalent amount of work. When once fortified within the walls of matter it is able to resist the most strenuous efforts to dislodge it, with the result that it must be decoyed into leaving the substance from choice. Heat can best be dislodged from one substance by placing near it another materially lower in temperature, in which case heat will flow from the substance having the higher to that having the lower temperature, just as water flows from a higher to a lower level. The result of the gravitation of heat from the substance having the relatively higher to that having the relatively lower temperature, is that the latter is heated and the former refrigerated. Since wherever there is a difference in temperature between two substances there is always a tendency for heat to flow toward that of the lower temperature, both heating and refrigerating may take place at any point above absolute zero.

Since refrigeration has to do wholly with the extraction of heat, in order to arrive at a clear understanding of the subject, one must first become familiar with the general thermal properties of the substances most commonly employed in the process, such properties for example, as their capacity for absorption of heat under different conditions.

Matter may exist in three different states; the solid, the liquid and the gaseous, according to the amount of heat which it contains.

The molecular theory of matter briefly stated, is that all matter consists of minute particles called molecules, the smallest division which retains the properties of the substance; the molecules are in turn made up of smaller particles called atoms, which may vary greatly in character. For instance, water (H_2O) is made up of molecules, which are composed of atoms, not of water, but of hydrogen (H) and the oxygen (O).

The kinetic theory holds that these molecules and atoms are in constant motion, in a much restricted way in the solid, that they move with slightly more freedom in the liquid, and with comparatively great freedom in the gas. The application of heat, supplying to the molecule certain energy, so that its motion is increased will cause the transition from the solid to the liquid and finally to the gaseous state. It follows that the removal of heat will have the opposite effect of reducing the gas back to the liquid and the liquid to the solid state, as the energy is withdrawn from the molecules.

Heat may effect change in condition as well as changes of state of matter. For example, the application of heat to a piece of iron first effects a change in its condition, i.e., increases its temperature. When the melting point is reached the further application of heat effects a change from the solid to a liquid state. A solid substance will absorb a certain definite amount of heat for each degree of rise in temperature, and the ratio of the amount of heat absorbed, per unit of weight per degree rise in temperature, to that of water, is known as the "specific heat of the solid."

When the melting point is reached the further application of heat will produce no appreciable rise in temperature until all is melted, and the amount of heat absorbed, divided by that amount required to increase the temperature of a unit weight of water one degree, is the "latent heat of fusion," or "latent heat of the liquid."

When all of the substance is melted the further application of heat will again produce a rise in temperature and the ratio of the amount of heat absorbed per unit weight per degree rise in temperature to that of water is "the specific heat of the liquid."

When the boiling point of the simple* substance in question is reached, the further application of heat will produce vaporization, which will progress at a constant temperature until all is evaporated and the ratio of the amount of heat absorbed to that required to raise the temperature of a unit weight of water one degree is the "latent heat of vaporization of the liquid," or "latent heat of the vapor."

Objection might logically be made to the term latent heat since the heat disappearing when solids melt and liquids evaporate really no longer exists as heat, but has been expended in doing work, just as heat units expended in a steam engine are converted into an equivalent amount of work measurable in foot-pounds. A common example of this is the raising of a weight of a certain number of pounds through a certain number of feet of space against the attractive force of gravity. Heat absorbed in the process of melting and evaporation is expended in doing interior work, or in overcoming gravitational forces between the molecules of the substance melted or evaporated. The expenditure of heat in separating one molecule from another against the force of molecular gravity, differs in no material sense from that expended in prime movers in separating heavy weights from the earth against the force of gravity.

When all the substance is evaporated the further application of heat will again produce a rise in temperature and the ratios of the amount of heat absorbed per unit of weight per degree of rise of temperature of the gas is the "specific heat of the gas."

The standard unit generally adopted in commercial work among English speaking people is the British thermal unit (B.t.u.) which is the equivalent of the amount of heat required to raise the temperature of one pound of water through one degree Fahrenheit at its temperature of maximum density, 39.1 degrees Fahrenheit. This, it will be remembered, Joule found by his historic experiment to be equivalent to 772 foot-pounds of work. More recent experiments, however, have led to the adoption of 778 as a more accurate equivalent. In more purely scientific work the French or metric unit, the calorie, is employed. This unit is the equivalent of the amount of heat required to raise the temperature of one kilogram (2.2 + pounds) of water through one degree Centigrade (1.8 degrees Fahrenheit) at its maximum density, a temperature of four degrees Centigrade.

By the following experiment, Doctor Joule determined the mechanical equivalent of heat. A paddle fitted to a shaft made to revolve by weights so constructed that the exact amount of work done by them could be measured, was placed in a cylindrical vessel containing a definite amount of water at a known temperature. As the paddle revolved the water was agitated and the temperature was found to rise. The energy of the weights had been converted first into motion of the paddle, then into heat in the water. He determined that when mechanical energy is converted into heat the amount of heat produced is proportional to the mechanical energy expended, and specificially, that one calorie represents 424 kilogram-meters of energy. The mechanical equivalent of heat then, or as it is called Joule's equivalent, is 424 kilogrammeters. In the English system the same equivalent is 772.55 foot-pounds, or one B.t.u., later definitely fixed as 778 foot-pounds.

Since the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit, has been made the standard unit by which other quantities of heat are measured, the latent and specific heats above defined are expressed in B.t.u. The specific heat of water is unity.

Since the amount of heat required to raise the temperature of one pound of ice one degree is one-half that of water, the specific heat of water in the solid state is one-half B.t.u., or simply 0.5. Similarly the latent heat of fusion of water is 144,* the latent heat of vaporization is 966.1, and the specific heat of steam, according to whether it is taken at constant pressure or constant volume is 0.480 and 0.346, respectively.

It will be noted that the specific heat of constant pressure is somewhat greater than the specific heat of constant volume. In determining the latter, the gas is not allowed to expand, in other words, it is maintained at a "constant volume." In determining the former, the volume is allowed to increase with the addition of heat just sufficiently to maintain a "constant pressure." The work done by the gas in expanding is equivalent to the excess of the amount of heat supplied in the case of constant pressure, over that required to produce the same rise in temperature at constant volume.

Under a given pressure there is for every substance, a definite and fixed temperature at which that substance will change from one of its three states, the solid, the liquid, or the gaseous, to another, and at each change that substance absorbs or liberates an amount of heat, which, though varying slightly at different temperatures, is more or less a constant and characteristic of that particular change of state of that particular substance. So constant is the temperature at which these changes of state take place that the simple determination of the boiling point under atmospheric pressure, of some of the more common liquids such as alcohol, water, ammonia, etc., is sufficient to establish their identities. At atmospheric pressure alcohol boils at 173 degrees Fahrenheit, water at 212 degrees Fahrenheit and ammonia at 28½ degrees Fahrenheit. Except for the question of the facility of conducting the experiments, the freezing points of the liquids in question might just as well be employed to determine their composition. Absolute alcohol, for example, freezes at 202 degrees Fahrenheit and water at 32 degrees Fahrenheit. In general, the addition of a foreign substance capable of being dissolved in either of these liquids has the effect of lowering the freezing point and raising the boiling point.

Similarly, the amount of heat absorbed or liberated when a fixed quantity of one of the above liquids changes state, is so nearly a constant that it could be employed to establish the identity of the liquid were it not for the difficulty of determining the exact amount of heat involved. A pound of water, for example, absorbs 144 B.t.u. in changing from the solid to the liquid state, and 965.7 B.t.u. in changing from the liquid to the gaseous state. Similarly, in changing from the liquid to the gaseous state a pound of anhydrous ammonia absorbs 555 B.t.u.

It has been stated above that there is always a tendency for heat to flow from one body to another wherever there is a difference in temperature. For the present purpose, it may be generally stated that the rate at which the passage of heat takes place is directly proportional to the difference in temperature, and inversely proportional to the amount of resistance offered to its passage by interposed substances. The flow of heat is analogous to the flow of electricity in an electrical conductor which is inversely proportional to the voltage as expressed in Ohm's law

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*In the case of complex substances, such, for example, as mineral oils, no absolutely fixed boiler points exist. A moderate application of heat to crude oil will drive off one after another of the distillates, such as benzine, gasolene, kerosene, etc., but while the temperature at which each distillate is driven off is different from that of the others, the increase in temperature as distillation progresses is gradual and that distillation temperatures define the product, rather than the product the temperatures.

*At the New York meeting of the American Society of Mechanical Engineers, a committee appointed by that society to suggest a standard tonnage basis for refrigeration proposed as a unit for measuring cooling effect, the equivalent of the heat required to melt one pound of ice, i.e., 144 B.t.u. The unit for a ton of 2000 pounds of ice-melting capacity was then fixed at 288,000 B.t.u., which, since the rating is always expressed in tons per 24 hours, makes a ton duty equivalent to the rate of 12,000 B.t.u. per hour, or 200 B.t.u. per minute.

$$C = \frac{E}{R}$$

where C is the current in amperes, E the voltage and R the resistance in ohms. The flow of heat may also be compared to the flow of water through a pipe, which is also proportional to the difference in pressure and inversely proportional to the friction.

Passage of heat may take place by convection, conduction, or radiation, and in the general case, by all three methods simultaneously, so that the mathematical expression for the total heat transfer between two bodies of different temperatures becomes somewhat complex. Fortunately, in the greater number of engineering problems the amount of heat transmitted by conduction is so far in excess of that transmitted by radiation and convection that the last two factors are ignored entirely or are introduced in terms of conduction.

Where the refrigerating effect is due to the direct passage of heat from one substance to another, it is almost axiomatic that the cooling of one can take place only with the equivalent heating of the other.

The most common examples furnished by nature, of processes by which the refrigerating of one body is accomplished by a corresponding heating of another, are the changing of water into ice and its subsequent melting to form water. In the former process, heat passes from the water to the air, the water being refrigerated and the air heated. Conversely in the latter process, heat passes from the air to the ice, the air being refrigerated and the water heated. Both these processes ordinarily take place under atmospheric pressure at 32 degrees Fahrenheit.

Among others of this class of examples, i.e., that in which the refrigerating effect produces change from the liquid to the solid state, may be cited the congealing and subsequent melting of mercury at 39 degrees Fahrenheit, and that of cast iron at about 2000 degrees Fahrenheit, or in fact that of any fusible substance at its temperature of fusion. In the foregoing examples, absorption of heat, or refrigeration, involves the latest heat of fusion of the substances, water, mercury and iron, in question. Another means of bringing about the absorption of heat or refrigeration is by the evaporation of liquids. This involves the latent heat of vaporization, and the fact that the latent heat of vaporization of a substance is greater than the latent heat of fusion is one reason why the former method is the more commonly employed in connection with artificial refrigerating systems.

Probably the most common example furnished by nature, of refrigeration produced by the evaporation of a liquid, is the cooling effect of summer showers, in which the evaporation of a part of the water precipitated cools the dry, hot air which absorbs it. Another well known cooling effect is encountered when we sit in a draft after perspiring freely. The effect of the motion of the air is simply to displace the stratum of warm, saturated air lying next the skin by cool dry air which, in absorbing more moisture produces a refrigerating effect, which often actually lowers the temperature of the body and produces such an unbalancing of its functions as to cause so called colds or even pneumonia. Among less common examples may be cited the methods of cooling drinking water sometimes employed on shipboard, i.e., by exposing it to the wind in porous tile vessels, the evaporation of a part of the water through the walls of which refrigerates the portion remaining in the vessel. In India ice is actually frozen by the rapid evaporation of water exposed in shallow earthen trays to the clean, dry night air.

Evaporation can take place only with absorption of a fixed quantity of heat. Conversely, absorption of heat, or refrigeration always occurs when a substance is evaporated, whether the evaporation takes place rapidly at the boiling point or slowly at a temperature far below its boiling point. Since no two simple substances boil under the same conditions of temperature, it follows that it may be possible to produce refrigeration either at a given temperature by the evaporation of different substances at different pressures, or within certain limits, to produce refrigeration at quite widely different temperatures by the evaporation of the same substance at different pressures.

While the availability of the latent heat of vaporization of water at 32 degrees Fahrenheit requires the usually prohibitive vacuum of 29.76 inches of mercury, the fact that the latent heat of fusion of water is available at 32 degrees Fahrenheit under atmospheric pressure, allows artificial ice to become as important a factor in our general scheme of domestic and commercial economy as natural ice, the freezing and subsequent melting of which in our lakes and rivers protects both animal and vegetable life through the tempering of extreme temperatures in natural economy.

Much of the popular misconception of the art of refrigeration has arisen through our proneness to group temperatures into such technically meaningless classes as "warm," "hot," "cool" and "cold," according to their apparent relation to the widely varying temperatures of our surroundings, which we erroneously come to look upon as a kind of variable zero from which all other temperatures should be measured. This in turn gives rise to the erroneous notion that a substance is capable of heating or refrigerating other substances according to whether it appears hot or cold to the touch.

If a person places his hand on a body much lower in temperature than his body, he describes it as being cold. If, in the other case, it has a temperature higher than that of his body, he describes it as being hot. Since heat and cold are relative terms arising from comparison with normal temperatures, a temperature described as being hot in winter might be regarded as being cold in summer. The fact that a body is already cold does not prevent its being made still colder by the further removal of heat. The coldest substances known are still possessed of a large quantity of heat, only part of which can be abstracted by any known method. Could all of this heat be removed from a substance, the resulting temperature would be absolute zero, or about 46 degrees below our present Fahrenheit zero. At this point all chemical action would cease, and neither animal nor vegetable life could exist.

Since refrigeration, which occurs whenever there is a flow of heat from a relatively warmer to a relatively cooler body, may take place at any temperature regardless of whether it is above or below that of our surroundings, the melting of iron in a blast furnace may be said to refrigerate the contents of a blast furnace just as truly as the melting of ice does the contents of a refrigerator.

In general the most desirable working medium is that substance which has either its latent heat of vaporization or latent heat of fusion available under not too abnormal conditions. The melting point of ice being 32 degrees Fahrenheit under atmospheric pressure, the latent heat of fusion of water is available at a most convenient temperature and that substance has accordingly found an almost unlimited application in the arts. But, while evaporation of water in the tubes of a water-tube boiler refrigerates the furnace gases, in almost the same way as the evaporation of other liquids (so called refrigerating media), do the air surrounding the similar pipe coils in direct-expansion refrigerating systems, the fact that the latent heat of vaporization of steam is not generally available at less than 212 degrees Fahrenheit under atmospheric pressure, has up to the present time, prevented the extensive use of that medium in connection with what are commonly termed "refrigerating systems."

Using a Sewer for Condensing

> *Increasing capacity of the McDonald-Road station at Edinburg by installing low-pressure turbines and condensers using water from the sewer for cooling, has reduced the cost of fuel per unit generated 33.4 per cent.*

In a paper recently presented to the Incorporated Municipal Electrical Association, at Glasgow, Frank A. Newington, chief electrical engineer of Edinburgh, told how the capacity of that station had been increased and the cost per unit reduced by the use of low-pressure turbines, using the water of a neighboring sewer for condensing purposes. The plant is that at the McDonald-Road station at Edinburgh, and was designed for the use of cooling towers, the installation of which had to be abandoned on account of the fear that the towers would be unsightly and might cause a nuisance by scattering moisture about the neighborhood.

In 1907 it was found that the Leith main sewer, which passed about 180 yards from the station, had an average flow of about 750,000 gallons per hour, which could be utilized for condensing purposes. Four suction pipes, each 18 inches in diameter, and one discharge pipe 26 inches in diameter were laid from the shaft to the station. Straining screens were fitted in the shaft in front of the suction pit and the end of the discharge pipe in the sewer is sealed, so that siphonic action is maintained. There is considerable air in the water, so that two extracting pumps have been provided, one acting upon the air on the suction side of the pump, and the other on the delivery. In addition to the screening in the pit, the water is passed through a Bailey & Jackson rotary strainer, in which the returning water is made to wash away the deposit left by the water as it enters. Four surface condensers of the contrà-flow type, each having a cooling surface of 7750 square feet, with Edwards air pumps are used. The temperature of the water in the sewer averages about 60 degrees in the winter and 70 in the summer. As it must be returned to the sewer at not above 90 degrees, the range is not large, and the quantity of circulating water required per pound of steam considerable.

At present only two exhaust-steam turbo-generators have been installed, but there is sufficient steam from the reciprocating engines for two more exhaust turbines, and these will be added as soon as the increase in output requires it. The turbines are of the Rateau type, running at 1250 revolutions per minute, with a guaranteed output of 1200 kilowatts, about 2200 amperes and 560 volts, with 45,600 pounds of steam per hour, at a pressure at the inlet to the turbine not exceeding 14.7 pounds per square inch absolute, and at the exhaust 1.3 pounds absolute—say 27¼ inches vacuum. This output is easily attained in every-day work. Live steam is connected to each turbine through a reducing valve in case the supply of exhaust steam at any time should not be sufficient. The rotor of each turbine has 11 wheels with 265 blades to a wheel. The exhaust-steam piping is arranged so that the steam from the reciprocating set can pass either to the exhaust-steam turbines or direct to two other condensers. Oil separators of the Blake type are placed in the exhaust pipes between the reciprocating engines of the turbine, which extract the greater part of the oil and the feed water is further treated by the Davis-Perrot electrical process to free it from the rest. The water from the air-pump discharge passes into wooden tanks, in which are a number of iron plates spaced about 2 inches apart. A current of electricity is passed through these tanks with the object of precipitating finely divided oxide of iron. The small globules of oil adhere to the oxide of iron and are easily separated from the water. The treated oily water then passes into filters containing gravel and sand, which retain the oil. The current required is one kilowatt-hour per thousand gallons. The makers guarantee that the filtered water will not contain more than 0.1 grain of oil per gallon, and after 12 months' steaming the boilers show hardly a trace of oil. The author concludes his paper as follows:

Cost of Plant.—The condensing plant is sufficient to deal with the steam from reciprocating and exhaust turbines of 11,000 kilowatts.

The capital cost was as follows:

	£	Per Kw. £
Sewer shaft with four 18-inch suction and one 26-inch discharge pipe in roadway	5,600	0.5
Circulating-water exhaust-steam pipework inside the station and oil-extracting plant	6,000	0.5
Rotary water strainer	2,000	0.2
Four condensers circulating and air pumps	11,000	1.0
	£24,600	2.2
Two 1,200-kilowatt turbo-generators with switchboard	13,800	5.7

The economy caused by the exhaust-steam turbines and condensing plant is shown by the following figures, which are for similar periods in 1908 when only reciprocating engines were used without condensation, and in 1910 with exhaust-steam turbines and condensation. In 1908 the price of coal was considerably lower—10d. a ton—than in 1910, but for the purpose of comparison the 1910 price has been taken for both periods. In the costs per unit generated for the 1910 figures, the units used by the condensing plant have been deducted. As only three of the boilers are fitted with superheaters the amount of superheat is quite small:

	1908.	1910.
Duration of test	3 weeks	3 weeks
Date	Jan. 16 to Feb. 6	Jan. 20 to Feb. 9
Units generated	715,491	888,644
Units used by condensing plant	....	43,045 equals 4.8%
	715,491	845,599
Load factor	22.8%	25.1%
Plant factor	89.6%	76.9%
Pounds of water per unit generated	39 lb.	22.2 lb.
Pounds of water evaporated per pound of coal	6.67 lb.	6.6 lb.
Pounds of coal per unit generated	5.85 lb.	3.9 lb.
Maximum output	6.227 kw.	7,021 kw.

COSTS PER UNIT GENERATED

			Saving
Fuel	0.251d.	0.167d.	33.4%
Oils and stores	0.009d.	0.004d.	50 %
Water	0.027d.	0.006d.	78 %
Salaries and wages	0.054d.	0.047d.	
Repairs (estimated)	0.057d.	0.057d.	
	0.398d.	0.281d.	30 %

At present at the time of top load in the winter months, three or four reciprocating engines are exhausting direct to the condensers, so the economy will be increased when the other two turbines are installed.

Some engineers seem to think that exhaust-steam turbines are only a passing fashion and will disappear in a few years, but in stations with existing reciprocating engines, either exhaust-steam or mixed-pressure turbines seem the most suitable for extensions, as the combination of reciprocating engine and turbine is as economical as high-pressure turbines and obviates all necessity of scrapping the old sets until these are worn out.

As shown by the figures giving the capital costs, the price of the sewer shaft and pipe work in the roadway comes to £5 per kilowatt. This is considerably less than cooling towers would have cost. In places where a river or other sufficient supply of water is not available for condensation, a main sewer, if near the generating station, is well worth consideration.

On December 31, 1908, the length of the steam railroads of the United States was 232,045.9 miles. Accurate figures for electric railroads are not available, but on approximately the same date the length of the street and interurban railroads was in round figures, 35,000 miles, or about one-seventh the railroad mileage of the country.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Primer of Electricity

FRICTION LOSSES OF A DYNAMO

Besides the losses in the armature core, in the armature winding and at the contacts between the brushes and the commutator due to contact resistance, the rubbing of the brushes on the commutator causes a loss due to friction, the turning of the shaft in its bearings causes a similar loss and there is a slight loss due to the rapid motion of the armature surfaces through the air; this latter loss is called "windage," because the energy thus lost is used up in setting the air in motion.

BRUSH FRICTION

The friction loss at the brush faces varies with the quality of the carbon in the brush, the condition of the commutator surface, the speed of rotation and the pressure of the brushes against the commutator. It is impracticable to estimate the effect of different conditions of the commutator surface, but the effects of brush quality and pressure and commutator speed are quite regular. The friction loss in watts per square inch of brush contact surface is equal to

0.0226 × *brush pressure per square inch* × *velocity* × *k,*

the velocity being that of the commutator surface in feet per minute and the symbol *k* representing the coefficient of friction. This coefficient depends on the quality of the brush and the condition of the commutator surface. The velocity of the commutator surface in feet per minute is, of course, equal to 3.1416 × diameter in inches × revolutions per minute ÷ 12, the division by 12 being made to reduce inches to feet.

For commutators in reasonably good condition and brushes of the hard and soft grades ordinarily used in this country, the chart, Fig. 57, gives the friction loss that will usually occur at brush faces. For example, suppose a commutator 6 inches in diameter runs at 900 revolutions per minute and soft carbon brushes are used. The diameter × speed = 6 × 900 = 5400. If the spring pressure on the brushes is 1½ pounds per square inch of contact area, the friction loss will be 21½ watts per square inch of contact area.

The loss is ascertained from the chart by starting at the point on the side scale that would be marked 5400 if there were room to mark every division of the scale, tracing along this line until the diagonal line marked "Soft Carbon 1½ Lbs. Pressure" is reached; then tracing straight up or down from this point to either of the horizontal scales. Each division of the horizontal scales represents one watt, and the diagonal line for soft carbon at 1½ pounds pressure crosses the 5400 horizontal line about half way between the vertical lines corresponding to 21 and 22 watts per square inch; hence, the result is 21½ watts per square inch.

Now suppose the brush pressure were increased to 2 pounds per square inch. Starting at 5400 in the vertical scale again, follow that line until the diagonal line for soft carbon at 2 pounds pressure is reached. This falls between the

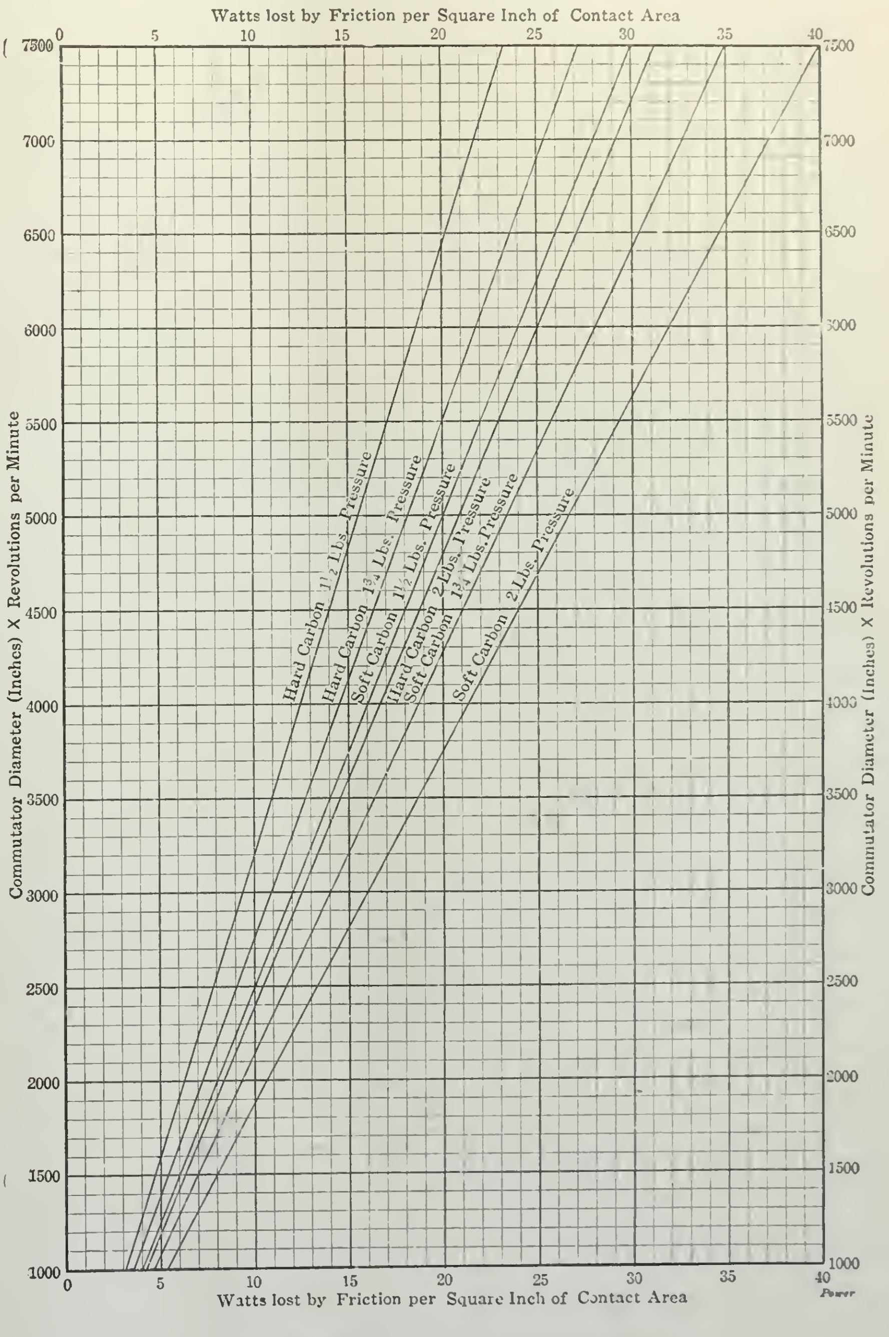


FIG. 57. FRICTION LOSS AT CARBON BRUSH FACES

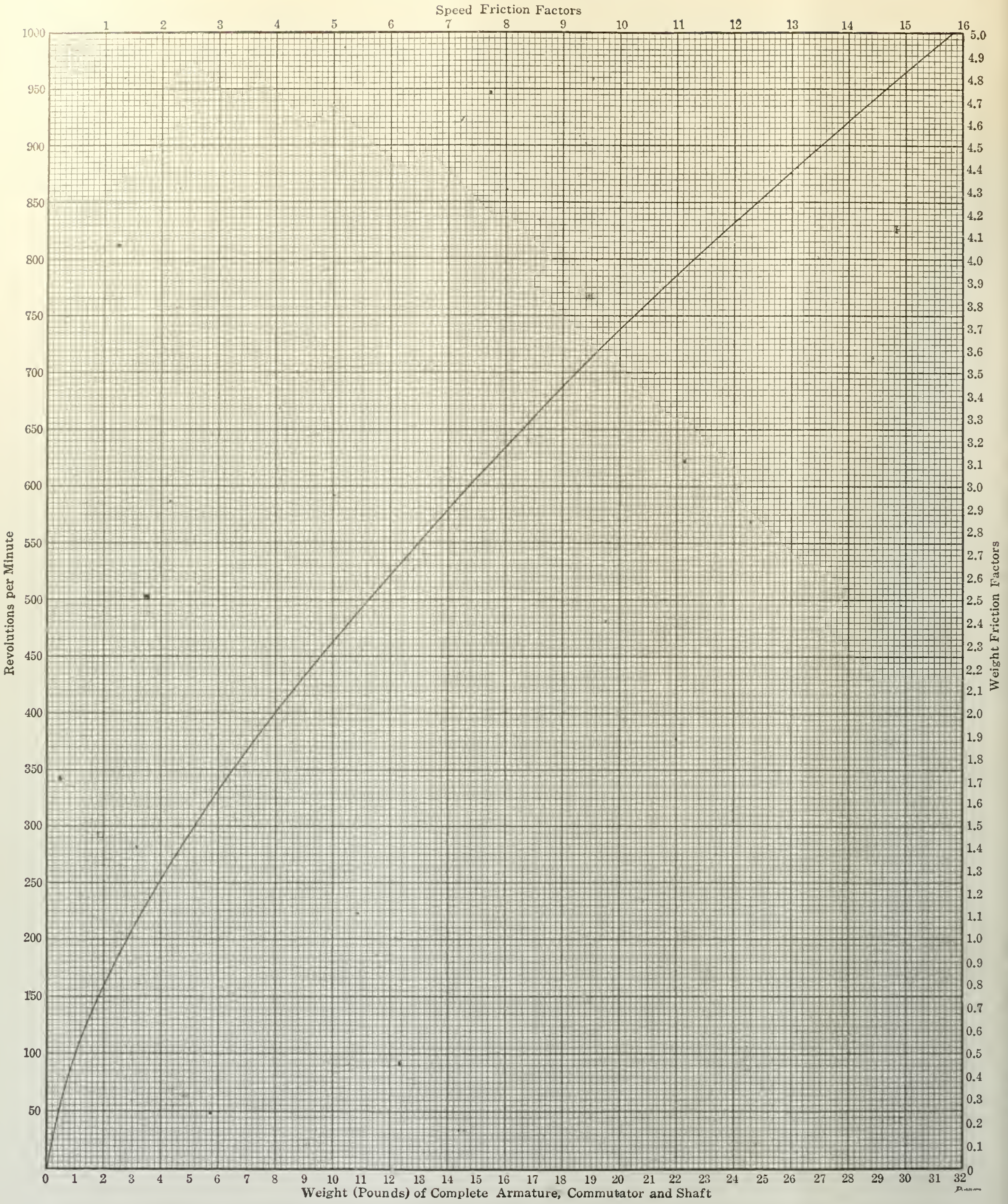


Fig. 58. Chart Giving Approximate Loss by Armature Bearing Friction and Windage

vertical lines corresponding to 28 and 29 watts, and is about three-quarters of the distance from the one to the other; the friction loss, therefore, would be about 28¾ watts per square inch of contact area.

The student will find it interesting to compare the results obtained with hard and soft carbon, as well as the differences caused by changing the brush pressure on the commutator. For example, if a commutator 14¾ inches in diameter and running at 400 revolutions per minute were worked with soft carbon brushes at 1¾ pounds per square inch pressure, the friction loss would be 27½ watts per square inch of contact area. Now if hard carbon brushes were substituted, the speed and brush pressure remaining the same, the friction loss would fall to 21½ watts per square inch of contact.

The question of using hard or soft brushes, however, cannot be decided merely from the comparative friction losses; it is governed by much more important considerations, which will be explained in a future lesson.

It should be noted and remembered that all of the quantities discussed in

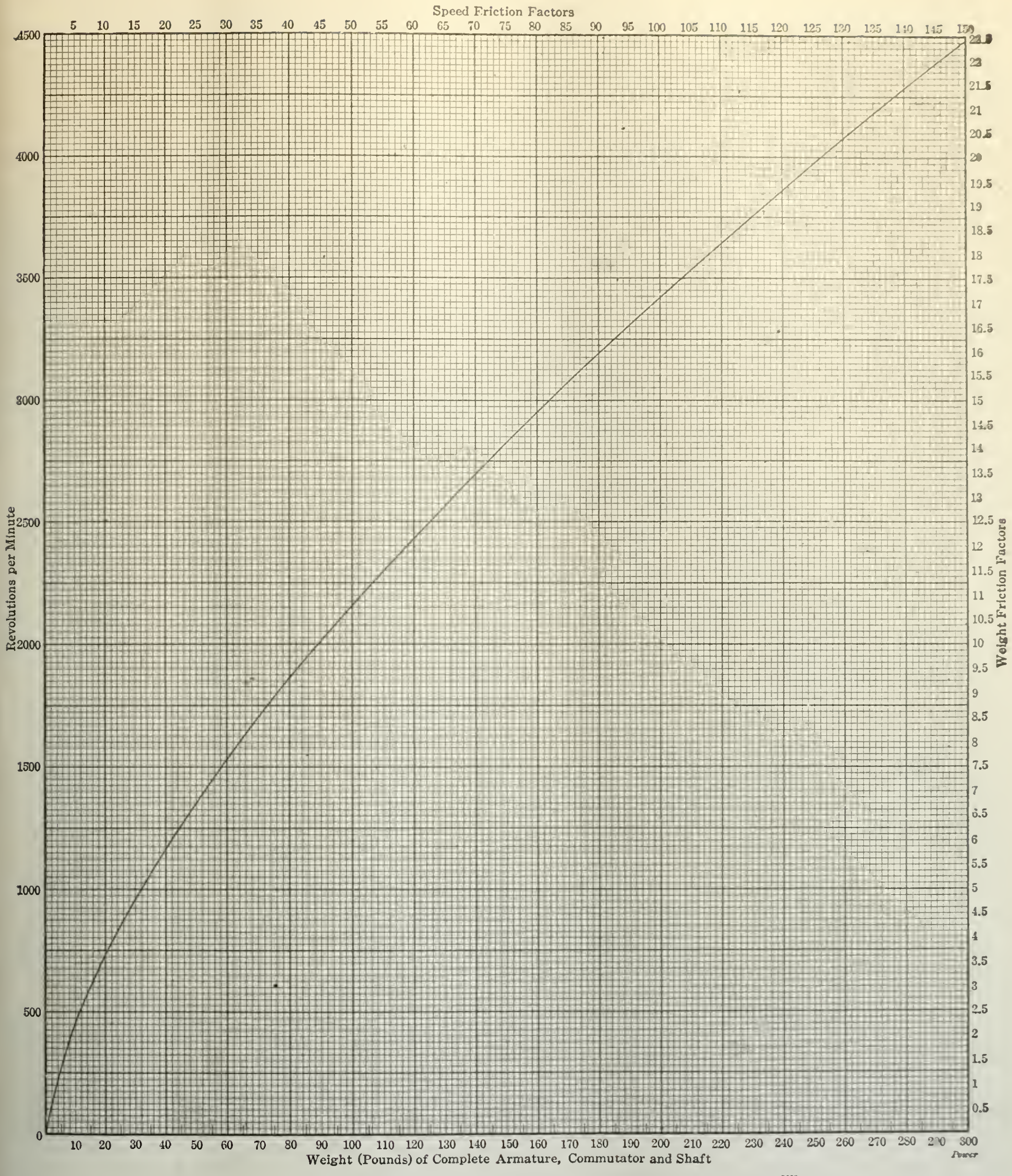


Fig. 59. Chart Giving Approximate Loss by Armature Bearing Friction and Windage

this lesson and the previous one, excepting the voltage drop at the brush contacts and the velocity of the commutator surface, relate to one square inch of contact area; thus, the contact resistance, the watts lost by that resistance, the brush pressure on the commutator and the watts lost by friction are all given *per square inch of contact area*. This means that the total watts lost by resistance are equal to the watts per square inch multiplied by the total area of all the brush faces on the machine, and the total loss by friction is, similarly, equal to the watts lost per square inch multiplied by the total area of all the brush faces.

Resistance varies in the opposite direction, and the total resistance of all brush faces, therefore, is equal to 4 × resistance per square inch ÷ total area of all brush faces, because one-half of the brush faces are in series with the other half. There is no need to bother with contact resistance, however, because it is only the voltage drop and the loss in watts that are of practical importance, and these can be estimated from the charts given in the preceding lesson.

In practical operation, the only feature that the dynamo attendant can control is the pressure of the brushes against the commutator, and he should understand very clearly, therefore, what is meant by pressure per square inch. It is the total spring pressure on one brush, divided by the area of the contact surface of that brush. It is seldom necessary to carry this pressure beyond two pounds per square inch of brush face and seldom practical to reduce it much below 1½ pounds per square inch. The practical worker, however, should understand all of the underlying principles of the apparatus in his charge, even though he may not control the application of them.

JOURNAL FRICTION

At the journal speeds commonly employed with dynamos and motors, the loss in the shaft bearings is proportional to the 1.5 power of the velocity of the bearing surface and to the pressure per square inch of bearing surface due to the weight of the armature, commutator and shaft. The loss due to windage is proportional to the cube of the velocity of the armature periphery. The proportionalities depend on the character of lubrication (for the bearings) and the character of the armature surface (for the windage). It is, therefore, impractical to attempt any workable formulas that would give accurate results for all cases. If the bearings are well lubricated and the length of bearing is not less than three times its diameter, and if the armature has not an unusually uneven surface or high velocity (beyond 3800 to 4000 feet per minute), the loss by bearing friction and armature windage will not be very far different from the result obtained from Figs. 58 and 59. These charts are plotted from a formula which is based on the assumption that the combined friction and windage loss is proportional to the cube of the square root of the speed and to the square of the cube root of the weight of the armature, commutator and shaft. This formula is as follows:

$$\frac{\sqrt{S^3}\,\sqrt[3]{W^2}}{4000} = \mathit{watts}$$

lost by bearing friction and armature windage. In this formula, S is the speed in revolutions per minute and W is the weight of the armature, commutator and shaft, complete.

The charts serve to give both the speed and weight factors of the friction loss. They are used as follows: Find the weight in the scale at the bottom of the chart and follow a straight line from that point up to the curve; from the point where the vertical line strikes the curve, trace horizontally to the right-hand vertical scale, where a number will be found, either directly or by reference to the numbers immediately below and above the point in the scale. This number is set down. Then find the speed in the left-hand vertical scale and follow a straight horizontal line from that point to the curve; from the point where you strike the curve, trace a vertical line up to the scale at the top of the chart; the number at the edge of the chart is the "speed factor," and this multiplied by the number first found gives the approximate loss in watts.

Example I. An armature weighing 22 pounds runs at 850 revolutions per minute. What is the approximate friction loss?

At the bottom of Fig. 58, find the number 22 and trace straight up to the curve; from the point at which the curve crosses the 22 line, trace over to the right-hand scale. The point at which the scale is reached corresponds to 3.925, being just one division above 3.9. Now find 850 in the scale of speeds at the left-hand edge of the chart, and trace that line until it strikes the curve; then trace up to the top scale, where the number 12.4 would be found if every scale division were marked. The weight factor was 3.925, and the product of 3.925 × 12.4 = 48.67 watts.

Example II. An armature weighing 144 pounds runs at 1125 revolutions per minute; what is the approximate friction loss?

Both the speed and the weight are beyond the range of Fig. 58, so the other chart must be used. Tracing from 144 in the bottom scale up to the curve and thence across to the right-hand scale gives 13.75 for the weight factor. Tracing from 1125 revolutions (in the left-hand scale) across to the curve and thence up to the top scale gives 19 for the speed factor. The loss, therefore, is about

$$13.75 \times 19 = 261.25 \text{ watts.}$$

Example III. An armature weighs 14 pounds and runs at 1800 revolutions per minute; about what is the friction loss?

In this case, the weight factor must be taken from Fig. 58 and the speed factor from Fig. 59. Tracing from 14 in the bottom scale of Fig. 58 up to the curve and thence over to the right-hand scale gives 2.9 for the weight factor, and tracing from 1800 in the left-hand scale of Fig. 59 over to the curve and thence up to the top scale gives 38 for the speed factor. The approximate loss, therefore, is

$$2.9 \times 38 = 110.2 \text{ watts.}$$

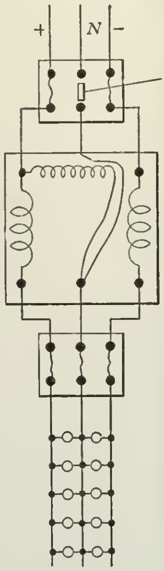
As the results obtained from Figs. 58 and 59 are only approximate, it is unnecessary to carry them out to decimal places. The result in the first example would be called 49 watts, that in the second example 261 watts and that in the last example 110 watts.

As the last example demonstrated, cases will frequently occur in which one chart must be used for the weight factor and the other one for the speed factor; for this reason the two charts have been plotted so that the values are interchangeable.

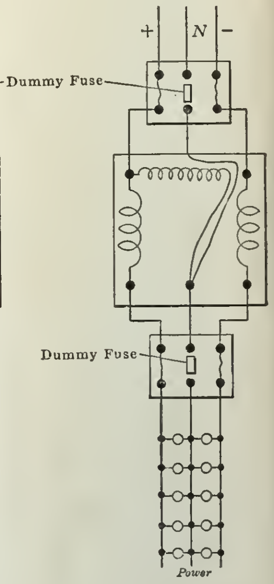
While the weights and speeds covered by the bottom and side scales of Fig. 58 are also included in the scales of Fig. 59, it is much better to use Fig. 58 for all the values within its range and to use Fig. 59 only for weights and speeds not contained in the scales of Fig. 58.

Cheating the Electric Meter

Referring to Mr. Howard's article on this subject in the June 14 number of POWER, I would like to point out that Mr. Howard is mistaken in thinking that the removal of the main neutral fuse, as in Fig. 3, will prevent the meter from registering. The potential coil of the meter will receive practically normal voltage, and the meter will therefore record about as accurately as if the fuse were not removed, because the coil remains in parallel with the lamps between the + and the neutral wires. In order to prevent the meter from working, both of the neutral fuses would have to be removed, as shown in the diagram sent with this letter.



MR. HOWARD'S FIG. 3



MR. PERCY'S DIAGRAM

P. C. PERCY.

San Francisco, Cal.

In Mr. Howard's article on "Cheating the Electric Meter" he says that the meter connected as in Fig. 3 will not register, and that he has "had several cases of this kind." I don't believe he investigated these "cases" very closely; if he had he would have found that current will flow in the potential winding of the meter and it will record normally, if the load is balanced, because the winding gets half of the total + to — voltage.

W. C. DRAKE.

Terrell, Texas.

Gas Power Department

A Train Load of Gas Engine

The accompanying photograph shows a train load consisting of a single gas-engine blowing unit, recently shipped to the Lake Superior Iron and Steel Corporation at Sault Ste. Marie, Canada. The gas engine is of the twin-tandem type with cylinders 34x48 inches, to work on the four-stroke cycle. It was built by the Allis-Chalmers Company and conforms to the company's standard practice in all respects. This is the third blowing unit that has been shipped to the Lake Superior Corporation, and four engines exactly like this one have been supplied for driving 2300-volt three-phase alternators, which supply current at 25 cycles to motors about the steel mills. Each of these units weighed about 1,000,000 pounds and required twelve flat cars for its transportation.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Surplus Power Available in Steel Works*

By Richard Bechtel

A blast-furnace plant having an output of 250 tons of pig iron per 24 hours requires about 10 tons of coke per hour. Of the quantity of heat put into the furnaces by this coke, 60 to 65 per cent. leaves the furnace again in the gas. About 28 per cent. of the heat, or 43 per cent. of the gas leaving the furnace, is used

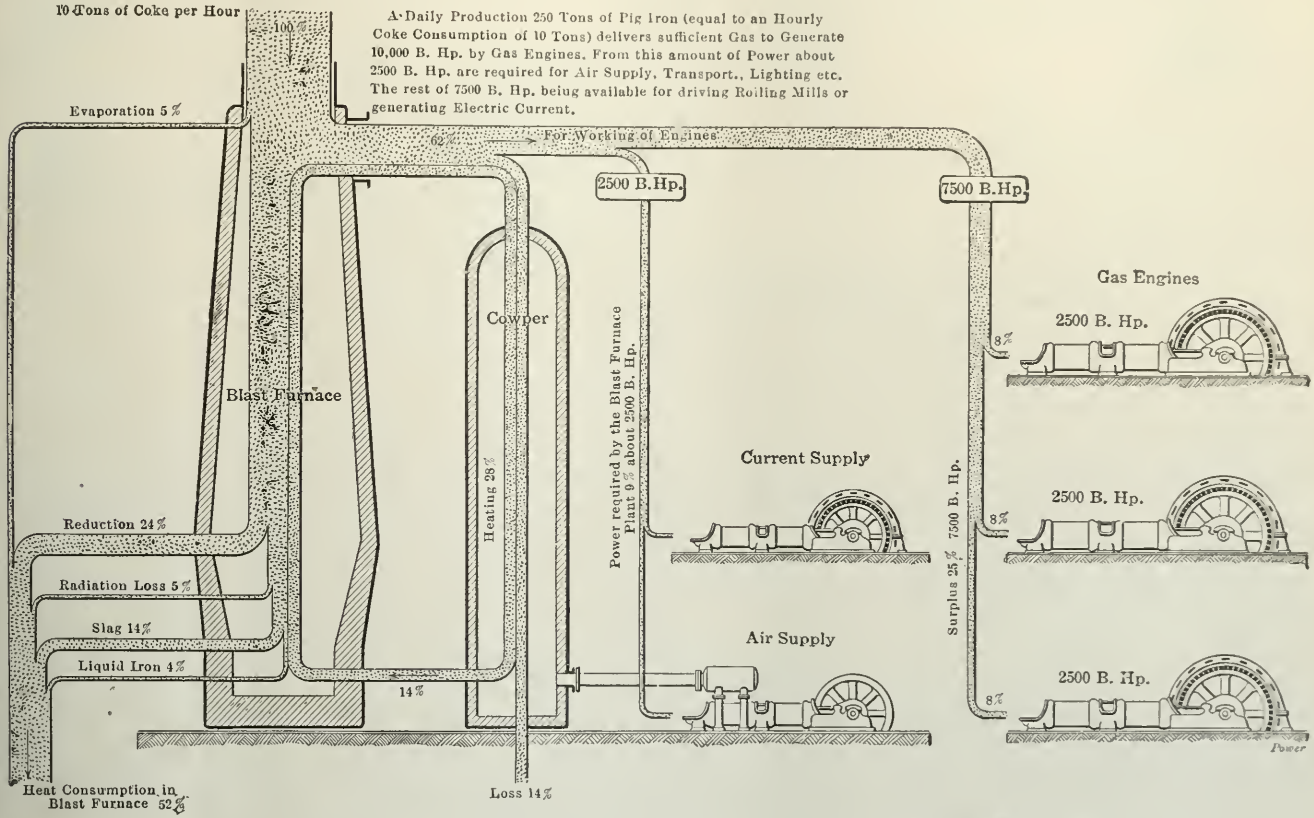
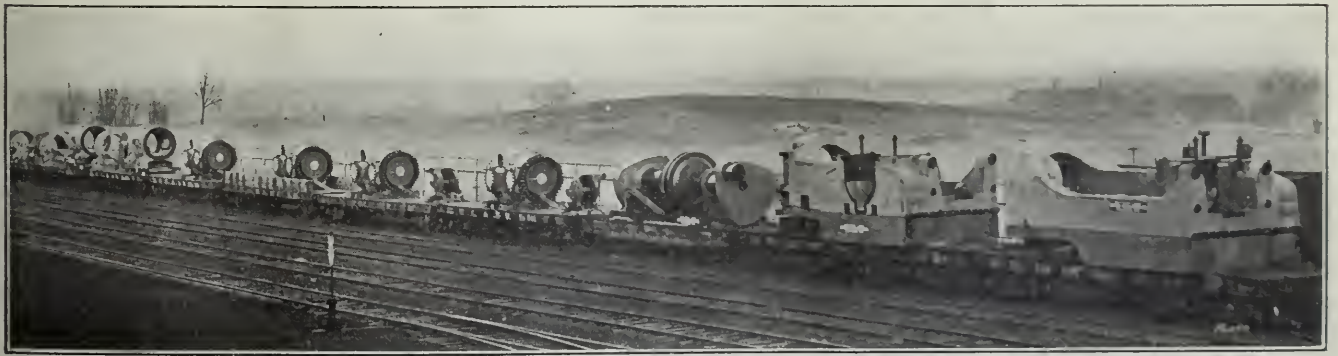

10 Tons of Coke per Hour
100%
A Daily Production 250 Tons of Pig Iron (equal to an Hourly Coke Consumption of 10 Tons) delivers sufficient Gas to Generate 10,000 B. Hp. by Gas Engines. From this amount of Power about 2500 B. Hp. are required for Air Supply, Transport., Lighting etc. The rest of 7500 B. Hp. being available for driving Rolling Mills or generating Electric Current.
Evaporation 5%
62%
For Working of Engines
2500 B.Hp.
7500 B.Hp.
Gas Engines
2500 B. Hp.
8%
Cowper
Blast Furnace
Power required by the Blast Furnace Plant 9% about 2500 B. Hp.
Heating 28%
Current Supply
2500 B. Hp.
8%
Surplus 25% 7500 B. Hp.
Reduction 24%
Radiation Loss 5%
Slag 14%
Air Supply
2500 B. Hp.
Liquid Iron 4%
14%
8%
Power
Heat Consumption in Blast Furnace 52%
Loss 14%

Fig. 1. Diagram of the Distribution of Heat in a 250-ton Blast-furnace Plant



A Train Load of Gas Engine

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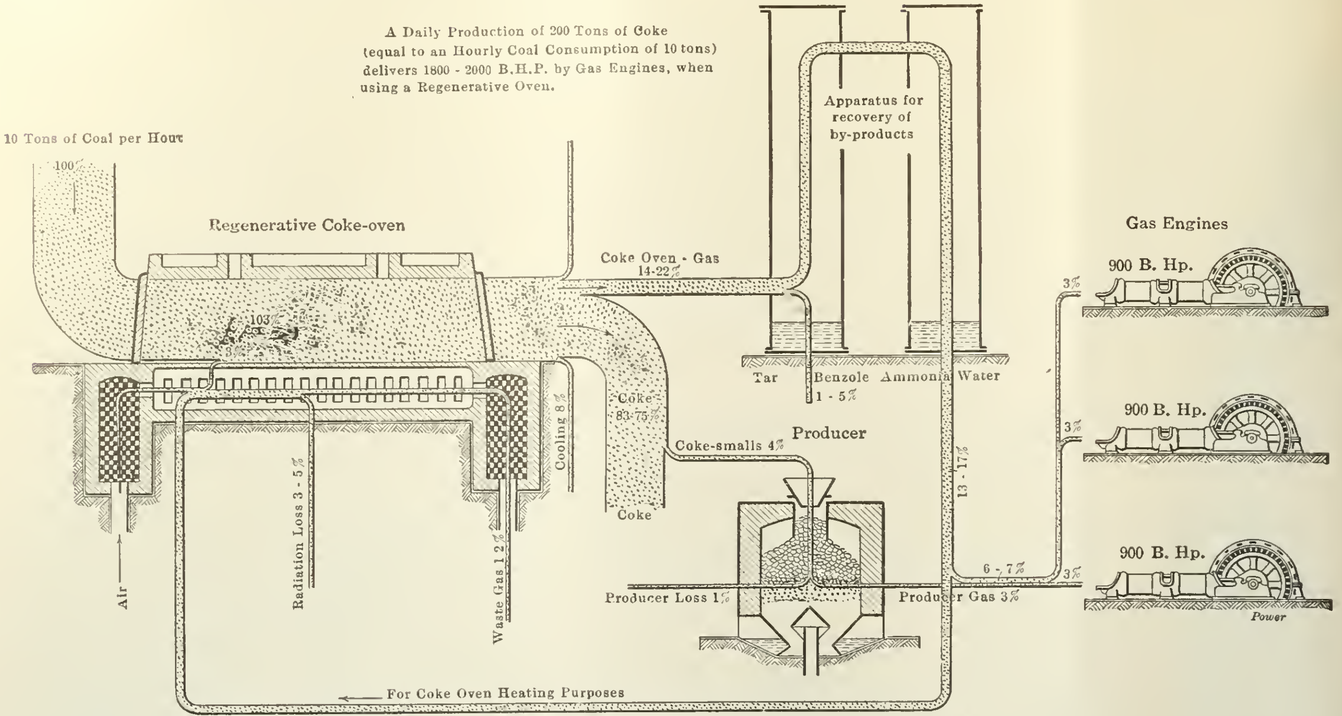
*Extract from a paper read before the Birmingham (England) Association of Mechanical Engineers.

for heating the blast in the stoves, and the rest is available for power purposes. Generally 8 to 10 per cent. of the heat, or 12 to 15 per cent. of the gas, is used for driving the gas-blowing engines and for generating electric current for the hoists, lighting and water supply of the furnaces. The power still remaining—about 24 per cent. of the heat, equal to not less than 7500 brake horsepower generated in gas engines—is available for any other purpose. Fig. 1 illustrates this distribution of heat energy.



By Utilization of the Waste Coke in the Nuremberg Producer especially designed for this Purpose, 900 B. Hp. additional is obtained, so that altogether 3000 B.Hp. is available continuously.

Fig. 2. Diagram of Heat Distribution in a 200-ton Coke-oven Plant

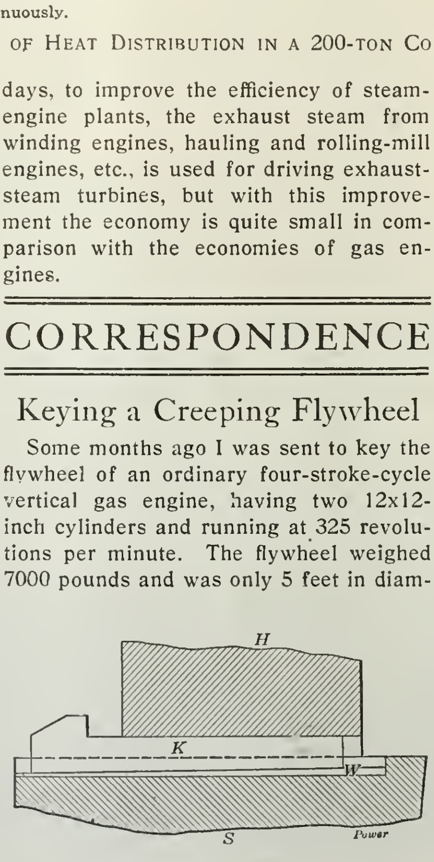
Similar conditions prevail in collieries having uptodate coke-oven plants with by-product recovery and surplus gas. Fig. 2 represents the heat distribution in a coke-oven plant of a production capacity of 200 tons of coke per 24 hours. This quantity is obtained by the consumption of 10 tons of coal per hour, which gives, after deducting all losses, a quantity of gas equal to 1800 to 2100 brake horsepower in gas engines. If the coke ashes are utilized in special producers a further 900 to 1000 brake horsepower can be obtained by gas engines, giving a total of 3000 brake horsepower.

If the gas from the furnaces is burned under boilers about 200 cubic feet of blast-furnace gas of, say, 100 B.t.u. per cubic foot is required per brake horsepower by first-class boilers and steam turbines so long as the plant is new, but for ordinary working conditions ample addition must be made. Professor Bonte states as the gas-consumption figure for gas-heated boilers in coal and iron works, according to reports from many uptodate and best managed works, 22.5 cubic feet to generate one pound of steam. In old boiler plants, with long piping and old-fashioned engines, the consumption is even three or four times higher. Nowadays, to improve the efficiency of steam-engine plants, the exhaust steam from winding engines, hauling and rolling-mill engines, etc., is used for driving exhaust-steam turbines, but with this improvement the economy is quite small in comparison with the economies of gas engines.

CORRESPONDENCE

Keying a Creeping Flywheel

Some months ago I was sent to key the flywheel of an ordinary four-stroke-cycle vertical gas engine, having two 12x12-inch cylinders and running at 325 revolutions per minute. The flywheel weighed 7000 pounds and was only 5 feet in diameter, on account of the high speed at which the engine ran, the weight being obtained by using a very heavy rim 18 inches wide. When I arrived at the plant the engineer had the wheel in place but by mistake had put it on the opposite way to that in which it had been running before. He had no jacks or block and tackle strong enough to lift the wheel, which had made it quite a task to get the wheel in place and promised to make it still harder to take it off again; to make matters worse, the engine was in a very crowded location. It was explained that the wheel had a tendency to shift endwise on the shaft while running in the old position. The hub of the flywheel was 12 inches long, with two keyways. On calipering the keys the faces were found to be practically parallel, as also were the keyways in the shaft and hub. This meant that the keyway in the hub would have to be tapered, but no facilities for making the proper taper were at hand.



Key and Wedge in Flywheel Hub

Although the wheel was on wrong according to marks on the hub, the keyways in the hub and shaft lined up perfectly; so I decided to try the scheme shown in the illustration. This shows a part section of the hub *H* and the shaft *S* on a line with the side of the key. We had a nearby blacksmith forge a wedge for each keyway, as indicated at *W;* this was made to fit the keyway along the sides, and was filed true at the top and bottom, using the flat side of the wheel rim as a surface plate. The thick end of the wedge butted against the end of the keyway in the shaft, and the key *K* was then fitted as shown. It has run for five months and seems to be a first-class job.

Frank E. Booth.

Toronto, Can.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Right Angled Drives

Two interesting examples of right-angled drives using belt transmission were seen recently by the writer at the plant of the Standard Box Company, Baton Rouge, La. The main shaft from the engine passes across the center of the machine room at right angles to the countershaft driving the machines, which are distributed throughout the length of the room. The two shafts clear by about 2 inches at the point where they cross.

Fig. 1 shows how the connection was made between the two shafts. There are only two pulleys in addition to the driving and driven pulleys, one of these being a transfer pulley and the other an idler, which being adjustable, serves also to tighten the belt. A study of the sketch will show that there is practically no tendency to stretch the edges of the belt or to cause friction due to the sliding of the belt on the face of the pulleys, as is the case with the usual quarter-turn drive when the load is heavy. The belt has only one-eighth turn at the two points where the driving strain is greatest, these points being on each side of the transfer pulley between the driving and driven pulleys. The only place where the belt has a quarter turn is on the slack side between the driving pulley and the idler, and as this is on long centers, there is no tendency to stretch the edges of the belt.

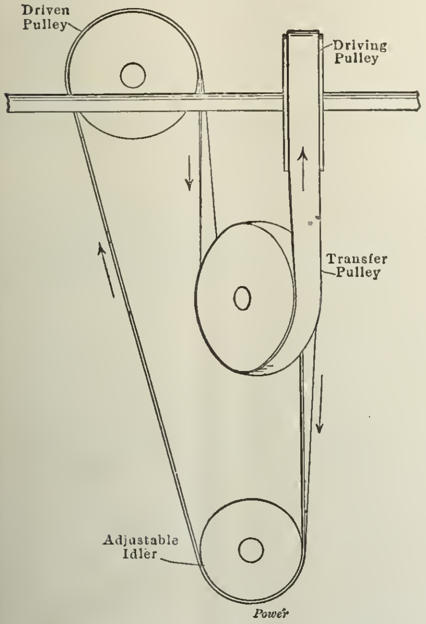


Fig. 1. Right-angled Drive with Main Shaft Close to Countershaft

One valuable feature of this drive is the fact that the belt turns over in passing from the driven pulley around the transfer pulley and onto the driving pulley, which allows of an equal wear on both sides. This drive has high efficiency, runs perfectly true, and is practically noiseless.

Another novel drive in this same factory is that shown in Fig. 2. In this case it was desired to drive a small machine with tight and loose pulleys, as it was used only occasionally for a few minutes at a time. On account of the limited space available it was found necessary to place it at some distance from the line shaft, and also to set it at right angles to this shaft. It was belted up as shown, *A* and *B* being the guide pulleys, the lower one of which is stationary, and the top one hinged and operated by a lever from the floor, which is not shown in the sketch. When set in the high position, the belt runs on the loose pulley and when thrown to the lower position, as shown by the dotted lines, the belt runs over onto the tight pulley. The belt runs perfectly true in either position. These drives were designed and installed by Mr. Walker, foreman of the above mentioned company.

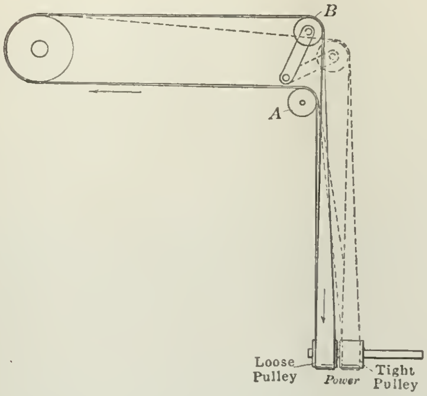
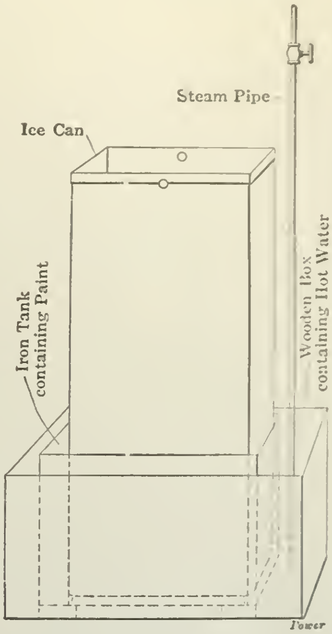


Fig. 2. Simple Means of Shifting Belt

New York City. S. Kirlin.

Repairing Ice Cans

As the ice season is now well advanced, and the time for trouble with leaky cans is at hand, I will describe my method of temporarily repairing them. I have made a small wooden box, inside of which is placed an iron tank that will hold the ice cans and extend 10 or 12 inches up the sides. The wooden box is filled with water after the iron tank is in place, and then the iron tank is filled about one-half full of paint. Steam is turned into the wooden box and the water and paint are raised in temperature to nearly 212 degrees. The ice can is then placed in the iron tank containing the hot paint and left there about ten minutes, or until the paint has had time to flow into all the small holes that may be in the can. It is then removed from the tank and the surplus paint is wiped off with a paint brush. The can is now turned upside down, so that the paint will not drain from the leaks, and is left in this position for about three hours, after which the can may be put into service again.



Scheme for Dipping Cans

The object of heating the paint in hot water is to prevent fire, also so that the repair can be made indoors. Heating the paint thins it so that it will find its way into the smallest holes or cracks. It is not necessary to paint more than 10 or 12 inches up the sides, as 99 per cent. of the leaks occur at the bottom.

I have repaired a great many cans in

this manner and seldom have them come back for repairs during the season.

W. H. Beisley.

Natchez, Miss.

Damper Regulators and Upright Water Tube Boilers

Will some of the readers of Power relate their experience with automatic damper regulators, stating the saving in coal bills when using the best types. I should also like to get some ideas as to homemade regulators.

A few pointers upon the care and operation of upright water-tube boilers would be appreciated, particularly as to the advisability of using asbestos-packed plug cocks for the blowoff on this type of boiler.

R. G. Parker.

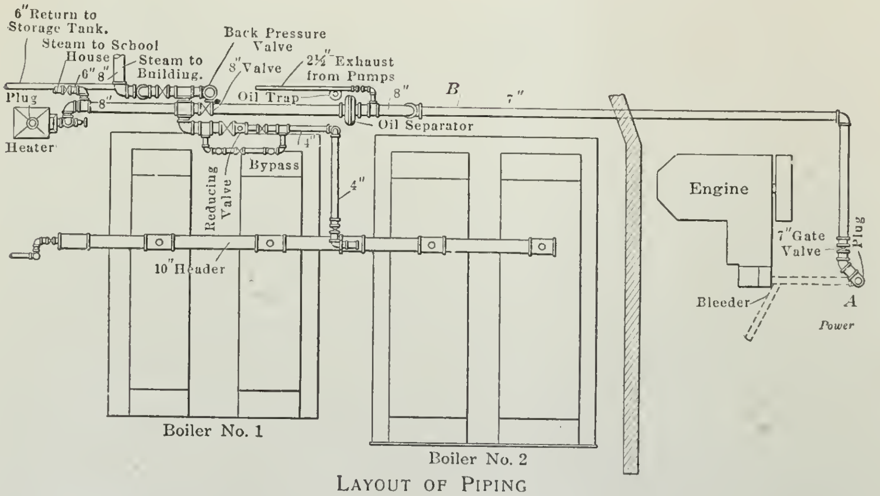
Detroit, Mich.

Piping Layout for Criticism

The accompanying diagram shows the layout of piping which is being installed in the plant in which I am employed. I should like the opinion of the readers of Power as to its embodiments of good design.

The point I wish to bring out is this: *Is it a safe system to use?* I claim that the back-pressure valve is located in a dangerous position, having been taught that a back-pressure valve assumes the same relation to an engine as a safety valve does to a boiler. If so, then this is not a safe plan, as there are two valves between the engine and the back-pressure valve.

Again, if the engine is shut down with from two to five pounds pressure on the system with the bleeder open on the exhaust pipe, steam is being lost all the time. and if the bleeder is closed the exhaust line will fill with water of condensation, a condition that will cause trouble when starting up. If one of the valves on the exhaust should be closed and not opened again before starting up, more trouble would result.



Layout of Piping

I believe it would be much safer to place the back-pressure valve at the point marked *A* and a check valve at *B*, which would then keep the back pressure from the engines when not running.

D. L. Temple.

Olathe, Kan.

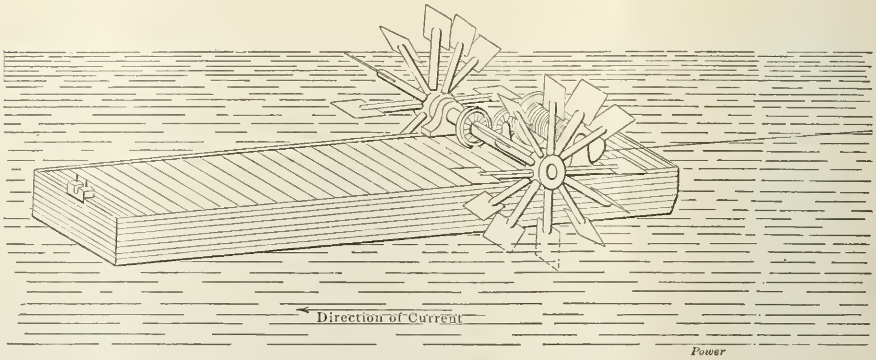
An Unique Application of Power

In a certain river in New England there is a high-water period in the spring when a fair sized ocean steamer could easily manœuver. At the place in question there is 22 feet fall per mile, which causes a rapid current.

Some years ago a good old Yankee wanted to transport some material to an island located in the center of the river at a point where the river was about two-thirds of a mile wide and the island was about seven-eighths of a mile up stream. He had no power except that which nature was producing, and by means of this and a wire cable about ¼ inch in diameter, one end of which was attached to a tree on the island, the other fastened to a reel on board the flat boat, which in turn was connected by gears to a shaft that carried paddlewheels on each side of the boat, he accomplished his task. After the boat was loaded, the paddlewheels were connected to the reel, which was slowly revolved by the action of the current and commenced to reel up the wire cable, at the same time drawing the boat and its cargo up to the island. The accompanying sketch illustrates the scheme used.

I have seen water put to many uses, but have never run across just this form of hydraulic power. It is easy enough to figure how a boat could be made to pull itself up stream by means of a wire and reel properly reduced by gears, but theory and practice do not always agree.

Brooklyn, N. Y. G. C. Abbe.



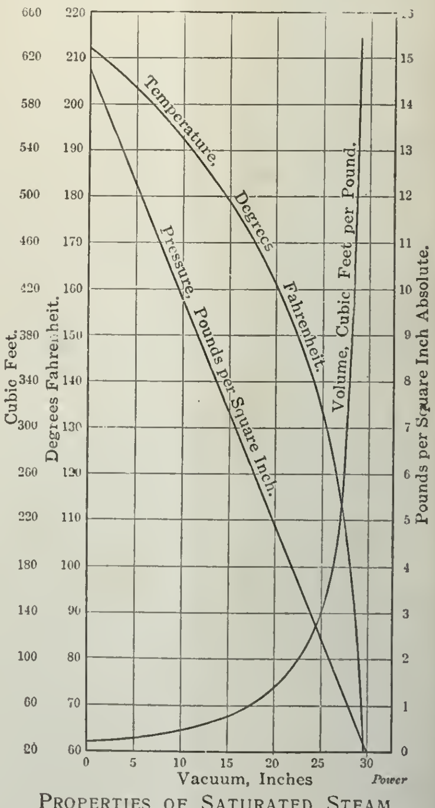
Boat Propelled by Action of Current

Some Properties of Saturated Steam

The accompanying set of curves shows the relation between vacuum, in inches of mercury, and volume, temperature and pressure for saturated steam. They should prove useful to all engineers who may have occasion to solve problems involving these quantities.

W. Vincent Treeby.

London, England.



Properties of Saturated Steam

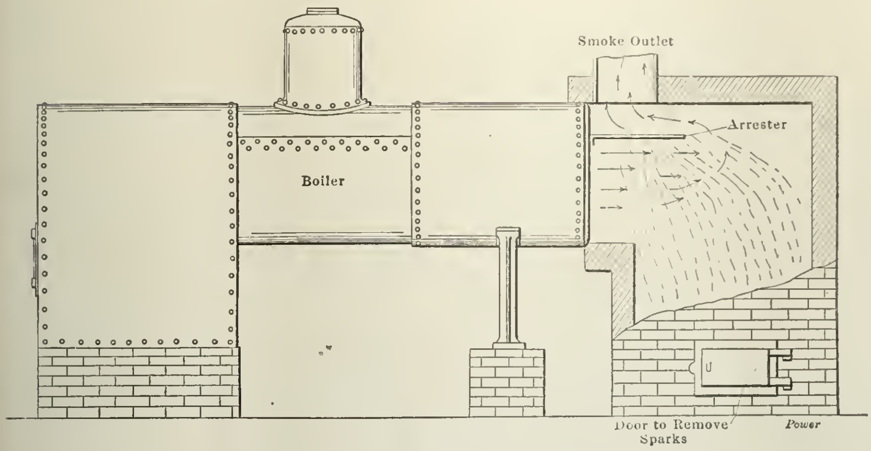
Spark Arrester

Some time ago the question was asked in the columns of Power as to the best means of preventing sparks from spreading when burning tanbark.

I have never burned tanbark, but have burned hay, straw, sawdust and shavings, in connection with which I have used several spark arresters now on the market, and have put screens in the smoke box and at the top of the stack, only to have them clog up and shut off the draft.

The accompanying sketch shows the means I finally employed and which proved to be the best I had ever used.

There was a 60-foot stack and before the arrester was installed the sparks usually blew nearly one-quarter mile to a neighbor's building. After the arrester was installed, however, I never saw a spark go more than twenty feet from the stack. The space between the arrester and the vertical wall should be greater than that between the arrester and the top of the smoke box.



Spark Arrester in Place

I. Sawyer.

Hartsdale, N. Y.

An Emergency Piston

Some time ago a bad knock was noticed in the first-stage cylinder of a two-stage air compressor. Upon investigation the piston was found to be badly cracked so as to necessitate another piston being installed.

The machine was needed badly and the repair shop was ransacked for another piston. The nearest that could be found was one an inch too small. This was upset on the edge and babbitt poured around it and then turned to the right size. No rings of any kind were used.

The machine was started inside of 24 hours and gave good satisfaction until a new piston could be obtained.

G. B. Longstreet.

Lowell, Mass.

Unusual Boiler Behavior

A strange condition presented itself in our boiler room about four years ago, which can now be told, as the boilers are still in first-class condition and the principals of the incident are working elsewhere.

It was on a Monday morning, and nothing unusual was observed before starting time, there being 80 pounds on the steam gages and the water was well up in the glasses. The load usually came on in the course of five minutes, and the two boilers handled it easily. Both furnaces contained a bright fire and the engines were started as usual, but the pressure dropped unusually fast and the water in No. 2 boiler fluctuated violently and evaporated so fast that it was necessary to keep the pump going at a high speed. On the contrary, the water in No. 1 boiler was very inactive, notwithstanding the fact that there was just as hot a fire under it. When looking at the fire I observed water sizzling from the girth seam over the bridgewall. Things looked alarming, so I gave the safety valve of the inactive boiler a gentle pull to see if it were in order; this immediately started action within the boiler, as indicated by the water in the gage glass fluctuating violently. The gage pressure now rose with surprising rapidity and we had to shut off the blowers to keep the safety valve from popping.

The following is my explanation of the incident: The firebox of No. 1 boiler had been repaired the Saturday and Sunday previous, No. 1 being cut out on Friday night and cooled down by changing the water. The other fire was allowed to go out on Saturday night, and there being no pressure on either boiler on Sunday, they were connected together.

At 5 a.m., Monday morning, the watchman fired up in an easy but dangerous way; he wheeled in about half the usual quantity of slab-wood and started a fire in No. 2 boiler, the one containing the hot water, and when the gages indicated about 40 pounds pressure, which was enough to operate his blowers, he transferred a few shovels of hot coal to the other firebox. In a short time he had a good fire in No. 1 boiler which contained cold water with a film of air on top of it, and 40 pounds pressure on top of that. The top of the boiler was cold and contracted, causing the girth seam to leak. The water was absorbing enormous quantities of heat, with pressure on top preventing ebullition and circulation, and I probably pulled the safety-valve chain at the right time, which started the boiler into action and gave up its accumulated energy to the engines.

Charles Haeusser.

Albany, N. Y.

Oil Fuel in the British Navy

At present the question of liquid fuel for the British navy is assuming undue prominence. The average man is an earnest student of the daily press, and it should be noted that as there is quite a boom in oil shares, there emanate from interested quarters pointed paragraphs and selected statistics tending to show that the authorities are remiss in their conception of the determining circumstances. If we review the facts, as they must appear to men whose mental attitudes are not obsessed by speculative issues, we may find that the Admiralty have not been lacking in decisive action.

In the first place we must note the admitted advantages of oil fuel over coal when used in steamships. Briefly these are:

1. Weight for weight, it occupies less space than coal.
2. When efficiently used it has about double the calorific value of ordinary coal, weight for weight.
3. Its use involves a considerable saving in stoking.
4. Saving in bunker capacity.
5. It is subject to easy control.

It does not require any very extensive study to grasp these facts, and it is quite reasonable to assume that during the time which the British Admiralty have been experimenting with liquid fuel, they have had ample opportunity of weighing the importance of these advantages. In 1902, after several years of tentative experimental work, preliminary trials fully established the claims of oil and much headway was made in determining the best type of furnace for its use. As a result the authorities came to the conclusion that ten tons of oil were about the equivalent of thirteen tons of best Welsh coal, but it must be noted that at the time they were paying fifty shillings (approximately $12.50) per ton for oil whereas they were able to obtain coal at from sixteen to seventeen shillings (approximately $4) per ton.

At the meeting of the "Royal Commission on Coal Supplies" the following question was put to Dr. Boverton Redwood, the home-office adviser on petroleum: "Do you think there is any likelihood of petroleum supplanting coal in the navy?" His reply is interesting as it contains the gist of the question of oil versus coal for British warships:

"I think that the likeliest form in which petroleum could be employed is as a source of power for the small engines that form part of the equipment of a modern warship where it would be used as an agent in an internal-combustion engine. I do not know where we are going to get adequate supplies for driving the main screws. If the existing output of petroleum in the world were doubled it would only be equivalent to about 5 per cent. of the present coal consumption."

The following table gives the world's production of coal and oil for the years 1901, 1905 and 1909.

	Coal Production, Million Tons.	Oil-field Production, Million Tons.
1901	798	22
1905	940	28½
1909	1100	40

Even on the basis that oil fuel possesses twice the calorific value of coal we find that in 1909 its equivalent weight only amounted to 7.2 per cent. There are many other considerations to be taken into account in addition to that of calorific value. Principal among these, for the British people, is the fact that oil fuel is almost entirely a foreign product, and the following table gives the estimated production of the leading oil-producing countries in 1901 and 1907:

	Per Cent. 1901.	Per Cent. 1909.
United States............	42.25	64
Russia	51.0	22
Galicia	2.05	4.5
Roumania	88.0	3.0
Mexico	....	1.25
British Empire..........	3.15	5.25

Of the 5.25 per cent. credited to the British Empire in 1909 it must be noted that India claims 2 per cent., Borneo about 3 per cent., Canada about 0.20 per cent., Scotland and other places about 0.05 per cent. Unquestionably there are other great potential oil resources in other parts of the British Empire, notably in Egypt and India, and it might appear quite a simple matter to provide oil stations or bases for the fleet in the same manner as is already done in the case of coal, but many reasons dictate that the source of power of the fleet should be placed outside the region of speculative issues.

R. H. Coulson.

Wigan, England.

Hints to Beginners

In handling boilers, as well as engines, we will do well to keep in mind the following: First, safety; second, continuous operation; third, economy. *Paste that in your hat.* It has helped many an engineer to keep on the right track. Many engineers do their own firing and cleaning, which requires some Sunday work, and they naturally do not care to work overtime any longer than is necessary.

Assume you are to wash out a boiler tomorrow, which we will say is Sunday. First, prepare for the work today, as every tool and all supplies looked up today will shorten the time required tomorrow. Be sure you have manhole- and handhole-plate gaskets of the right size, and, if of rubber, see that they are not rotten. Have an extra handhole plate also a crab or yoke. See that the hose is in good condition and have a packing for it as it will take time to cut one tomorrow. Have a sharp cold chisel, a hammer and a 2-foot monkey wrench, also a piece of 2-inch shafting long enough to reach from the floor of the combustion chamber to the nut on the rear handhole-plate bolt. See that all scaling tools are in good condition. Lay out the work as far as possible. If the check valve or globe valve needs repairs make a mental note of it. Burn the fire low and have the ashes off the grate soon after closing down. Leave the damper and ashpit doors open, which will allow air to be drawn in and evenly cool the grates, setting and boiler. Do not make the mistake of opening the smoke-box doors or the door in the rear of the setting or you will have a hot boiler to enter in the morning.

If a watchman is employed, have him open the blowoff valves about 6 a.m. Be on the job soon after the water is out to wash out mud before it is baked on the tubes and shell. Do not knock in the manhole plate till you are sure no steam is in the boiler, it takes but very little to burn you. Having done this, remove the front handhole plate, first being sure the water is all out of the boiler, for although the blowoff valve may be open the pipe may be stopped with scale.

In removing the rear handhole plate the nut can sometimes be turned off with a 2-foot wrench, but if it does not start easily, split it off with a chisel, first placing the above mentioned piece of shafting under it so as to have something solid to pound on. Now take out any loose pieces of scale that might be washed into the blowoff valve. Never carry an exposed light into a boiler until it has had a chance to air out, especially if there is any possibility of sewer gas having collected in it. After taking out the manhole and handhole plates close the smoke-box doors and with the damper open, a circulation will be started down the manhole and out the handhole which will clear the boiler of foul air. Wash out the boiler with water under a good pressure before entering.

Carry a light hammer and something to scrape with when entering the boiler and look along the seams, get up to the front head and sound the braces; if one is loose it will rattle when hit. Inspect the rivets and look down between the tubes for scale. Take a look at the feed pipe where it comes through the head; it may be nearly broken off. Assuming that this is a top feed entering the front head, see that it is securely fastened; if it can be wriggled around, it will be time break off at the threads where it enters the bushing in the head. Examine the rear head, tubes, braces and safety plug; scrape this off so that water can come in contact with the soft metal. If you see anything suspicious looking hit it, do not be afraid of breaking the boiler, if you can break it with a hammer it is time it went to the scrap pile.

After getting out of the boiler, take a look at the front and rear ends, the tubes and especially the blowoff pipe. Next, get into the furnace and note the condition of the walls and arch, look for fire cracks between the rivets and the edge of the plate on the circular seam; if you have a water arch over the fire door look particularly at the bottom castings. Note the surface of the grate and examine the connecting bars, cotter pins, etc., proper inspection of the grate will frequently save a shutdown.

You are now ready to close the boiler. Scrape all the old packing from the handhole plates and inside the boiler around the handhole; replace the plates and draw them up tight; turn on the water and in about five minutes look for leaks around the gaskets. If one leaks, draw off the water and remove the plate; you will usually find that some of the old packing had not been scraped off.

You can now feel reasonably sure that your job is well done; many troubles may arise that I have not mentioned, but when you meet one, sit down for a few minutes and plan how to overcome it.

C. C. Harris.

Springfield, Mass.

The following figures relating to the relative cost and efficiency of coal and of oil are current in California: Two and one-half barrels of oil are the equal of one ton of coal in thermal units. In other words, the same amount of heat can be obtained from 2½ barrels of oil as can be obtained from one ton of coal. But the difference in price is very great. Coal, producing the same amount of heat per ton as 2½ barrels of oil, costs in California anywhere from $6 to $8 per ton, wholesale. Two and one-half barrels of oil, figured at the market-delivery price of $1 per barrel, costs $2.50—a saving of from $3.50 to $5.50 on every ton of coal displaced by oil.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Fan Formulas

In the remote issue of July 27, 1909, on page 154, appears an article entitled "Power Required by Fans," in which are given by William Sangster two formulas to calculate approximately the horsepower required by a fan used to exhaust hot air from brick kilns. The formulas are as follows:

(1) *Multiply the diameter of the wheel in feet by itself five times and by the number of revolutions per minute three times. Divide this result by* 10,000,000,000 *and the answer will give the horsepower required to drive a fan handling air at* 50 *degrees Fahrenheit.*

(2) *Divide* 511 *by the sum of* 460 *added to the temperature of the air in degrees Fahrenheit and multiply the horsepower found by rule* 1 *by this quotient. The result will give the horsepower required by a fan handling hot air.*

This last formula clearly shows that the horsepower required to drive the fan varies *inversely* as the absolute temperature of the air.

There is another formula given in "Steam Boiler Construction," by W. S. Hutton, on page 84. It is as follows:

Let

$P =$ Pressure of air delivered by fan in pounds per square foot,
$V =$ Volume of air at 32 degrees Fahrenheit, in cubic feet, used per pound of coal,
$W =$ Weight of fuel in pounds burned per square foot of grate per minute,
$A =$ Area of the grate in square feet,
$T =$ Absolute temperature of air entering fan in degrees Fahrenheit,
$C =$ Coefficient of the efficiency of the fan, which varies in practice from 0.2 to 0.5.

Then, the indicated horsepower required to drive a fan equals,

$$\frac{P \times V \times W \times A \times T}{33{,}000 \times (461° + 32°) \times C}$$

It is true that this latter formula assumes data quite different from those which the former does. But into both the item of absolute temperature enters and has been taken as a determining factor.

It should be noted, however, that the formula given by Hutton makes it quite clear that the horsepower required to drive a fan varies *directly* as the absolute temperature of the air.

Thus it will be seen that the formulas are contradictory one to the other, or, at least, they seem so to be.

As air or gas is heated it becomes lighter in weight and larger in volume. It might be reasoned, therefore, that to handle a comparatively light gas at a higher velocity, a smaller amount of power would be required. But, on the other hand, it is to be observed that as the air is hotter it is greater in volume, and as the sectional area of the air passages remains the same, the column of air becomes longer and, thus, the space through which work is done increases.

Manilal K. Desai.

Ahmedabad, India.

Heat Value of Coal

In the very remote issue of Power of July 14, 1908, on page 50, appears a contribution on the "Heat of Fuels, Etc.," by Prof. William D. Ennis. After elaborately discussing the process of combustion and the losses incurred in heating and evaporating the elements in the fuel and air, he comes on to the "Chemical Composition Determining its Heat Value." Here he says that the one pound of fuel contains 4 per cent. of hydrogen and 3 per cent. of oxygen, the rest, 93 per cent., being carbon. Of the 0.04 pound of hydrogen,

$$\frac{0.03}{8} = 0.00375$$

pound (one-eighth part of the oxygen in the fuel) is combined with oxygen and is driven off as water at moderate temperatures, and is, therefore, unavailable for combustion. The remaining

$$0.04 - 0.00375 = 0.03625$$

pound of hydrogen produces

$$61{,}800 \times 0.03625 = 2230 \text{ B.t.u.}$$

These heat units produced by the hydrogen uncombined with oxygen, added to the heat units produced by carbon, viz.,

$$0.93 \times 14{,}500 = 13{,}500,$$

amount to 15,730. Here, the forty-fourth and forty-fifth lines of the last column on page 51 are susceptible of correction. They read thus, "B.t.u., leaving 15,357 B.t.u. as the net heat of the coal 15,730 B.t.u. from which should." With the correction they should read thus, "B.t.u., leaving as the net heat of the coal 15,730 B.t.u., from which should." What I mean to say is that by taking only the uncombined hydrogen and carbon as the fuel, we have to deduct a quantity of heat which is equal to the difference between the heat produced by the whole hydrogen,

$$0.04 \times 61{,}800 = 2472$$

and the heat produced by the uncombined hydrogen

$$0.03625 \times 61{,}800 = 2230,$$

that is,

$$2472 - 2230 = 242$$

B.t.u. Deducting this from the total heat

$$13{,}500 + 2472 = 15{,}972$$

B.t.u., we get as the net heat of the fuel

$$15{,}972 - 242 = 15{,}730$$

B.t.u., and not 15,357 B.t.u. Of course, this may be a misprint or perhaps a misunderstanding and it is not very important.

Going down further, the author deducts the heat units which are necessary to heat and evaporate this uncombined free hydrogen, thus,

$$9\,(180 + 966) \times 0.03625 = 1146 \times 9 \times \left(H - \frac{O}{8}\right) = 373$$

B.t.u. Thus, the net heat value is,

$$15{,}730 - 373 = 15{,}357$$

B.t.u. So far, all is right. But in the formula, this last item, 373 B.t.u., is not deducted, as it should have been. The formula, as given, stands thus:

$$14{,}500\,C + 61{,}800\left(H - \frac{O}{8}\right)$$

According to this, the B.t.u. in the one pound of fuel are

$$14{,}500 \times 0.93 + 61{,}800\left(0.04 - \frac{0.03}{8}\right) = 13{,}500 + 61{,}800 \times 0.03625 = 13{,}500 + 2{,}230 = 15{,}730$$

while the heat units in the one pound of fuel, as calculated above, are 15,357. I think, therefore, that the formula needs correction, thus:

$$14{,}500\,C + 61{,}800\left(H - \frac{O}{8}\right) - 10{,}314\left(H - \frac{O}{8}\right)$$

According to this, the one pound of fuel has

$$14{,}500 \times 0.93 + (61{,}800 \times 0.03625) - (10{,}314 \times 0.03625) = 13{,}500 + 2230 - 373 = 15{,}357$$

B.t.u., which is correct.

The above formula, as corrected, can be expressed thus:

$$14{,}500\,C + 61{,}800\left(H - \frac{O}{8}\right) - 10{,}314\left(H - \frac{O}{8}\right) = 14{,}500\,C +$$

$$\left\{(61{,}800 - 10{,}314)\left(H - \frac{O}{8}\right)\right\} = 14{,}500\,C + 51{,}486\left(H - \frac{O}{8}\right)$$

With all due respect for the learned professor and others concerned, I draw their attention to this, and if I am not wrong, I ask for the adoption of the formula, as corrected above. If I err, will someone kindly explain where and how?

MANILAL K. DESAI.

Ahmedabad, India.

Lead Gaskets

Mr. Binn's article on manhead gaskets in the May 24 issue recalls my experience with the manheads as sent out with the Atlas boilers. I did the same as he but instead of using the lead alone I found, after filling and using it once or twice, thereby making a groove or channel, the exact thickness of the metal around the manhole and by using a layer or strand of asbestos wicking properly graphited and placed in the groove, made an excellent job. The wicking can be quickly and cheaply renewed and preserves the lead, its purpose being to prevent the metal from cutting the channel any deeper. When cleaning the boilers every week the lead would last from 8 to 12 months while carrying 120 pounds pressure and give entire satisfaction.

JOSEPH STEWART.

Hamilton, O.

Installing Globe Valves

This question is deserving of a large amount of consideration because dependent upon it are the lives of engineers, boiler attendants and other persons. The fallacy of connecting valves to a boiler with the pressure beneath the disk is frequently proved. The threads upon the valve stem are the only resistance against the load, which is obviously equal to the area of the valve multiplied by the pressure per square inch; consequently the threads on a 6-inch valve, where one hundred pounds pressure is carried, must withstand a load of about 2800 pounds. The threads soon become worn out from constant use and leave insufficient thread contact.

The principal argument against connecting valves with the pressure above the disk is that in this way it is difficult to pack them. That is very true, but if valves are packed regularly when the boiler is shut down and not let go until they begin to leak, this difficulty would not take place.

Another very important feature which should not be overlooked is that of placing intermediate valves in the lines. This affords an easy access for repairs which otherwise would be neglected. The question is of vital importance, especially where changes are to be made and the steamfitter takes the job on contract. Unless there are specifications for him to follow in regard to proper piping, it is probable that he will make it as profitable as possible regardless of safety. If the supervision of this class of work were left to a competent engineer, he would undoubtedly see that the job was done properly, for his own safety as well as that of others; instead the responsibility is often left to those who introduce such unreliable methods.

The writer recalls an instance in which there were no intermediate valves in the steam lines, and one morning just before starting time, the main stop valves were being opened on the boilers when one of the engines unexpectedly started up cold and came to a sudden stop on the center. This was due to the pressure being beneath the disk of the throttle valve and a water hammer had driven the disk upon the valve stem. However, after the fireman had closed the main stop valve which had just been opened, the engine was moved off center, and the main stop valve was opened gradually until the engine was up to speed. After the day's run the stop valves were closed and a new disk put in place. There was no drip pipe at this engine to relieve the steam pipe of condensation above the throttle, consequently the only means of doing so was by way of the lubricator.

The whistle was an old-time affair and was used very frequently during the day. In this case a globe valve was used with the pressure acting below the disk and from constant use it gave way early one afternoon and the whistle continued to blow the remaining part of the day. There was no other valve in the line and steam could not be shut off until night when the pressure was dropped on the boiler.

The writer could enumerate many more inconveniences of this kind but it appears that people will not be brought to their proper senses until the penalty is paid.

JAMES G. SHERIDAN.

New York City.

Loose Crank Pin

Referring to P. C. Forgard's contribution in the June 7 issue under the above heading, I suggest the following as the cause of the trouble, the pin is out of line.

As a remedy, I suggest a new pin.

H. B. BRAND.

Brooklyn, N. Y.

Brains vs. Temper

In the May 3 number of POWER there appeared an article entitled "Brains vs. Temper." I have looked each week for some comment on this article, but thus far have seen nothing. It seems to me that in the Commonwealth of Massachusetts it is remarkable that an engineer (presumedly holding a first-class license) should be found who would connect up the drips of a pump as stated, and still more remarkable that an expert (?) from the manufacturers of the pump should work all day without discovering what was amiss. Really, I think the first part of the title is superfluous, there being nothing in evidence except "Temper."

X. Y. EGAN.

Bennington, N. H.

Boring a Pump by Hand

While looking over the practical letters in POWER for June 14, I became very much interested in the article by F. C. Holly regarding the method of boring a pump cylinder by hand power. While I do not want to detract from Brother Holly's credit, owing to a similar experience of my own, I think I can add a few suggestions in the way of improvement. In the first place, in order to have the cylinder bored true, it would be better to have the guide *A* bolted to the face of the end of the cylinder, using two of the cylinder bolts for the purpose. The guide or strap *A* having two oblong holes to faciliate centering the shaft. After centering, these two bolts or nuts should be screwed up as tightly as possible. By so doing all possibility of springing the shaft while turning is eliminated. Regarding the feed, I don't quite understand the diagram as the feed screw is on the steam end. This being the case, and as the cut was started from the outer end of the cylinder, the feed screw would have to pull the cutter along thus creating quite a strain on the angle iron *B*, a tendency to pull or spring it out of position and a tendency to bind the thread.

Another little suggestion is that two men will be required to do the job, one to turn the shaft and another to attend to the feed screw. Anyone who has had lathe experience probably is aware that when turning up a piece of work where there is the least possible tendency to spring if the feed is not continuous and the tool turns around twice in the same place there will surely be a groove dug and as no two men can work absolutely together we adopted another and I think a better way. In place of the angle iron *B*, shown in Brother Holly's sketch, we put another strap across the face of the end of the steam cylinder and centered the rod or shaft at that end. A thread 24 to the inch was tapped in the center of the strap on the water end and a thread cut correspondingly on the shaft, back of the cutter, long enough to take the whole length of the cylinder. By this method the feed was made automatic, as when the shaft was turned around the feed worked correspondingly. An old throttle wheel was used in place of the handle shown in Brother Holly's diagram thus giving a better grip with both hands and a more steady pull and cleaner cut. Two cutters were used, one back of the other, the last one being a broad-nosed tool. The cylinder in question was 8x12 inches in size and brass lined. The job was done by one man without removing the pump from its original position.

CHARLES H. TAYLOR.

Bridgeport, Conn.

Clutch Pulley Trouble

In the June 7 issue of POWER, F. E. Haught asks for information about a clutch pulley.

If his line shaft is not too small, a quill could be installed with a saving in the matter of belting, pulleys, etc., and space taken up by a countershaft.

In the power house in which I am employed, we have a quill running on a 6-inch shaft and driving a compressor that was installed last August. We have had no trouble whatever with the installation.

WILLIAM BEATON.

Gold Roads, Ariz.

Shrinkage Fits

In the issue of June 14 appeared an article on putting in new crank disks or pins. It has always been a question in my mind whether it is safe practice to apply heat. Considering the entirely unknown shrinkage effects, which no one can figure as there is so much to consider and know, great caution should be exercised when subjecting the disk to the strain of cooling off. It is the usual practice of some engineering firms to allow 0.0001 inch per each inch of pin or shaft diameter. But usually the pins are pushed in cold except in the case of a built-up marine crank which is always made of steel. In such a case heating can hardly do any damage. But with the usual cast-iron disk, I think that every time shrinkage is resorted to a chance is taken of splitting the disk.

In making repairs it takes but little to cause an accident that would disable an engine until another disk could be cast or secured from the engine builders. In either event the delay would probably be long and might prove costly. I had occasion once to put an 8-inch crank pin into the low-pressure disk of a cross-compound engine which drove a 1010-kilowatt generator and, although not believing in shrinkage jobs, to save time it was suggested that the disk be slightly heated. But the builders would not allow any heat whatever although they were ready and willing to assume responsibility for any damage resultant from pressing in a pin which was 0.0001 inch to the inch large.

J. W. HAMILTON.

Redondo Beach, Cal.

Drop in Steam Pressure

In the June 21 issue, Mr. Satterfield asks for enlightenment as to what a reasonable drop from boiler pressure to the maximum pressure in a Corliss-engine installation should be. While 16 pounds is too much for good practice it is by no means uncommon. The engine in his case may have insufficient area in the steam ports or an excessive steam lap.

A number of diagrams in the writer's possession taken from 40 Corliss engines and six automatic four-valve engines show an average of 10.6 pounds drop in pressure in the cylinder. These engines were running under all sorts of conditions as to load, etc. With few exceptions the diagrams were not taken under test conditions, the steam-pressure readings being taken from the boiler gages in the various plants in which the engines were located.

The drop in pressure ranges from 0 to 30 pounds, the diagrams showing a 30-pound drop having been taken from an engine located over 100 feet from boilers.

Out of the 46 engines only one, a slow-speed Corliss pumping engine with 125 pounds steam pressure, showed no drop in pressure; one engine showed a drop of one pound. Three of the engines have a drop of 21, 24 and 30 pounds respectively. As this must be due to unusual and abnormal conditions a fairer average of 9.37 pounds drop is obtained by eliminating them from the calculation.

F. C. HOLLY.

Yazoo City, Miss.

Concerning the Superintendent

I was much interested in Mr. Stewart's letter in the June 21 issue of POWER, for I have had the same experience myself.

It is very discouraging to an engineer to have a man by virtue of his authority completely overrule all sensible ideas.

What does an engineer think of his superior when the latter advances theories that are at once foolhardy and ridiculous? A man in such a position should compel the respect of those under him by his treatment of them and by making them think that he is in control by virtue of his knowledge. I do not mean that he must know everything, for that is impossible. But he can approach a subject in such a manner that the man in the engine room will feel that his advice is being followed and thus have more interest and confidence in his position.

I know of a plant where every little contemplated change is put up to a number of technical students and after more or less consideration some of them will saunter down to the engineer, find out his idea and then walk away. In a few days the change will be made and it is so much like the one proposed by the engineer that one would think that he should get credit for it. But more often one of the students is "patted on the back."

This also brings to our attention the editorial, in the same number, on consulting the engineer. Cases similar to the one mentioned are so common that we are getting used to seeing them. There is, for instance, a large power house being built for a manufacturing plant. All the machinery is being laid out by a man with a pull, who is just about as capable as a two-year-old child for the work.

The engines were first erected in the open and a temporary wooden structure put over them. The roof of this was made so flat that the first rain left a small pond on top which kept the generator well sprinkled for several days after the storm. Some temporary foundations were built in which some of the bolts were about 3 inches out of line with the holes in the machine and, of course, this was not noticed until the concrete had set. The offending bolts were cut off.

When cold weather came, regardless of the warning of the engineer, there were no precautions taken to prevent freezing. As a result the plant did not get started in time, for most of the piping was frozen. Then, there was a great hurry and steam was piped about a hundred yards from another plant; holes were stopped up and by using the greatest care the plant was kept in operation. These are only a few of the blunders that were made and there will be many more before the work is finished. Any of these could have been prevented if the engineer had been consulted and the money saved would have paid his salary for several years.

G. H. KIMBALL.

Southbridge, Mass.

Low Pressure Turbine

I read with interest the editorial in POWER for June 28 on the suggestion of using the low-pressure turbine as a second cylinder in connection with a single-cylinder reciprocating engine.

In 1906 a low-pressure turbine was installed as a third cylinder of a cross-compound engine in a large manufacturing plant. The original equipment consisted of two large cross-compound engines, located in two power houses. One engine furnished power through shafting to the various machines in one portion of the plant and also operated a small three-phase alternating-current generator. The other and larger engine operated a three-phase direct-connected generator. Improvements were to be made and the entire shops were to be equipped with induction motors. There was a shortage of power while the new power house was being built and while the old engine was being moved. The direct-connected unit being too small for the extra load, a low-pressure turbine was erected just outside the engine room and connected to a receiver, which was connected to the exhaust pipe of the engine.

This installation was made without any interruption of service, the exhaust pipe being connected at night.

When all was complete, the engine was started up. The average receiver pressure between the low-pressure cylinder and the turbine is 10 pounds. The turbine starts to move with from 3 to 6 pounds pressure in the receiver. When it is up to speed and voltage, the generator is connected in parallel with the one on the engine, and runs as long as the engine does with practically no attention. The turbine has no regulator, it simply takes the steam from the low-pressure cylinder. Tests and the general running of this installation proved so satisfactory that the company decided to use a second low-pressure turbine in connection with the other compound engine. This will greatly increase the capacity of the power plant without requiring much additional space for the turbine.

R. A. Cultra.

Boston, Mass.

On Engineers and Wages

From time to time there appears in the columns of Power discussions of the question of payment for services rendered by the steam engineer in charge of a plant, as compared with the payment for services rendered by other craftsmen, and I have thought a few remarks from one who has had an experience that seldom falls to the lot of man, might be acceptable even though they do not solve this very complex question.

A graduate from the machine shop and engine room rather than from the technical college, one who is both employer and employee, who at times scratches the bald spot on a very poor man's head, and at other times sails his own yacht "over the bounding billows," and who at all times is a believer in the square deal, I think I am without bias, and only regret that in discussing this question I may have to tell of my own experience more than I would wish to, but hope that the readers will excuse my seeming egotism if any appears.

After several years of experience in marine shops in New York, I obtained my first position of real responsibility, one as "chief engineer" (there was the fireman and myself) of a factory out in the country.

Now this factory had not been kept up in good shape. The fireman had been allowed to do as he pleased, and the proprietors thought that an engineer that would grab a squirt can and squirt oil all over the engine whenever the boss came round and sweat and stink while giving a nice song and dance about how difficult it was for him to keep up steam, etc., must of necessity be a first-class man.

I took charge; the first thing to be done was to reform the method of firing. I had to "lick" the fireman before he could be made to understand. There were four small boilers in use, and it required the power of two of them to run the engine while the steam from the others was used in the dye house and for heating, etc. Everything was driven to the limit.

Knowing something of the "science of steam," I concluded that the boilers were strong enough to carry a ten pound higher pressure which would enable a back pressure of five pounds to be carried on the engine without reducing the transmitted power, and with this five pounds back pressure the exhaust could be made to run the dye house, serve for heating, etc.

After much pleading the change was made with very gratifying results, to the engineer at least. During the following December, about a year after I had taken charge, and at the time of annual stock taking, one of the proprietors came into the engine room and said that they were all at sea regarding the coal consumption during the last year. As I remember it, he said:

"William, at the time you came here we were about to buy another 50-horsepower boiler in order to keep up steam. We did not buy the boiler, have never had to shut down for want of steam, have during the past year turned out our usual complement of finished product, but it seems as if we had burned 600 tons less of coal. Now, how is that? Could it be possible that a mistake was made in the inventory last year, or have the coal merchants neglected to charge for some cargo?"

Then it was William's time to grin, as he explained how it had all come about, and he took the hot air he was deluged with in the expectation that his Christmas turkey would be a fat one. But as Christmas came and went without his seeing the bird, the hope was that it would come as a New Year's present and as the coal delivered in the bunkers cost over $4.50 per ton the disappointment was great when nothing was offered except the hot air referred to above. However, when later on I asked for a raise of $1 a day, I got it, P. D. Q.

Now I know that others have done for their employers as well as in the above case, and have never secured any reward. This brings up the question, why?

Is it not because the engineer was too modest to assert himself and ask for it? Or, is it because having brought a rundown plant up to a high state of efficiency where the engineer can take his comfort, he knows that his employers can, for a time at least, get along with a cheaper man and he is afraid to ask better compensation for fear that he will lose the job. If not this, then what is it that compels him to submit to inadequate compensation for the service rendered? Who can answer?

As an employer, I find that the most skilful men are the cheapest to hire. By paying above the "prevailing rate" I win out, as the class of men employed are intelligent enough to know that my success in business means the continuation of the easy boss. I admit, however, that this scheme could not be successfully carried out where large numbers of men are employed.

But this is a digression. What we want to determine is why are there so many poorly paid and hard-worked stationary engineers.

This class of men cannot well avail themselves of a trade union, for obvious reasons. Take the way in which many of them are made. A good lusty greenhorn is put to work firing, and right here the writer admits that it is his preference to take the greenhorn in preference to the seasoned fireman, as he can generally teach the greenhorn to do the work the way he wants it done, while the seasoned man has his way which he cannot, or will not change.

Soon the fireman is allowed to stop and start the engine, help the engineer take up lost motion and make repairs, and soon the "boss" concludes that by raising the fireman's wages a trifle he can dispense with the services of his higher-priced man.

Now, whose fault is this? The average mill or factory owner is busy with the office end of the business, must look out for the purchase of raw material, the sale of the finished product, etc., and can scarcely be expected to distinguish between the really first-class man who generally is modest and retiring, waiting in the hope that by some miracle the boss will learn of his merits, and the loud-mouthed faker who by his cheek manages to get the center of the stage.

Is it not rather up to the really good engineer when he is about to make a change to first look over the plant, determine what improvements he can surely make, and then make his bargain *before* he takes employment?

It is the writer's practice when he is about to take employment to state his price for services, and he is terribly disgusted when he is about to employ help and asks what wages he is expected to pay to be answered by, "Whatever you say is right, goes." If the man who has the service for sale does not know its value, how can the proposed employer be expected to know?

In an experience of over forty years in office and engine room, I have never heard of an employer going to his engineer with a threat to discharge him unless he took more pay, and I believe that the horrible conditions under which many engineers work is their own fault because of their failure to insist on better sanitary conditions.

An article in Power of recent date intimated that there is a project on foot to improve the condition and standing of the stationary engineer and while wishing it God speed I cannot see how it can be done unless we can get rid of the beer-guzzling bums and grafters that hold down real engineers' jobs, for while the writer is not by any means a blue-ribbon man he believes it is this class of man that conveys the impression to the public that the engineers of small plants at least, are low-lived fellows whose chief end in life is to draw a theoretical diagram with the end of a dirty finger in a pool of beer on a barroom table. All of which means that before we can greatly improve the condition of the engineer we must improve the character of the men. Then we can show the employer that it is good business to pay higher wages to a clean-cut man who knows the meaning of good treatment.

This rather rambling harangue has been written at intervals and amid much confusion. It is not as explicit as I would have it, but I trust that, such as it is, it will set some members of a very worthy class of men to *thinking*.

W. H. Odell.

Yonkers, N. Y.

Questions for Hoisting Engineers

I was much interested in the questions for hoisting engineers which appeared in the July 5 issue and would like to offer a few criticisms.

The majority of the questions are perfectly fair and of such nature that anyone who was fit to serve in the position for which they were given should have been able to answer. But there are two or three questions such as No. 4 and No. 11 which are rather too much on the theoretical side and would be more in place for an examination for a designing engineer than for a practical operating engineer. While it is undoubtedly a good thing for engineers to be acquainted with methods of design, it is hardly an essential requisite for a practical operating engineer.

As regards questions No. 5 and No. 6 the data given are insufficient to allow of any answer at all. In question No. 5 the bursting pressure of a 72-inch by 18-foot return-tubular boiler is asked. As the bursting pressure of any boiler depends mainly on the thickness of the plate used, it would be impossible to state the bursting pressure unless the thickness were given. In question No. 6 the revolutions per minute of an engine are asked, given the diameter, stroke, steam pressure and horsepower. As the mean effective pressure depends upon the cutoff, no satisfactory answer can be given to this question unless the cutoff is given.

W. L. Durand.

Washington, D. C.

I have read carefully the questions used at the examination of hoisting engineers at Terre Haute, Ind., May 10 and 11, and printed in Power for July 5.

Those questions which, in my estimation, are unfit for such an examination I will discuss in their order and in each case give the reason why I take exception to them.

5. (a) It is impossible to answer this question without knowledge as to the thickness of the metal, kind of joints, size of rivets, etc. (b) Impossible to answer due to the insufficient data given in 5 (a). (c) Assuming that this refers to 5 (a), an answer is impossible for the same reason as in the case of 5 (b).

6. (a and b) As the mean effective pressure is not given it is impossible to determine the necessary speed.

11. This question would hardly concern an operating engineer. Decisions on such questions are made by the designing engineers.

12. This question is an impractical, theoretical one and one which, in practice, I am convinced, would never confront an operating engineer.

16. (a) Is it theoretically possible to draw water vertically more than 30 feet?

22. (b) This question is ambiguous. The question should read, "When the pitch diameter," etc., instead of "When the diameter," etc.

23. (a) This question also appears to be ambiguous.

Ernest Progst.

Rocky Ford, Colo.

Trouble with the Tension Carriage

With reference to Mr. Thompson's request, in the July 5 issue, for opinions on the arrangement of his tension carriage. I will venture the assertion that his idea about reversing the two idlers is a good one, although I fail to see any reason why they should not work either way. All that I ever had any experience with were located as he suggests.

With the carriage located as per the sketch in his article, the traveling idler is required to take up the slack of the rope, and also to overcome the friction of the rope in the sheave of the stationary idler as well as the friction in the bearings. But even at that it seems strange that it requires so large an amount of counterweight to move the carriage.

It sometimes happens that the track will settle out of alinement and cause the carriage wheels to bind on the rails, more especially if the rigging for the carriage track is made all of timbers.

As to taking up the slack without adding so much weight I always found an efficient remedy in moistening the rope with an ordinary sprinkling can enough to allow a small amount of slippage.

Of course, there must be sufficient weight to hold the slack after taking it up. When a given weight is not sufficient to take up a small amount of slack, due to the friction of the carriage and the rope in the sheaves, a little moistening helps out wonderfully. The slack usually shows most on one side of the strands of rope and the rope after becoming damp will slip some in all of the grooves and finally equalize.

In my experience with this style of rope drive I have never found one but that was first class and I consider the type to be ideal for many locations.

Joseph Stewart.

Hamilton, O.

Use of Superheaters

In the July 5 issue Mr. Holley makes inquiry as to the advisability of installing steam superheaters in return-tubular boilers in order to obtain dry steam at the end of a line 900 to 1500 feet long. While it is quite true that by using superheated steam at the beginning of a long run the steam would still be dry at the receiving end, it is very doubtful whether there would be any actual saving in such a procedure. Superheaters cost money and are only justifiable where the use of superheated steam will result in an increase in overall efficiency. In the present case there would not only be the expense of the superheaters but a considerable extra expense due to having to install them in boilers already in operation, and this extra item of expense would not be offset by any gain in efficiency.

With a well insulated pipe the loss of pressure even in a run of 1500 feet should not be over 2 or 3 per cent.; and by using a separator at the receiving end and slightly raising the pressure at the generating end, the steam would be in practically the same condition as if superheaters were used and less precaution taken as to the insulation of the pipe. The majority of steam-supply companies, by taking extra precautions as to insulation, are able to transmit steam from one to two miles with a loss of pressure not exceeding 10 per cent., and by the use of separators to furnish steam which is in every way satisfactory for use.

W. L. Durand.

Washington, D. C.

We are furnished city water which passes through a 2-inch meter, then to a triplex pump and thence through another meter to a service tank about 60 feet above the pump. The pump was installed because the city pressure was too low to supply this tank.

Can some of the readers of Power tell me why the readings of the two meters vary from 30 to 60 per cent.?

H. Blue.

Kirksville, Mo.

Notes on Barometric Condensers

By Warren H. Miller

> *When selecting condensing apparatus it is of great importance to keep the factor of simplicity in mind, avoiding such devices as will add to the cost of upkeep and necessitate constant supervision. The barometric condenser is simple as well as reliable, and is particularly suited for many classes of service.*

The tendency in modern power-plant design is toward reducing the equipment to the simplest form possible. As one looks back over the day's work at the power house, or turns over the sheets of the log book, it is always the repairs to the small auxiliaries that outnumber the troubles with the main engines.

It is the writer's belief that the men best qualified to lay out power plants are not those with mere drafting-room experience, but rather the engineers who have *lived* with the machines themselves, and have run, not one, but many power plants. To such an engineer, a multiplicity of small complicated apparatus, all tending, no doubt, to increase the total theoretical economy, awakens no enthusiasm. He has had to repair hundreds of such devices, and has never yet seen one that did not represent a larger repair account than any economy it had effected.

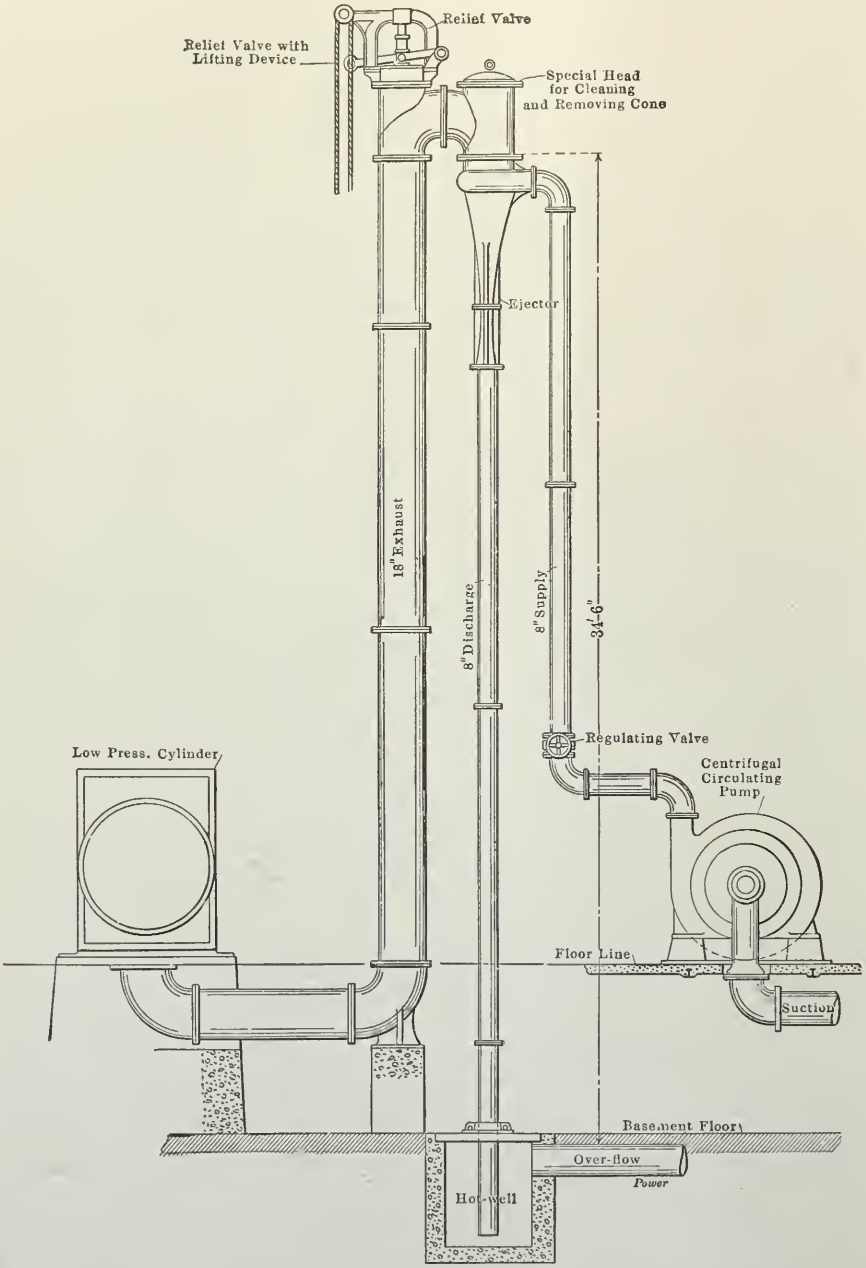


Fig. 1. General Arrangement of Barometric Condenser

Particularly important in this respect is the condensing apparatus, which should be made as simple as possible to avoid the risk of shutdowns. For handling the circulating water, the substitution of an induction motor and centrifugal pump for a duplex or flywheel pump is one of the first improvements in the right direction. The combination of a small reciprocating engine or turbine with the centrifugal pump is sometimes used, but it is well to keep away, if possible, from both small engines and small turbines. As motive power for a centrifugal pump the induction motor is hard to equal, for simplicity, durability and general foolproof qualities. The pump itself will give little trouble if the suction is kept tight. A very little air, however, if entering the suction side, will soon stall the pump. Most of the air leaks in through the stuffing box, and the better makes have a double packing between which is a brass cage. This space is always full of discharge water, which, being of higher pressure than the suction, causes water to leak outward instead of air inward.

It is in the condenser itself that the greatest simplification can be obtained. The best economy of operation lies in the use of the surface condenser, as all the condensed water can be returned to the boilers as feed water, but, against this must be offset the fact that very little heat is returned with the condensed water. Practically all cylinder lubrication must be foregone unless one cares to risk the boilers with automatic devices for trapping the oil; and, if the water is salt or dirty, there is a possibility of further troubles with the boilers from salt and mud getting in through leaky condenser tubes. Finally there is the high first cost of the surface condenser—better spend the same money on better boilers and engines, or more of them. Unless one has to pay high prices for the boiler feed water itself, the surface condenser hardly shows enough superiority over the others to warrant its high cost.

If it is decided to waste the water and use jet condensers, the first simplification is to combine the circulating and air pumps into one, as is often done. Some form of flywheel pump is generally used for this purpose, but they often give trouble, because of having to handle such large volumes of water with so little clearance as is necessary in order not to destroy the vacuum through reëvaporation of the water vapor.

BAROMETRIC CONDENSERS

The simplest condensing device of them all is to eliminate the dry-vacuum pump entirely, use a centrifugal pump for circulating, and a barometric condenser to both condense the steam and remove the

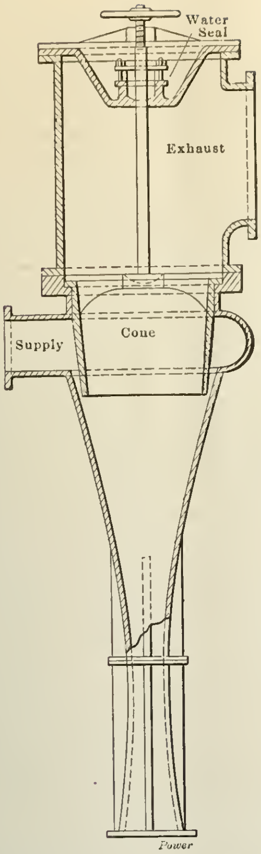


Fig. 2. Section Through Bulkley Condenser

air. Briefly stated, the barometric condenser is an ordinary jet condenser, discharging through an ejector into a vertical tail pipe 34 feet long. As is well known, a column of water 34 feet high will pull a vacuum by its own weight, and the ejector removes the air in much the same way that a steam siphon draws water. The water will never back up into the cylinders, as with a jet condenser, if the vacuum pump stops, for the column of water in the tail pipe is too heavy to be sucked back; if the circulating pump stops, the water simply runs out, the relief valve opens, and the engine runs noncondensing. When the water comes on again, the relief valve closes with a snap, and the engine runs condensing.

The argument usually raised against the barometric condenser is, that it cannot possibly take care of all the air without the help of a dry-vacuum pump, except with very short and direct exhaust connections. This argument is usually advanced by the dry-vacuum pump salesman, and he is very earnest and positive about it, so much so that the writer was once scared into having a boss cast on a new barometric condenser, and drilled and tapped for a 2-inch dry-vacuum connection, but it was never used. The last installation put in by the writer handled two 300-kilowatt cross-compound marine engines, a 100-kilowatt simple engine and a 70-horsepower duplex air compressor, all on a 14-inch exhaust pipe. There were 80 feet of 14-inch exhaust pipe and three 6-inch flanged tees on the line, besides a 14-inch base ell. With the two engines alone, 300 gallons of condensing water were required per minute at 70 degrees entrance and 118 degrees discharge temperature, the vacuum being 28½ inches. With the 100-kilowatt simple engine added, the vacuum fell to 25 inches; and with the air compressor further added, the vacuum dropped to 18 inches, while the discharge temperature rose to 135 degrees. This drop of vacuum was not because of air, but because the condenser was of only 1000 horsepower capacity on a 26-inch vacuum, and the exhaust of the two simple engines brought up the actual steam to be condensed far above this.

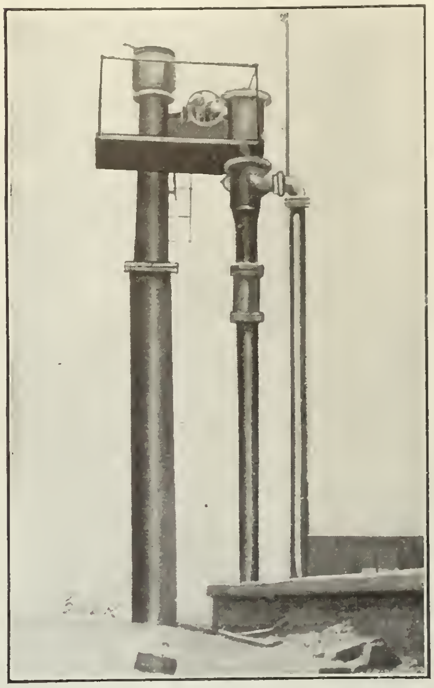


Fig. 3. Bulkley Condenser in Service

TYPES OF BAROMETRIC CONDENSER

Fig. 1 shows the general arrangement and Fig. 2 a sectional view of the Bulkley condenser. However, there is often a gate valve placed between the condenser and the exhaust main, not shown in the general-arrangement layout. The hight should not be less than 34 feet 6 inches from the joint in the condenser head above the water inlet to the level of the overflow water in the discharge sump. The distance from the center line of the riser to the center line of the tail pipe, with the valve inserted, is about 4 feet 2 inches; hence it is quite feasible to have the exhaust riser come up inside the power-house wall, and the tail pipe on the outside, spanning the wall, as shown in Fig. 3.

The interior of the condenser is simply a hollow, cast-iron, conical ring, with a spider and screw bolt for raising or lowering the cone. This is set at the factory, and should not be readjusted, except when absolutely necessary, as it gives just the right amount of water into the ejector. As will be noted from the illustration, the water is directed down the walls of the condenser, while the steam enters and is condensed inside the hollow cone. With it is the air, and the mixture of air, steam and condensing water shoots downward into the ejector where it acquires a high velocity, and, once the air is past the neck of the ejector, it must pass through the tail pipe. It is practically impossible to flood this condenser, as the cone moves in a tight fit, and the only conditions under which the water could reverse its direction would be if the cone were raised too high, admitting more water than the ejector will pass, and thus backing up into the exhaust pipe.

With jet condensers, where this is sure to happen if the air pump stops without the water being shut off, a good protection is to put a snifting valve on the low-pressure cylinder cover, with a spring to hold it open when once its seat is raised.

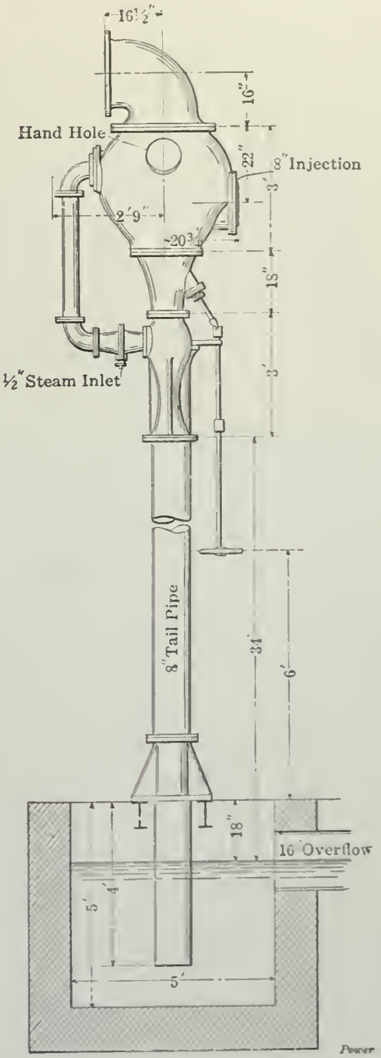


Fig. 4 Wheeler Condenser

The water always tends to go where the vacuum is best, so on flooding, a little of it immediately shoots back into the low-pressure cylinder. Here it is immediately driven out through the relief valve and the snifting valve, and the latter, remaining open, breaks the vacuum, and the exhaust pipe immediately fills with steam and drives the water out.

Fig. 4 shows the Wheeler barometric jet condenser. In this type the condenser and ejector are separate, and the latter has an air connection outside the condenser to suck the air out above the spray. A valve controls this spray, which spurts upward over a mushroom baffle plate. It must reverse before acquiring any downward velocity, and therefore has no means of catching the air above it, hence the outside pipe connection to the ejector. Both this condenser and the Bulkely can be connected to any vacuum pump, but it should never be necessary to use one.

Fig. 5 shows the Alberger barometric condenser with automatic spraying adjustment. In this condenser a small air pump is a necessity, as no ejector is combined with the condenser. All the water is carried off, however, by the tail pipe, and the hight of the column suffices to pull the vacuum; all the air pump has to do is to take away the air as fast as it accumulates.

Regarding the relative amounts of circulating water required by the surface condenser and the barometric condenser, it is more than likely that the actual figures of the former equal, if not exceed, those of the latter.

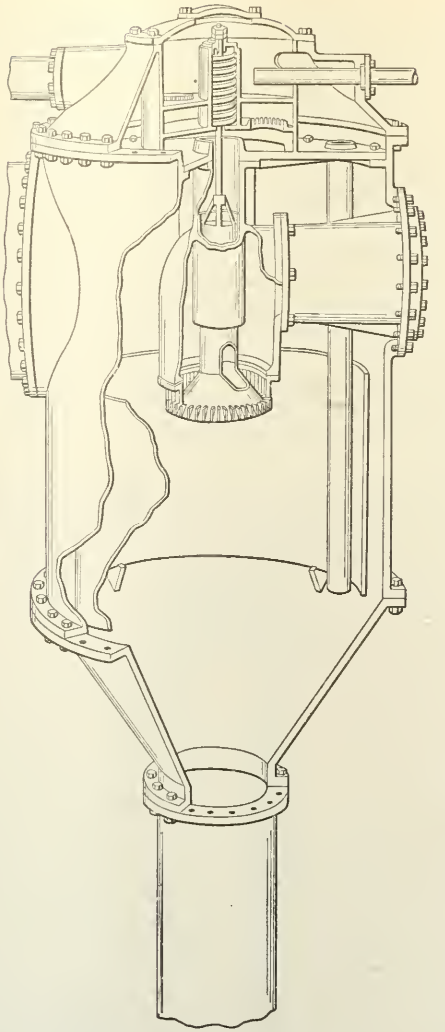

Power

Fig. 5. Alberger Condenser

A deciding factor in the selection of a condenser is the amount of vacuum that is actually needed. With the barometric type, without any dry-vacuum pump, 28 inches is easy to obtain and hold; even with leaky piston rings it will hold up to 26 inches. But, with a turbine, 29 or 29½ inches are well worth trying for, because the turbine can use these last inches to a great advantage. The ordinary compound, or the triple-expansion, or even the quadruple-expansion engine cannot utilize these last 2 inches, even if the condenser can produce them, because the volume of every pound of steam at 29½ inches vacuum is excessive, and no cylinder will take care of it. The best the ordinary compound engine can do is to exhaust into the condenser at about four pounds absolute, at which the steam occupies about 90 cubic feet per pound. This corresponds to a 22-inch vacuum, and, if one gets 28 inches, or even 26 inches, it helps principally by giving the exhaust a greater velocity into the condenser and cutting down the necessary area of the exhaust pipe. For turbines, however, very short and large exhaust mains should be used, and as a very little air will reduce the vacuum considerably, a dry-vacuum pump would more than compensate for the trouble and expense of keeping it in first-class condition.

[There are a number of types of barometric condenser on the market which are not mentioned by Mr. Miller. These include the Tomlinson, the Weiss, the Le Blanc and others.—Editor.]

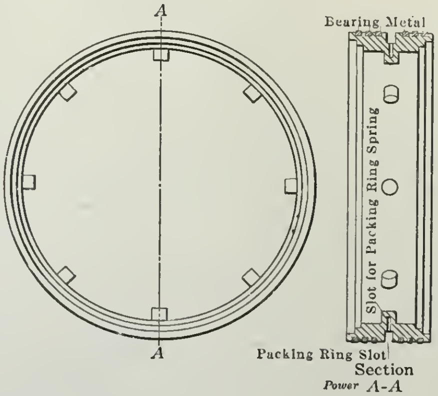
Frictionless Bull Ring

By C. L. Greer

To those who operate engines with large cylinders or have trouble from the wearing of bull rings, the following experience in the power station of the Northern Texas Traction Company may be of interest:

There was installed a 1500-horsepower cross-compound engine, the low-pressure cylinder of which was 52 inches in diameter. From the beginning much trouble was experienced from wear of the bull ring despite the fact that the ring was frequently shifted around to various positions. A liberal supply of cylinder oil and graphite failed to relieve the trouble and at the end of a year the ring was so nearly worn out that it would little more than hold the packing rings; these being in sections, it was feared that should water come over into the cylinder a section might be forced out of position and do damage.

The bull ring had two packing rings one near each side, while near the center were two strips of Allan metal which were embedded in slots around the ring. These strips of metal, about one inch wide, while helping some, did not furnish sufficient bearing surface to carry the heavy piston without excessive wear.



Power

Bull Ring

To overcome the difficulty a new ring was made as shown in the accompanying sketch. Only one packing ring was used. This was 1½ inches wide and placed in the center of the bull ring, while on each side were placed three rings of Allan metal, each ⅞ inch wide. This gave a total bearing surface of 5¼ inches.

The slots in which the Allan metal was placed were made about 1/16 inch wider at the bottom than at the top in order that the metal might be retained. The metal was tightly peened into the slots and finished about 1/32 inch larger than the bull ring.

After nearly a year's service the rings have shown very little wear, the cylinder has an excellent polish, and the bull ring has not been shifted from its original position. The bull rings of three other 900-horsepower engines in the plant have been similarly fitted and are giving excellent satisfaction.

Where trouble of the kind mentioned is experienced, it is economy to fit the pistons with some good frictionless bearing metal as it effects a saving in oil, power and wear on the cylinders and bull rings.

Power and The Engineer

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for July, 1910, *was* 138,000.

August 2.............. 34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Editorial

Engineers' Opportunities

How often are the educational opportunities afforded by meetings of engineers' associations taken advantage of and improved by those who attend such meetings? After the routine business is despatched and the question, "Has anyone anything to offer for the good of the order" is asked from the chair, what is the usual response? In most cases, it is safe to say, instead of an announcement from the educational committee that there has been provided a lecture or a discussion on some live topic, or the opening of the question box and the subsequent reading of the queries, the stereotyped "We will close in due form" follows.

Any one of the thousands of subjects upon which most engineers might easily be better informed would afford material for an evening's discussion, profitable to all present and especially to those who participate.

To the man who takes part in a discussion with a view to mastering a subject which to him has been a sealed book, there comes a new interest, and he is apt to seek other knowledge and become a more valuable member of the organization as well as a better informed engineer.

There are successful engineers by the hundred who owe their start in the proper direction to the fact that they have been compelled to grapple with special problems and struggle with these until all obscurities were mastered, the knowledge thus gained adding to their mental equipment.

Steam engineering is not an exact science, to be learned completely from textbooks, nor can its principles be absorbed by a desultory course of reading in which the "knotty" subjects or statements are carefully skipped. Hours or days given to a problem will be time well spent if in the end the mastery is complete, and the value of such mastery is immeasurable. In other words, it is the study, not mere reading, that counts.

Through many channels the outsider is invading the field that is the engineer's own, and with a success depending upon the mental alertness of the engineer himself. It is obvious that the man in responsible charge of any power plant should know more about every detail of its operation than any outsider, however well trained. Special knowledge is the result of special training, and the engineer has better facilities for special training in his vocation than any other man in the world; he should follow up his advantage.

Economical Firing

If a man should give an agent a sum of money to invest in stocks he naturally would expect the agent to exercise sound judgment so as to insure the greatest possible return upon the investment. In a boiler plant, the coal pile represents a certain amount of the employer's capital entrusted to the fireman's judgment for economical expenditure. If a large percentage of the heat is allowed to escape up the stack it represents just that much money thrown away.

In many large hand-fired plants it is common practice to place one responsible man in charge of the boiler room and have the actual firing done by cheap foreign labor, the fitness of a man being judged not by what he knows about keeping a good fire, but by how much coal he can shovel. The advisability of such practice is open to question and it is doubtful if it tends toward ultimate economy. It seems that with the growing practice of forcing boilers it would be better policy to employ a more intelligent class of firemen who could be made to realize that more money can be wasted or saved in the boiler room than in any other part of the plant. Of course, to make this effective the fireman should be remunerated accordingly.

There are certain fundamental rules which must be observed in all boiler rooms in order to conform to good practice. These include maintaining clean, even fires, feeding moderate amounts of coal at regular intervals, delivering a uniform supply of feed water, blowing down and cleaning boilers at regular intervals (dirty heating surfaces mean more coal for the fireman to handle), regular times for cleaning the fires; if there be a battery of boilers the fires should be cleaned in regular order and the periods so arranged as not to affect the steam pressure on the line nor place undue load upon the other boilers.

Beyond these general rules the conditions at each individual plant will require special study and treatment accordingly. Different coals require different methods of firing and various conditions of load demand different rates of combustion. It is very important that the proper amount of air be supplied for complete combustion; insufficient air supply will cause smoke, representing unburned particles of carbon wasted from the stack, while too much air means a loss in heat units which, instead of going to heat up the water, are utilized for raising the temperature of the excess air which passes up the stack. An experienced fireman can often tell by the appearance of the fire whether he is obtaining approximately perfect combustion.

One of the greatest aids to efficient boiler-room operation is the keeping of a complete log or record of the coal burned, feed water used, steam pressures, intervals of firing and cleaning, draft, etc. This not only permits the chief to keep a close supervision over the boiler room but also enables the fireman, by a comparison of results from day to day, to see where his methods can be improved. The fireman who takes the proper interest in his work will read up on combustion and will make a careful study of the conditions confronting him. The old saying that "the fire room is the place where brawn counts more than brains" can no longer be accepted.

Settings of Return Tubular Boilers

Although it is claimed by many, and with reason, that in the matter of progress the builders of boilers have been distanced by the builders of steam engines, there has been a marked improvement over the boilers that were in use twenty-five years ago. This applies particularly to the material put into them and the methods employed in their construction. A weak point in boiler practice, however, is found in the boiler setting, particularly in small plants where return-tubular boilers are largely used. When a manufacturer sells a return-tubular boiler, he loses all engineering interest in it as soon as it is loaded on the car, and forgets about it entirely as soon as a check balances the account. When the boiler reaches its destination, it is unloaded by most anybody and is probably set in position by some inexperienced person. Then the local brick mason is set at work, using either his own ideas or those of others connected with the plant as to the best manner of bricking in the boiler. The probabilities are that the engineer of the plant has never before had anything to do with setting a boiler, the superintendent knows less than the engineer about the work, while the mason's experience has been restricted to foundation piers and domestic chimneys.

A properly built boiler setting will not develop cracks in the side walls and rear connection within a few months after it is put into use, but if the expansion of the boiler is not provided for, the setting will be pushed out of place, no matter how well it may have been built; and if the workmen do not know how to set a boiler, proper provision for its expansion will not be made. It is highly probable that over fifty per cent. of return-tubular boilers are incorrectly set, not because of any lack of desire to do the work right, but because the workmen do not know how.

The makers of water-tube boilers do not sell them and allow the first mason found to do the brickwork. They know that it is absolutely necessary to have a boiler properly set if it is to come up to its rating when put to work, and for this reason the selling of a water-tube boiler means that the maker will also attend to the brickwork. The result is that the boiler is provided with a setting and furnace such as experience has shown to be best adapted to the type being set up.

If the builders of water-tube boilers find it of advantage to construct the settings and furnace of their boilers, it would seem that the makers of return-tubular types could learn a lesson from this practice that would be of advantage to them.

Perhaps building the furnace and setting of a return-tubular boiler would be a greater departure from the present general practice than is expedient or even necessary. However, if the builders of this type of boiler would determine by experiment just what type of setting is best suited for their particular product, with various makes of grates and grades of coal, and embody the results in specifications to be sent to each purchaser of their boilers, better results would be obtained from the boilers in regular work.

Coal Analysis

There are two methods of analyzing coals. The "proximate" analysis determines the proportions of moisture, volatile matter, fixed carbon, sulphur and ash in the coal; the "ultimate" analysis determines the proportions of carbon (total), hydrogen, nitrogen, sulphur and ash, and what is left is classed as oxygen. Just why oxygen should be saddled with all the mistakes and uncertainties of the analysis does not seem clear. The reason commonly offered is that there is no means of directly ascertaining the proportion of oxygen in coal; therefore, when everything determinable has been determined, what is left must be oxygen! This reasoning would be defective even if all the other constituents could be measured with absolute certainty, and when it is remembered that fixed carbon and ash are about the only constituents that can be measured with anything like accuracy, the defectiveness of reasoning degenerates into absurdity.

Aside from the scientific view of the matter, however, it is not only unwarrantable to charge all the uncertainties of analysis to oxygen but it gives a distorted character to the fuel. Free oxygen makes for free burning, and when an ultimate analysis shows ten per cent. of oxygen as compared with less than two per cent. of moisture in the proximate analysis, it looks to the uninitiated as though that coal were just reeking with free oxygen, waiting for a chance to grab some of the combustibles.

If there is any advantage or satisfaction in recognizing the oxygen which forms part of the moisture (which is of no importance whatever from the fuel standpoint), it can be assumed to be eight-ninths of the moisture content with less violation of scientific scruples than the present custom affords.

Why not be at least truthful and write the actually unknown difference as "undetermined?"

Man Is a Failure

When he has no confidence in himself nor his fellow men.

When he values success more than character and self-respect.

When he does not try to make his work a little better each day.

When he becomes so absorbed in his work that he cannot say that life is greater than work.

When he lets a day go by without making someone happier and more comfortable.

When he tries to rule others by bullying instead of by example.

When he values wealth above health, self-respect and the good opinion of others.

When he is so burdened by his business that he finds no time for rest recreation.

When he loves his own plans and interests more than humanity.

When his friends like him for what he has more than for what he is.

When he knows that he is in the wrong, but is afraid to admit it..

When he envies others because they have more ability, talent or wealth than he has.

When he does not care what happens to his neighbor or to his friend so long as he is prosperous.

When he is so busy doing that he has no time for smiles and cheering words.
—*Louisville Times.*

The man who makes the same mistake a second time does not deserve to be excused.

To lend a hand is good policy; you never can tell when you will need a lift yourself.

On the average, a cent will purchase one hundred thousand heat units in the shape of coal; this is no excuse, however, for wasting any.

Inquiries of General Interest

Condensing and Noncondensing

Should the valves of a Corliss engine be set differently when running condensing from what they would be for running noncondensing?

T. R. W.

No, the valve setting should be the same for both conditions.

Change of Cutoff

How can I change the point of cutoff on a Corliss engine?

D. R. J.

You cannot. That is the function of the governor which controls the speed by shifting the point of cutoff as the load and steam pressure vary.

British and United States Gallon

What is the difference between the British and the United States gallon?

C. F. P.

The British gallon contains 277.274 cubic inches and the United States gallon 231. A British gallon of water weighs 10 pounds and the other 8.34.

Diameter and Hight of Chimneys

Will you tell me what dimensions of chimney are necessary to properly take care of 66 square feet of grate surface? Our chimney is 30 inches in diameter and 70 feet high. Is that adequate? Also will you give me a rule for finding hight of a chimney without the use of a transit?

A handy rule for calculating the diameter and hight of a chimney for a given grate surface is: Make the area of the chimney one-seventh the area of the grate and make the hight of the chimney 25 times its diameter. This, in your case, would give a chimney 42 inches in diameter and about 90 feet in hight. Directions for finding the hight of a chimney without the use of surveyor's instruments were published in the issue of July 5.

Foaming and Priming

What is the difference between foaming and priming in boilers?

J. C. D.

Foaming is a violent frothy ebullition of the water in the boiler and is caused by an excess of impurities, while priming is the mingling of a fine spray with the steam and is usually caused by an insufficient steam-liberating surface.

Low Water

What should be done in case the water is out of the water column?

H. A. M.

The fire should be covered and the boiler allowed to cool until it can be entered and examined.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Boiler-feed Piping

How should water be fed to boilers?

S. B. R.

The feed pipe should enter the boiler through the front head near the shell just above the tubes, pass to the rear, cross to the opposite side, turn downward and discharge the water between the shell and the tubes.

Ratio of Expansion

The diameter of the high-pressure cylinder of a compound engine is 15 inches and that of the low-pressure cylinder 30 inches. If the cutoff takes place at one-third of the stroke in both cylinders, how many expansions are there?

H. A. S.

When steam is expanded in two cylinders the number of expansions is found by multiplying the number of expansions in the high pressure by the cylinder ratio. Neglecting clearance, condensation, etc., a cutoff at one-third of the stroke in the high-pressure cylinder would result in three expansions, which, multiplied by the cylinder ratio, which in this case is 4, would give 12 as the total number of expansions in both cylinders.

Strength of Boiler Plate and Pitch of Rivets

Will you please tell me why it is that boiler plate is not used that has a tensile strength of 100,000 pounds or as low as 30,000 pounds, but is always around 55,000 to 65,000 pounds?

How do you find the right pitch of rivets in a boiler joint with a single-riveted lap, a double-riveted lap, a double-riveted butt with two cover plates and a triple-riveted butt with two cover plates, when the thickness of plate and the diameter of rivets are known?

T. F. B.

One very good reason for not using steel of the tensile strength mentioned is because steel of such strength is not made for boiler plate. Steel of high tensile strength is deficient in ductility, and when of low tensile strength the elastic limit is low, both conditions being directly opposite to what is desirable for material for boiler construction.

When the shearing strength of the rivets is equal to the tensile strength of the plate, the maximum strength of a riveted joint, taking all the dimensions in inches, is expressed as follows:

$$\frac{\text{Pitch of rivets} - (\text{diameter of rivet hole})}{\text{Pitch of rivets}} =$$

$$\frac{\left(\begin{matrix}\text{Diameter of}\\ \text{rivet hole}\end{matrix}\right)^2 \times 0.7854 \times \text{number of rows of rivets}}{\text{Pitch} \times \text{thickness of plate}}$$

$$\text{Pitch of rivets} - (\text{diameter of rivet hole}) = \frac{(\text{diameter of rivet hole})^2 \times 0.7854 \times \text{number of rows of rivets}}{\text{thickness of plate}}$$

$$\text{Pitch of rivets} = (\text{diameter of rivet hole})^2 \times 0.7854 \times \frac{\text{number of rows of rivets}}{\text{thickness of plate}} + \text{diameter of rivet hole}$$

Where there are two rows of rivets, then,

$$0.7854 \times 2 = 1.5708,$$

and for a double-riveted joint, measuring along one line of rivets.

$$\text{Pitch of rivets} = 1.57 \times \frac{(\text{diameter of rivet hole})}{\text{thickness of plate}} + \text{diameter of rivet hole.}$$

Where there are three rows of rivets, then,

$$0.7854 \times 3 = 2.3562,$$

and for a treble-riveted joint, measuring along one line of rivets.

$$\text{Pitch of rivets} = 2.3562 \times \frac{(\text{diameter of rivet hole})^2}{\text{thickness of plate}} + \text{diameter of rivet hole}$$

These rules apply to all thicknesses of plates and diameters of rivet holes.

Boiler Feeders and Repairs

What are used for boiler feeders? What should be done if they fail to work?

N. A. H.

Pumps, injectors and return traps. If for any cause the apparatus used fails, it should be immediately put in working order.

Safety Valves

What is the safety valve for, and how constructed, and where should it be placed?

F. E. D.

The safety valve is an appliance which prevents an excess of pressure by opening when any predetermined pressure is reached, allowing steam to escape. In a general way the safety valve is made like other valves of the angle type except that it is closed by a spring or lever instead of a screw. It should be attached to the steam space of the boiler independently of all other piping.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Vacuum and Gravity Return Systems

An arrangement of the Stickle open feed-water heater and receiver with a vacuum pump discharging into the receiver, together with a boiler feed pump conected to the receiver and heater, and working in conjunction with them, is shown in Fig. 1. The idea is to prevent the waste water from the return drips, passing from the heater through the overflow pipe, in case an excessive amount of condensation should be returned.

Generally there is not enough space in the open heater, between the overflow line and the point where the automatic control admits water to the heater, to allow any considerable amount of condensation accumulating before some of it will escape through the overflow pipe.

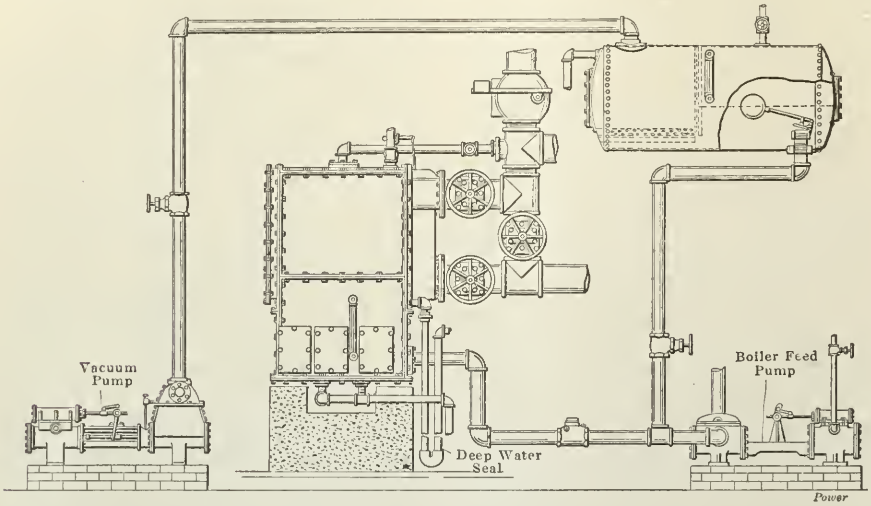


FIG. 1

The illustration shows the method of connecting up the heater with a vacuum heating system. With the receiver arranged as shown, the heater operates entirely by induction, no more steam entering the heater than is necessary to heat the water, consequently excellent results are obtained in oil separation, and this prevents the amount that is carried in the vapor from entering the heater; the balance of the steam going through the oil separator to the heating system.

The receiving tank, located above the heater, has a filter chamber packed with excelsior, with a downward filter that traps any oil that may get into the system. The float in the receiver drops when water is taken out by the boiler feed pump, which closes the valve. This allows the check valve in the section pipe between the heater and boiler feed pump to open, and allows water to come from the heater only when the supply in the receiver has been used, thus preventing any of the returned condensation to go to waste. When there is such a surplus of condensation returning that the feed pump will not handle it, the excess is stored in the receiver.

The heater can be cut out for cleaning without interfering with the heating system. In the summer time the receiver can be cut out, the exhaust steam passing through the heater to the atmosphere.

Fig. 2 shows a combination of feed-water heater, receiving tank and low-pressure traps. The boiler feed pump is connected to the receiver, which is piped to the atmosphere. The receiver works in conjunction with the heater automatically, and all returns come by gravity from the system to the receiving tank. The requirements are that the pump and receiver shall be located to conform to the gravity line of the returns.

By the arrangement shown, the open heater can be operated with a back pressure sufficient to deliver the exhaust steam to a point where it is used, free from oil, to the heating system, etc. The returns of condensation flow to the low-pressure steam traps, and thence to the receiver, which is open to the atmosphere and is connected to the suction pipe of the boiler feed pump, which operates automatically in conjunction with the open heater.

Each return is so installed that each coil will operate independent of one another discharging from pressure to atmosphere, which keeps the system clear and maintains a positive circulation.

By the method shown in Fig. 2, the receiving tank is sufficiently large to take care of the returns and will not allow any of the make-up water to come from the heater until the supply in the receiving tank has been used. Therefore, even though the amount of return may be excessive, the capacity of the receiver is such that none is permitted to escape through the overflow pipe.

The stickle open-coil heaters and traps are used with these systems, which are designed by the Open Coil Heater and Purifier Company, 502 S. Pennsylvania street, Indianapolis, Ind.

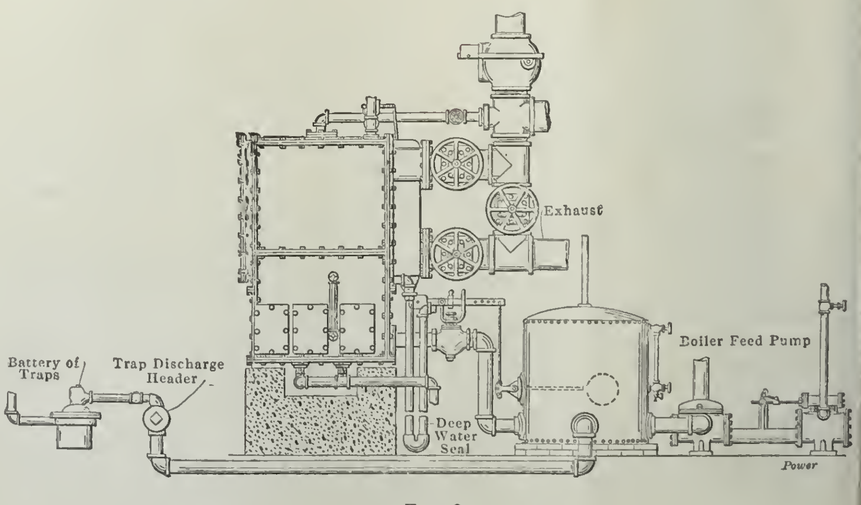


FIG. 2

Harrison Aertube Heater

The various savings that would be possible by the use of a direct fuel-to-air heater have led to the invention of several direct-heating systems, one of which is described herewith.

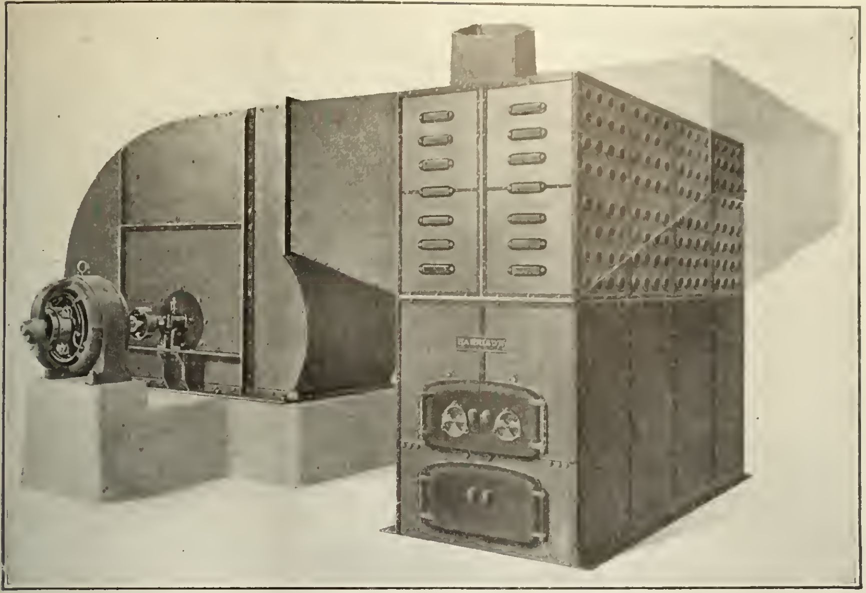


Fig. 1. Harrison Aertube Heater and Direct Connected Electric Fan

The Aërtube plant consists of a furnace under a bank of horizontal tubes, through which the air to be heated is driven by a fan to the distribution ducts connecting with the building to be heated. Fig. 1 illustrates a motor-driven fan and Aërtube heater. The heat from the fire strikes directly on the heating tubes and is distributed among them by baffle plates, as shown by Fig. 2. The transfer of the heat of the furnace to the air passing through the tubes is direct and immediate. An efficiency of 80 per cent. or more of the heating power of the fuel is guaranteed by the makers. In this type of heater the air passes through the tubes in a straight line from the fan to the distributing ducts, and parallel with the heating surfaces. The heater is made on the sectional plan, and so may be adjusted to the heating load in any special case.

While the Harrison system is applicable in any type of building, it is especially satisfactory where the heating system must also provide mechanical ventilation, and is particularly adapted to buildings which do not require steam for power purposes, but which have electric power.

The heater is made in sections of 32 tubes each, the tubes being 4 inches in diameter, 4 feet long, and supported at the ends in cast-iron tube plates. The patent joint between each tube end and the tube sheet consists of a ring of molded composition packing held in place by a cast-iron expansion ring. It is flexible enough to allow for expansion and contraction, yet absolutely prevents any gas from the furnace getting into the air. The different parts of the casing are jointed on the same plan, the composition packing being held in place by cover plates held by through bolts.

All parts are of standard sizes and are interchangeable. Consequently, the average mechanic can make the few repairs that may be needed, while the heater is in service.

While coal is the usual fuel, the Aërtube heater works well on the exhaust from gas engines, from forges, ovens, etc. Crude oil gives excellent results. Producer gas is a highly satisfactory fuel when a group of buildings is to be heated, since the gas can be piped from a central gas producer to heaters in the various buildings.

This heating system is made by the Harrison Engineering Company, New York City.

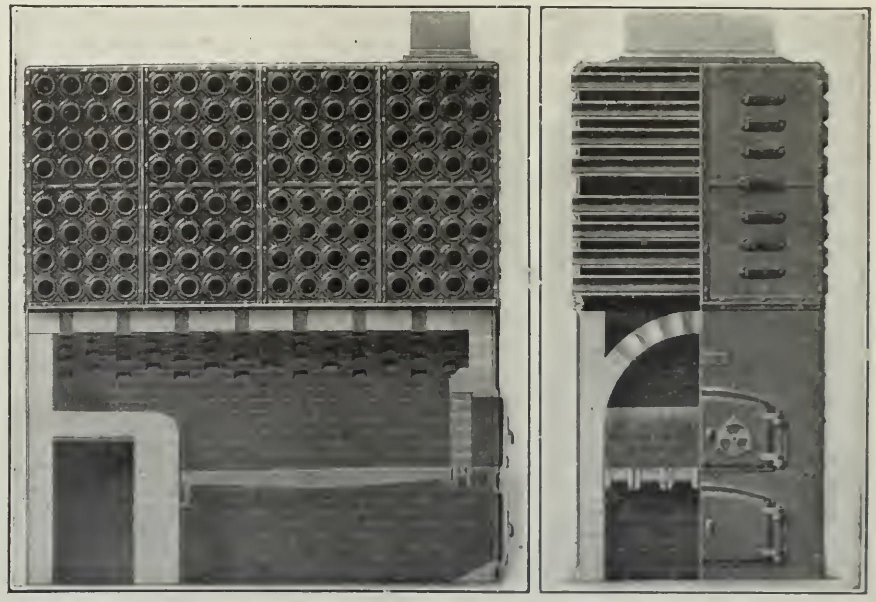
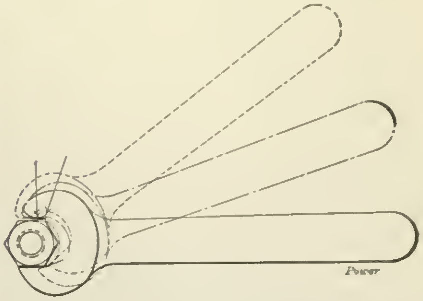


Fig. 2. Side and End View of Harrison Aertube Heater

The Williams Ratchet Wrench

This wrench contains no ratchet mechanism, although it does the work of a ratchet wrench. Its action is shown in the illustration. It is made exactly like an ordinary wrench, with the exception that the useless end of one side has been cut off. This produced a tool with a reciprocating action, and without wear on either nut or tool, it is said.



Positions Taken by the Williams Ratchet Wrench

There is no lifting of the wrench from the nut as it is turned on the thread. It can be used for either right- or left-hand adjustment. In operation the jaws are simply swung back, and the short jaw rolls on the flat side of the nut, releasing the jaw on the other side. When the long jaw comes in contact with the next flat surface of the nut, the short jaw hooks the corner diametrically opposite the one in contact with the point of the long jaw. A reverse motion of the handle causes the nut to turn. This wrench is made by the J. H. Williams Company, 150 Hamilton avenue and Richards street, Brooklyn, N. Y.

Michigan N. A. S. E. State Convention

Mayor Farrel, of Kalamazoo, was at his best when he made the address of welcome at the ninth annual convention of the Michigan State National Association of Stationary Engineers association at the armory. He reviewed the progress of the work done in the past by the association and displayed a remarkable familiarity with the organization and its purposes. In speaking of the exhibit. which occupied the same hall in which the opening exercises took place, he said that he was surprised and gratified to see that the display was of such mechanical usefulness.

Senator Walter Taylor also made a witty speech in which he welcomed the delegates and announced that the engineer was a man upon whom great responsibilities rested. National Secretary Fred W. Raven, of Chicago, made the response, and dwelt upon the progress and work of the association from the date of its inception. His remarks were especially well received by the delegates.

Addresses were also made by A. C. Benton, State president; A. B. La Franier, past president; E. J. Jacques, past national president; George D. B. Van Tassel, past national president, and J. D. Clement, secretary of the Kalmazoo commercial club.

At the business session which followed, A. C. Benton made a strong talk full of good advice and suggestions for the betterment of the association. He recommended that more interest be taken in the educational work, that the license law be supported loyally by every member of the order and that delegates at State conventions should by their conscientious devotion to the purposes which bring them together, their sobriety and general behavior, create a favorable impression wherever the meetings are held and demonstrate that the stationary engineer is progressing in company with the rest of the world.

Some earnest discussion was held regarding the passing of a State license law and it seems probable that the hopes of the engineers in this regard will be realized soon.

During the past year an educational contest has been carried out by the State organization and the awarding of the prizes was an interesting part of the program. Other State associations would do well to adopt this method of stimulating interest in the affairs of the order. Fred Muryne, of Kalamazoo, was awarded the first prize; William E. Fuller, of Battle Creek, second prize; Chauncy Davis, of Kalamazoo, third prize; Fred J. Tesch, of Flint, fourth prize; and W. E. Fuller, A. E. C. Smith, of Kalamazoo, and C. C. Hale, of Pontiac, the fifth, sixth and seventh prizes respectively.

Saginaw was chosen as the next place of meeting under the condition that the organization there be revived inside of a year. Saginaw had a flourishing organization at one time but different obstacles arose and the association failed. Since that time several attempts have been made to reorganize it, and in hopes of pushing the matter a bid for the State convention was made. There is little doubt but what the efforts to revive the organization will be succussful.

Officers for the ensuing year were elected as follows:

H. F. Brandan, of Detroit, president; William E. Fuller, of Kalamazoo, vice-president; G. B. Richardson, of Pontiac, secretary; William Hayton, of Kalamazoo, treasurer; George Turnbull, of Flint, conductor, and F. T. Buch, of Lansing, doorkeeper. William Moore, of Detroit, was elected trustee for three years, and F. W. Turner, of Owasso, for two years. A. B. La Franier, of Bay City, was elected State deputy.

Firms having representatives at the convention were:

Adam Cook's Sons Company, Advance Pump and Compressor Company, American Oil Company, Anchor Packing Company, V. D. Anderson Company, Arrow Boiler Compound Company, A. M. Byers Company, Carbox Company, S. P. Conkling Company, Crandall Packing Company, Garlock Packing Company, Greene, Tweed & Co., Hawkeye Boiler Compound Company, Hills, McCanna Pump Company, Holten-Weatherwax Company, International Correspondence Schools, Jenkins Brothers, Keasby-Mattison Company, Kenneth Anderson Manufacturing Company, Keystone Lubricating Company, L. R. Klose Electric and Manufacturing Company, Lake Erie Boiler Compound Company, Lunkenheimer Company, Lyons Boiler Works, Murphy Iron Works, National Vacuum Heating Company, New Era Manufacturing Company, Ohio Blower Company. J. H. Parsons Chemical Company, William Powell Company, POWER AND THE ENGINEETR, Quaker City Rubber Company, Quinn Manufacturing Company, F. Raniville Company, J. A. Rae Company, Schaeffer & Budenberg Company, Standard Oil Company, Steam Appliance Company, Strong, Carlisle & Hammond, Under Feed Stoker Company of America, United States Graphite Company, Westinghouse Machine Company, Wheeler Blaney Company, Wickes Boiler Company, Williams Gauge Company, J. T. Wing Company and C. C. Wormer Machinery Company.

SOCIETY NOTES

The next convention of the Oil Mill Superintendents will be held at Galveston, Tex., and not at Houston, as reported in our issue of July 12.

NEW PUBLICATIONS

LAW AND BUSINESS OF ENGINEERING AND CONTRACTING. By Charles Evan Fowler. McGraw-Hill Book Company, New York. Cloth; 162 pages, 5½x9 inches. Price, $2.50 net.

In the first chapter the author discusses the relation which should exist between the consulting engineer and the contractor. The second and third chapters deal with the ordinary forms of contracts and specifications. The two succeeding chapters take up the special forms of these documents. The chapters dealing with estimating, organizing and inspecting contract work contain numerous suggestions, in the form of actual illustrations, for estimate sheets, cost cards and inspectors report blanks. The concluding chapter deals with the essentials of contract law. The requirements of a valid contract are clearly defined and the various kinds of contracts are enumerated.

The matter contained in this book is presented in a clear, concise and logical manner. The book constitutes a handy reference work and guide for consulting engineers, builders, contractors, inspectors and all others having to do with the business end of engineering and contracting.

CORROSION AND PRESERVATION OF IRON AND STEEL. By A. S. Cushman and H. A. Gardner. McGraw-Hill Publishing Company, New York. 373 pages; 6x9 inches; cloth; illustrated. Price, $4 net.

So much has been written upon the corrosion of iron and steel that there remains but little new material to be added. The present work, however, is based upon such authoritative information as has appeared in various bulletins from time to time, and is supplemented by original investigations by the authors. Moreover, the subject is dealt with in a concise and logical manner, and the addition of a complete set of illustrations serves to fix the matter more firmly in the reader's mind.

The treatment is based upon the "electrolytic theory," for the proper understanding of which chapters upon the "theory of solutions" and the "application of the electrolytic theory" have been incorporated. The inhibition of corrosion by the action of various chemical solutions is fully discussed, and it is shown that some substances supposed to prevent corrosion actually stimulate it. The proper method of applying the various protective coatings is cited, together with the results of a number of tests; the chapter upon paints being particularly instructive. A complete bibliography appended to the text adds greatly to its usefulness.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. Harry Eason, New York, N. Y., assignor, by direct and mesne assignments, to Eason Engine Company, a Corporation of Pennsylvania. 963,880.

GASOLENE ENGINE. Chester Charles Jones, Beatrice, Neb. 963,898.

STEAM TURBINE. Isaac Oesterblom, New York, N. Y., assignor of one-half to Harry P. Wood, New York, N. Y. 963,927.

MOTOR WORKED BY EXPLOSIVE MIXTURE. Antoine Lavoix, Paris, France, assignor to Société Francaise de Constructions Mecaniques (Anciens Establissements Cail), Denain, France. 964,030.

ROTARY ENGINE. James W. Keating and Allan McDonald, Portland, Ore. 964,431.

BOILERS, FURNACES AND GAS PRODUCERS

BOILER. Harry S. Martin, Dunkirk, N. Y., assignor to United States Radiator Company, Dunkirk, N. Y., a Corporation of New York. 963,915.

SMOKE-CONSUMING FURNACE. John S. Smith, St. Louis, Mo. 963,953.

HYDROCARBON BURNER. Augustus C. Williams, Brookfield, Mo. 963,969.

HYDROCARBON BURNER. William Newton Best, Jr., New York, N. Y., assignor of three-fourths to William Newton Best, Sr., Brooklyn, N. Y. 963,985.

STEAM BOILER FURNACE. Herman A. Poppenhusen, Evanston, and Joseph Harrington, Riverside, Ill. 964,053.

LIQUID HYROCARBON-BURNING APPARATUS. Louis K. Leary, Los Angeles, Cal. 964,031.

CRUDE-OIL BURNER. Ernest F. Stanton, Wellington, Tex. 954,457.

POWER PLANT AUXILIARIES AND APPLIANCES

TWO-STAGE PUMP. William T. Gray, El Campo, Tex., assignor to El Campo Machine Company, a Corporation of Texas. 963,756.

AIR COMPRESSOR. Frank Heath Merrill, Plainfield, N. J., assignor, by mesne assignments, to Gas Engine Pneumatic Company, Plainfield, N. J., a Corporation of New Jersey. 963,788.

COMBINED GOVERNOR AND REGULATOR FOR AIR COMPRESSORS. John L. Osborne, Erie, Penn. 963,803.

CARBURETER. John Peterson, Chicago, Ill., assignor to Dibelka & Breska, Chicago, Ill., a Copartnership. 963,804.

GOVERNING MECHANISM FOR ELASTIC FLUID TURBINES. Frederick Samuelson, Rugby, England, assignor to General Electric Company, a Corporation of New York. 963,811.

MUFFLER FOR EXPLOSIVE ENGINES. De Wane B. Smith, Deerfield, N. Y. 963,822.

VALVE. Frank F. Varlie, Marianna, Ark. 963,836.

VALVE AND COCK. Alfred Williams, Streatham, London, England, assignor to Charles Henry Howe, London, England. 963,842.

TURBINE-CONTROLLING DEVICE. Raymond N. Ehrhart, Pittsburg, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 963,881.

CARBURETER. John R. Mack, New York, N. Y., assignor to John R. Locke, Brooklyn, N. Y. 963,914.

AIR PUMP. George McKerahan, Altoona, Penn. 963,923.

IGNITION APPARATUS. Arthur E. England, Boston, Mass., assignor to Arthur A. Maxwell, trustee, Worcester, Mass. 964,007.

SPARK PLUG. Irwin Floyd Kepler, Akron, Ohio. 964,028.

LUBRICATOR. Guy E. Ledgerwood, Hannibal, Mo. 964,032.

SEAM LOCK FOR TUBES. William H. McNutt, New York, N. Y., assignor, by mesne assignments to the McNutt Non-Explosive Manufacturing Company, New York, N. Y. 964,048.

BELT SHIFTER AND TIGHTENER. Hiram A. Perkins, Rochester, N. Y., assignor to American Wood Working Machinery Company, Rochester, N. Y., a Corporation of Pennsylvania. 964,051.

PIPE JOINT. August T. Pelugh, Hoboken, N. J. 964,052.

STARTING MECHANISM FOR INTERNAL COMBUSTION ENGINES. Lloyd Yost, St. Marys, Ohio. 964,138.

CARBURETER. John M. Kelley, Rochester, N. Y. 964,165.

TRIP VALVE. Adalbert W. Fischer, Philadelphia, Penn., assignor to Schutte & Koerting Manufacturing Company, Philadelphia, Penn., a Corporation of Pennsylvania. 960,872.

OIL CAN. Jacob Gullong, Holyoke, Mass. 960,898.

PIPE COUPLING. Clinton Guyer, Muncy, Penn., assignor of one-half to William C. Crawford, Muncy, Penn. 960,899.

PRESSURE GAGE ALARM. Charles D. Hill, Clifton Springs, N. Y. 960,920.

WATER GAGE. Carl Wilhelm Johnson, Brockwayville, and Charles Merridith Rathke, Pittsburg, Penn. 960,941.

STEAM, AIR AND WATER TRAP VALVE. Fred W. Leuthesser, Chicago, Ill. 960,972.

AIR PUMP FOR USE WITH CONDENSERS AND THE LIKE. Philip Francis Oddie, London, England. 961,000.

STEAM BOILER GAGE. William T. Piper, North Andover, Mass. 961,008.

STEAM SEPARATOR. Henry W. Spencer, London, England. 961,037.

VALVE MECHANISM FOR CORLISS ENGINES. Benjamin T. Allen, Harrisburg, Penn., assignor to Harrisburg Foundry and Machine Works, Harrisburg, Penn., a Corporation of Pennsylvania. 964,194.

PIPE COUPLING. William H. Goss, Stonington, Me. 964,238.

VALVE AND VALVE ACTUATING ARRANGEMENT FOR INTERNAL COMBUSTION ENGINES. Samuel Isaac Parker, Teddington, England. 964,308.

GAGE COCK. John Stockwell, Marine City, Mich. 964,329.

CARBURETER. James Clendenin Eckert, Stamford, Conn., administrator of Charles Fox, deceased. 964,409.

HEATER FOR CARBURETERS. James Clendenin Eckert, Stamford, Conn., administrator of Charles Fox, deceased. 964,410.

LUBRICATOR. Albert J. Oaks, South Park, Penn. 964,450.

VALVE. Samuel M. Guss, Reading, Penn., assignor of one-half to Herbert S. Guss, Reading, Penn.; Florence E. Guss, administratrix of said Samuel M. Guss, deceased. 964,468.

ELECTRICAL INVENTIONS AND APPLIANCES

ELECTRIC ARC LAMP. Thomas Edgar Adams, Cleveland, Ohio, assignor to the Adams-Bagnall Electric Company, Cleveland, Ohio. 963,736.

ELECTRIC THERMOSTAT. Fred W. Harris, Wilkinsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 963,763.

GALVANIC BATTERY. Stephan Benko, Budapest, Austria-Hungary, assignor to Company Limited for Exploitation of Inventions by Stephan Benko, Budapest, Austria-Hungary. 963,852.

ELECTRIC SWITCH. John L. Creveling, New York, N. Y., assignor to Safety Car Heating and Lighting Company, a Corporation of New Jersey. 963,996.

ELECTRICITY - MEASURING INSTRUMENT. Johannes Gorner, Eschersheim, near Frankfort-on-the-Main, Germany, assignor to the firm of Hartmann & Braun Aktiengesellschaft, Bockenheim, Frankfort-on-the-Main, Germany. 964,015.

CURRENT INTERRUPTING DEVICE. Oscar M. Leich, Genoa, Ill., assignor to Cre-craft-Leich Electric Company, Genoa, Ill. 964,169.

PORTABLE ELECTRICAL TOOL. Frank J. Backscheider, Cincinnati, Ohio. 964,197.

POWER PLANT TOOLS

PIPE COUPLING. Frederick W. Reed, Trafford, Penn. 964,315.

ROTARY FIELD MAGNET. Emil Volkers, Berlin, Germany. 964,343.

PIPE TONGS. George A. Lane, Florence, Cal., assignor of one-half to Edward C. Kellermeyer, Taft, Cal. 963,903.

ATTACHMENT FOR MONKEY WRENCHES. Frank Fisher, Reardan, Wash. 964,010.

WRENCH. Andrew S. Steen, Adams, N. D. 964,067.

WRENCH. James G. Portteus, Marion, Ind., assignor of one-half to Theodore Portteus, Indianapolis, Ind. 964,313.

WRENCH. George H. Watson, Marshall, Mich. 964,346.

LIFTING JACK. Erastus L. Ceddington, Vanburen, Ind., assignor of one-half to Charles M. Stuart, Marion, Ind. 964,394.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer in Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

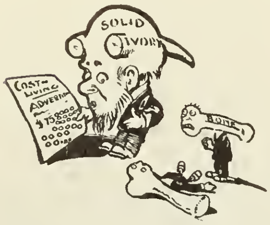
Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

The Senate Committee has gone and done it!

It has investigated and reported on the reasons for the "increased cost of living"—

And it has decided that one reason is *Advertising!*



We suppose their line of argument (if they had one) was that if John Jones makes shoes and spends $1000 advertising them, he must charge enough more for his footwear to get back that $1000.

Which is perfectly true—*but* the worthy Senators who make our laws luckily do *not* make our shoes.

If they did perhaps they would know that there are two distinct items of cost to be met before the consumer gets his boots.

One of these is the cost of manufacture—the other the cost of distribution—

And *advertising* is the cheapest distributor yet discovered.

Therefore, if John Jones had not disposed of his shoes by spending $1000 in advertising them, he would have had to spend much more on other methods of distribution.

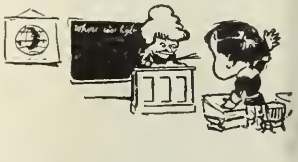
This would seem to prove that John Jones is a better business man than are the members of the Senate Committee.

In these days of keen competition is it reasonable to suppose that the business department of any manufacturing concern would use any but the cheapest possible method of distributing its product?

Advertising has reduced the cost of getting goods from the manufacturer to the consumer—and more—

It has made possible a *lower cost of manufacture* because it creates a demand for the goods, which means a larger production, which in turn means a lower cost because the more goods a man makes the cheaper he can make them.

Thus with Advertising *reducing* the cost of *distribution*, as well as the cost of *manufacture*, it is hard to see how even a Senate Committee could lay part of the "increased cost of living" at the door of Advertising.



We fear the Senators have followed about the same line of reasoning as the school boy who said, "Light comes from the sun—feathers are light—therefore feathers come from the sun."

The point is, Advertising has become a great force in modern commerce.

It has kept the cost of manufactured goods lower than it could possibly have been otherwise.

It educates buyers to spend their money wisely and guarantees that the consumer will get what he pays for.

The advertising pages of the world's periodicals are now and ever shall be the great, universal marketplace, where the "ultimate consumer" knows he will get a square deal.

Because Advertising is one of the newer sciences, only a comparatively few experts have known what it is all about. The average layman has considered it simply an expense.

But now it has become a recognized, necessary part of every manufacturing business—the *most* important part of any selling system—

Simply because it has proved to be the best and cheapest method of selling—the Senate Committee to the contrary notwithstanding.

Power and The Engineer

Vol. 32 NEW YORK, AUGUST 9, 1910 No. 32

THERE are several factors that enter into the successful operation of steam power plants, and not the least of these is the fireman.

There may be steam-jacketed cylinders with infinitesimal clearances and valves, and pistons as tight as the proverbial bottle.

Contraflo condensers equipped with air pumps of miraculous efficiency may produce vacuums rivaling the comet's tail in tenuity.

There may be boilers and economizers so correctly proportioned that the escape of a single heat unit is almost as improbable as the discovery of an honest professional politician.

There may be water-weighing apparatus so accurately adjusted that it could on a pinch be used for a barometer, and the walls may be decorated with flue-gas analyzing apparatus that resembles a chemical laboratory in its complicated completeness.

There may be continuous and intermittent steam-engine indicators, recording thermometers and pressure gages.

Superheaters may furnish steam so dry that its own weight of water will scarcely moisten it, and besides there may be a chief engineer worth a congressman's salary.

All these things and countless others that go to make the finished modern power plant may be there and under competent supervision.

But if the athletic, intelligent, ridiculously underpaid worker who is capable of delivering the largest percentage of the potential energy in the coal to the rim of the flywheel of the engine—the real fireman—is not there, all the rest is "sounding brass."

Coal per kilowatt-hour, coal per car-mile, and pounds of water per pound of coal are of frequent enough occurrence in power-plant reports to show that a part of the expense of power-plant operation is measured in coal units, and the man who handles these units is the most important factor in the steam plant.

The most highly economical engines in the world will not deliver low-priced power if supplied with costly steam, but a very ordinary equipment will do fairly well if the man behind the shovel knows his business.

If he knows that a heavy draft needs a thick fire, if he realizes the difference between heavy firing at long intervals and light firing at short ones, if he is aware that fresh coal should only be spread on an incandescent fuel bed, he knows enough of the other details of economical steam making to be the most valuable man about the plant if he so desires.

If such a desire exists, and the man feels the responsibility of his position and lives up to it, recognition in terms of dollars and cents should be immediate and commensurate with the service rendered.

Intelligent power-plant managers appreciate the value of labor-saving and labor-displacing appliances, and as much as possible the human element is being eliminated from all power-plant problems.

But as long as the burning of coal for power production continues, just so long will the human element be indispensable in the boiler room, and the best is none too good.

"Common good has common price; exceeding good, exceeding."

Dimensions of Steam Turbines

By Charles A. Edmonds

> *The preliminary calculations, forming the basis of all steam turbine design, are discussed and current practice in dimensioning the various parts is cited.*

In designing a steam turbine, the first and most essential step is to estimate the highest suitable speed of rotation in order that the turbine may be made as small as possible for the required efficiency. While there is now no great difficulty in accurately determining the proportions of turbines to insure certain results, with ample confidence to enable a stiff guarantee being made both for speed and economy, the principal dimensions involve considerable calculation.

Although Fig. 2 illustrates a Parsons' marine type of turbine, the method of calculation of the proper dimensions is carried out in exactly the same manner as for a land turbine, except that the most economical speed of the propeller has to be considered first of all. The Parsons' marine type of turbine differs from the land type in that it has no dummy cylinders to counteract the end thrust of the shaft, caused by the expansion of the steam issuing from each set of blades. These dummy cylinders *P P P*, Fig. 1, are not needed in the marine turbine because the end thrust of the turbine is taken up by the thrust of the propeller which is on the same shaft.

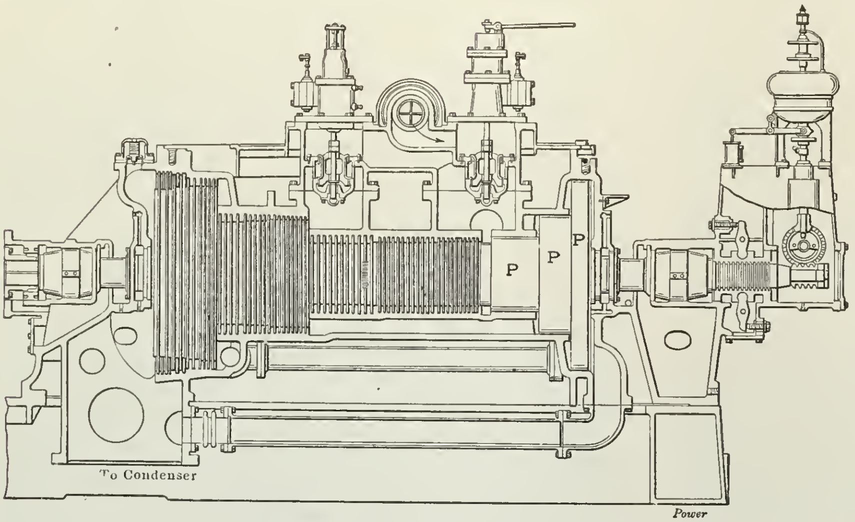


Fig. 1. Longitudinal Section through Typical Parsons Turbine

Roughly, the weight of turbines will vary inversely as the square of the revolutions, while the economy will remain almost constant if the turbines are designed for the same internal conditions.

The laws governing the best theoretical velocity of the steam and the blades in impulse turbines are similar to those for water turbines, but in practice some modification is necessary, and the best ratio of blade speed and steam speed is still a matter of opinion. The ideal condition for impulse turbines occurs when the peripheral velocity of the buckets is one-half that of the jet, or in reaction turbines when it is equal to it.

Parsons' turbines have been built with V_t, Fig. 3, varying from 0.25 to 0.85 of V_s, where V_t represents blade velocity at mean diameter, and V_s the steam speed across the row in question. A very usual ratio for large units in electrical work has been

$$\frac{V_t}{V_s} = 0.6$$

These ratios require very careful calculation. The steam consumption must be accurately known in order to proportion them correctly throughout the turbine, and the necessity of having the same area of openings in so many rows while the steam volume increases so rapidly adds to the difficulty of close calculation.

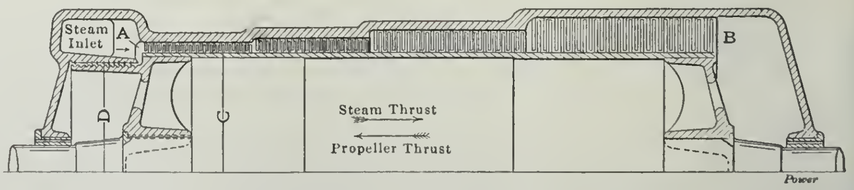


Fig. 2. High-pressure Cylinder of Parsons Turbine

The potential energy of the steam, corresponding to the "head" in water turbines, can easily be calculated for given pressure differences.

The well known expression connecting energy and velocity is

$$E = \frac{W V^2}{2 g}$$

where

E = Energy,
W = Weight,
V = Velocity,
g = Acceleration due to gravity.

Taking the weight W as one pound of steam, expressing the velocity V and the acceleration due to gravity g in feet per second, and the energy E in foot-pounds, $g = 32.16$ and $E = 778 \times B.t.u.$ utilized. Then,

$$V^2 = 2 \times 32.16 \times 778 \times B.t.u.$$

and

$$V = \sqrt{50{,}040.96\, B.t.u.} = 223.81 \sqrt{B.t.u.}$$

It is obvious that the speed ratio between jet and vane must affect the number of stages and that with a given turbine velocity the greater the ratio of V_t to V_s the greater will be the required number of rows. To obtain a small value for V_s necessary to produce a large value for $\frac{V_t}{V_s}$, a small pressure drop per row is necessary.

The mean diameter is a dimension capable of wide variation without affecting the efficiency, provided that the number of rows is correct.

Calling this mean diameter in inches d, the blade velocity in feet per second will be

$$V_t = \frac{d \times \pi \times r.p.m.}{12 \times 60}$$

and assuming a blade velocity from experience the diameter can be determined.

$$d = \frac{12 \times 60 \times V_t}{\pi \times r.p.m.} = \frac{229 \times V_t}{r\ p.m.}$$

Hence: *To find the mean diameter of the rows of buckets, multiply the bucket velocity in feet per second by 229 and divide by the revolutions per minute.*

To arrive at the corresponding number of rows, the revolutions being given, the ratio of V_t to V_s must be settled, from which the steam speed can be obtained. It is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis. The number of rows N on one diameter can be found by work-

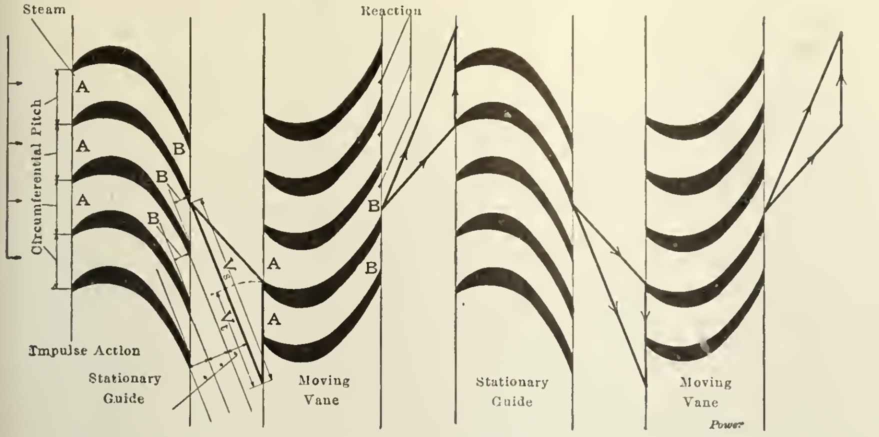


FIG. 3. ACTION OF THE STEAM IN THE PARSONS TURBINE

V_t = Blade velocity at mean diameter.
V_s = Steam speed due to expansion between A and B.

ing out the British thermal units necessary to give a certain steam speed at each row; see Fig. 4. The available energy divided by the energy it is desired to abstract at each row will give the number of rows required. This result may be arrived at by various ways, but the principle involved is the same in each case. Numerous empirical coefficients for approximating steam speeds and the corresponding number of rows are obtainable from experience. To put all the rows on one diameter would involve an excessive length of turbine and also inconvenient blade hights. It is, therefore, usual to divide the rotor into three or more stages, which has the advantage of shortening the turbine and reducing the number of rows. If $\frac{1}{n}$ = the fraction of power developed in the first cylinder or barrel, $\frac{N}{n}$ = the number of rows in the first barrel and with the change of diameter and increase of blade velocity in the succeeding stages, the number of rows on other barrels is so altered as to keep, for equal powers and efficiencies,

$$(\textit{Blade Vel.})^2 \times \textit{No. of Rows} = \textit{Constant}$$

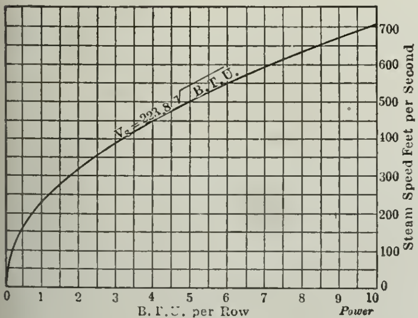


FIG. 4. VELOCITY DIAGRAM

The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and I think the Westinghouse Company, at Pittsburg, was the first to make a radical departure in this respect and adopt higher speeds. The maximum vane speed used for Parsons' blading is, as far as the author is aware, about 375 feet per second in the low-pressure blades, and 170 in the high-pressure blades for turbines driving electric generators. To some extent blade speed is governed by blade hight. The speed should be so modified that this may be at least 3 per cent. of the mean diameter to reduce the proportion of clearance losses. Leakage over the tips of the blades is perhaps not so detrimental on account of actual loss as in its superheating effect on the steam between the row past which it leaks and the last row, because the reheating effect upsets calculations regarding openings by increasing the steam volume and thereby effects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is made to resist distortion due to heat, the less may be the clearance. A clearance diagram based upon measurements of a large number of machines is given in Fig. 5.

TABLE 1. VARIOUS VANE SPEEDS.

Normal Output of Turbine.	Peripheralvane Speed		No. of Rows.	Revolutions per Minute.
	First Expansion.	Last Expansion.		
5000 kw...	135	330	70	750
3500 kw...	138	280	75	1200
2500 kw...	125	300	84	1360
1500 kw...	125	360	72	1500
1000 kw...	125	250	80	1800
750 kw...	125	260	77	2000
500 kw...	120	285	60	3000
250 kw...	100	210	72	3000
75 kw...	100	200	48	4000

Replying to the discussion on his paper before the Institute of Naval Architects in 1903, Mr. Parsons said that, "for all practical purposes, while the steam is traversing each set of blades, it behaves like an incompressible fluid, just as water would do, the expansion being very small at each set. The frictional losses and the eddy-making losses are practically identical, within small limits, with that of water; and the actual forces are in proportion to the density of the medium. In the turbine blades themselves, the efficiency is between 70 and 80 per cent."

This hydraulic analogy enables us to calculate the number of stages required in a different manner. The "equivalent head" due to the steam pressure, together with that at each row necessary to give the required velocity, may be found, and from these both the number of stages and the coefficient of expansion at each stage may be worked out.

In all types of turbines, Parsons, Rateau, Curtis, etc., a certain ratio must be maintained between the blade velocity and steam velocity, and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by a high rotary speed or by large diameters, or both. As the weight increases very rapidly with the diameter, and especially with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter. This should be done in cases where the weight increases more rapidly in inverse proportion to the rotative speed and the diameter, as in the Rateau or Zoelly types.

Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The hight of blade depends upon the volume of the steam and the speed at which it is to flow; also upon the ratio of the area of exit openings between the blades to that of the annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area for the steam to pass through being equal to volume divided by velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for ½, etc.) then

$$\textit{Hight of blade in inches} = \frac{\textit{Clear area in square inches} \times 3}{\textit{Mean circumference in inches}}$$

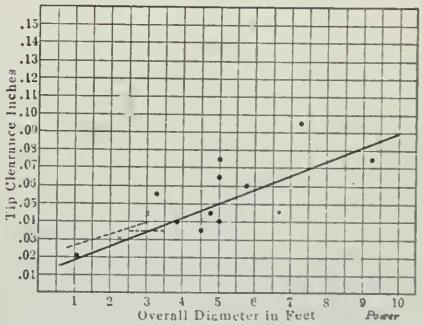
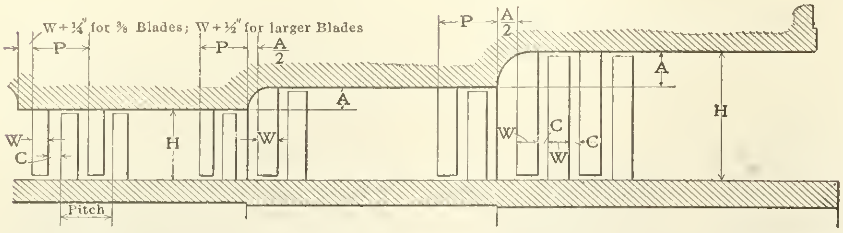


FIG. 5. CLEARANCE DIAGRAM

The ratio of blade hight to mean diameter should not be less than 3 per cent., or more than 15 per cent., because in the former the percentage of leakage will be excessive, and in the latter the bending moment and the radial divergence of the blades become too great. The width of blade, the shape of section adopted and the circumferential pitch, are standard considerations, and affect the factor 3, given above. It is not proposed to enlarge upon them in this paper. It may, however, be remarked that for $\frac{V_t}{V_s}$ greater than 0.6 the usual shape of Parsons' section, as shown in Fig. 2, should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of sufficiently strengthening the blade, the metal must be placed nearer the exit edge, thus increasing the angle between the face and the back of the exit edge of the blades, and giving an inferior shape of opening compared with that obtainable with a blade section adapted to ratios under 0.6.

Table 2 Standard Blade Dimensions



Height (H)	1"	2"	3"	4"	6"	8"	10"	12"	15"	18"	21"	24"	30"
Width (W)	⅜"	⅜"	⅜"	½"	½"	⅝"	⅝"	¾"	¾"	1"	1"	1⅛"	1¼"
Pitch (P)	1⅛"	1⅛"	1¼"	1⅝"	1¾"	2⅛"	2¼"	2½"	2⅝"	3⅛"	3¼"	3⅝"	4"
Axial Clearance (C)	3/16"	3/16"	¼"	5/16"	⅜"	7/16"	½"	½"	9/16"	9/16"	⅝"	11/16"	¾"

Power

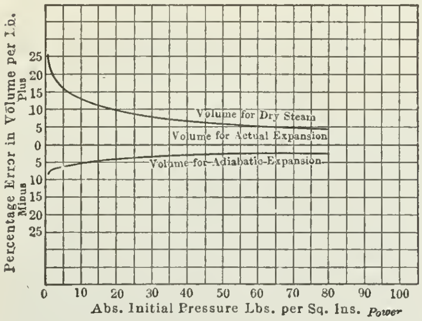


Fig. 6. Error Due to Incorrect Estimate of Steam Conditions

In Table 2 can be found a list of approximate widths for a given hight, and axial spacing of the rows. While this must be kept down to reduce the length of drum, it must be sufficient to allow for some play in overhauling, and sufficient clearance can be allowed here without affecting the economy. The openings between the blades to allow for the passage of the steam are very important, and must be carefully designed. The actual volume of the steam, not the volume per pound, as found in steam tables, or the volume due to adiabatic expansion, but the exact volume per pound at any point along the turbine must be determined in order to arrive at the desired adjustment of velocities. It is extremely doubtful whether the present blading arrangements give the best results. Greater accuracy of calculation and consequently improved pressure distribution and efficiency seem likely to follow the use of a more mechanical blading construction.

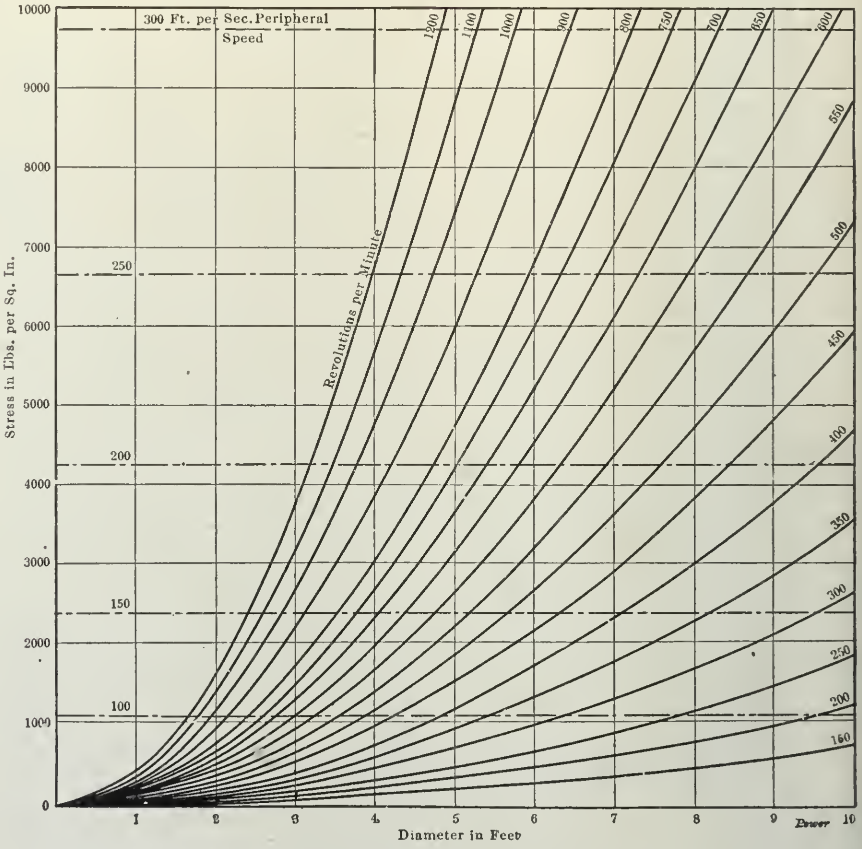


Fig. 7. Stresses Due to Various Speeds

Fig. 6 shows the percentage of error involved in using either the dry volume, or that due to adiabatic expansion, compared with the correct volume corresponding to the actual expansion in a large turbine using dry saturated steam at the first row of blades and 27 inches vacuum. Attention must be paid to the effect of approximate adiabatic expansion and the consequent moisture in the steam.

For convenience in manufacturing, as well as to allow for the expansion of the steam, the blade hights are stepped up, but no rule exists for this. The blades might be of 1, or of 65 hights, provided the blade openings were correct. It is, however, convenient to step them, as in Fig. 2, say, in 8 steps of 8 rows each, or 4 steps of 16 rows; even 9 steps of 7 rows would do to avoid any great variation from the annular area factor 3, as shown above.

To obtain hights and areas, it is best to plot graphically, volumes, steam speeds and the clear areas required. The use of standard blade hights will then enable the number of stages and rows per stage to be determined, and wide differences can be made in any arrangement without materially affecting the economy. The best arrangement is largely a matter of convenience and experience.

The material of which blades are usually made is a mixture of cheap brass containing about 16 parts of copper and 3 parts of tin. Alloys containing zinc are extremely unreliable for high temperatures, but blades containing about

98 per cent. of copper have been found very satisfactory for use with high superheats. More recently a material containing about 80 per cent. of copper and 20 per cent. of nickel has been adopted, and this is undoubtedly the best blading material in existence. Steel blading, drawn in the same way as the usual brass section, has been used in the United States with fairly good results. The process of drawing turbine blades gives an extremely tough skin to the metal, not only increasing the tensile strength, but greatly decreasing the chances of erosion.

It seems probable that the usual calking piece now adopted will be discarded in favor of a machine-divided strip into which the blades may be fitted, and instead of the slotting, wiring, lacing and soldering process at the tip, a similar machine-divided shroud will be used, giving a far stronger construction, and enabling finer clearances and better workmanship to be obtained, at the same time considerably reducing the cost of manufacture and the risk of blade stripping.

The chief causes of the latter may be set down to bad workmanship in setting the blades, defective blade material, excessive cylinder distortion (this is probably the most fruitful cause, and is a serious one, being due to bad design), whipping of turbine spindles (which is also due to bad design, or bad balancing), wear of bearings (which is very remote), and the introduction of extraneous substances such as water or grit. In fact, blade stripping may be said to occur generally from preventable causes. Small vibrations of very high frequency occasionally set up an action in certain rows of responsive length and fatigue the blade material, causing the loss of blades without any fouling at all.

In turbines driving electric generators, where the end thrust must be eliminated by the use of balancing pistons, the spindle is in tension when the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff, as well as strong enough, as any sag in the spindle will destroy the clearance. As will be seen from Fig. 7, the stresses due to centrifugal force are very low in the Parsons turbine, and except in occasional low-pressure barrels they do not exceed 7500 pounds per square inch, while at the high-pressure end they are usually under 2000.

The pressure on the bearings in a turbine is due only to the weight of the spindle. It may be taken from 80 to 90 pounds per square inch as long as the rubbing velocity does not exceed 30 feet per second. If it does, the pressure must be reduced so that the product of pressure and velocity does not exceed 2500 to 2700. In land work 50 pounds multiplied by 50 feet is very common. The friction heat of the bearings added to that due to conduction through the pedestals, necessitates the use of large oil coolers, and in the case of very high temperatures, of special kinds of oil. If possible, the bearing temperatures should not exceed 140 to 150 degrees Fahrenheit, though the writer has known of 190 degrees being used without trouble.

Space does not permit of more than passing reference to cylinders, but it would be difficult to exaggerate the importance of very careful design in this connection. Cylinders, with heavy flanges on the center line, distort in a very curious fashion when heated with their axes horizontal, and measurements taken of a hot cylinder on a surface plate with micrometer gages reveal some very remarkable facts. When working, the temperature along the cylinder falls from 400 to 100 degrees Fahrenheit in a distance of 6 or 8 feet, and unlike the reciprocating engine, this remains constant. The radial expansion is consequently more at one end than the other, while at any point along the turbine the tendency is to expand less at the flanges than at the top and bottom. For this reason ample clearance must be allowed; exactly what this will be when the spindle and cylinder are hot is hard to say, but it seems most likely that the total clearance area will differ but little from what it is when cold.

The longitudinal expansion when hot is often very marked, and in all turbines, necessitates provision for the resultant movement at one end.

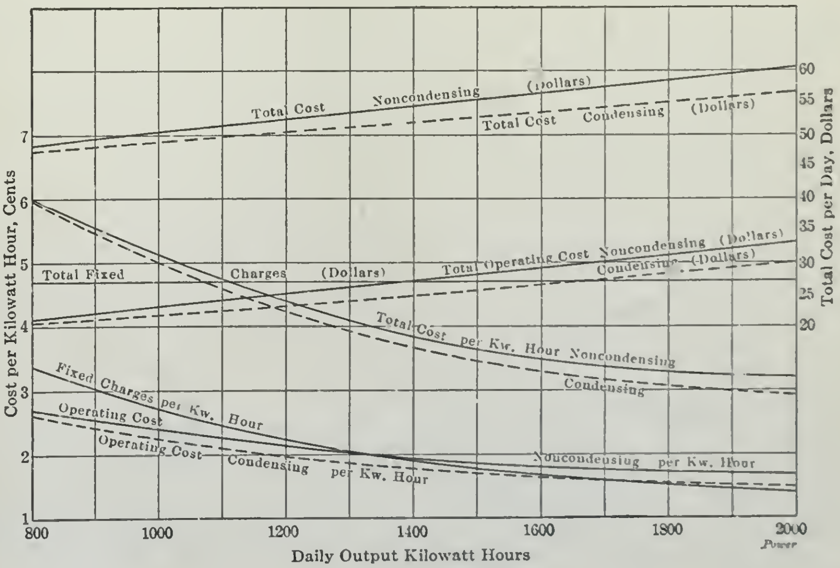
In the course of operation, where no superheaters are used, there is a distinct tendency for the turbine to be supplied with wet steam, the effect of which is very marked on the economy. Experiments that have been made show that the percentage increase in consumption is about twice that of the moisture in the steam. With 2 per cent. of moisture in the steam at the first row, the consumption is increased about 4 per cent.

Data on Central Station Operation

The recent report upon the operation of the new power plant at the coast artillery school, Fort Monroe, for the first three months of this year, contains some interesting figures on central-station operation.

This plant furnishes electrical energy for the interior and exterior lighting of the post (exclusive of fortifications), power for testing and shop purposes, live steam at 125 pounds for the engines in the laboratories, and either live or exhaust steam for heating the entire artillery school.

The daily output of the plant ranges from 800 to 2000 kilowatt-hours. The accompanying chart shows the total cost per kilowatt-hour based upon operating costs, fixed charges, interest and depreciation. The total cost of the installation is estimated at $42,209.55, the interest being placed at 3 per cent., and the depreciation at 7 per cent. The coal used was anthracite and cost $4.80 per ton of 2000 pounds.



Costs for Power at Various Outputs

Flue Gas Analysis and Calculations

By Julian C. Smallwood

The author briefly discusses the molecular theory, shows how to compute the amount of air required to burn a pound of carbon and presents calculations for determining the heat losses due to excess and insufficient air.

A few elementary conceptions are necessary to make calculations from a flue-gas analysis. One, that of the theory of the molecular structure of matter is commonly known, but it may be well to review it here. The smallest subdivision of any material which can exist by itself and yet retain all of the characteristics of the material is called a molecule. A molecule is composed of a number of so called atoms which are supposed to be the smallest subdivisions of matter. Any chemical compound is dependent for its identity upon the material of the atoms forming its molecules. Thus a molecule of CO_2 is composed of one atom of carbon and two of oxygen, while a molecule of CO has one atom of both carbon and oxygen. The molecule of the gas oxygen contains two atoms of oxygen. We may form a conception of these facts by imagining a pail of lead shot to be atoms of oxygen and one of iron shot to be carbon. If we stick a single lead shot to each iron one the result represents carbon monoxide. Similarly, oxygen gas would be represented by the lead shot stuck together in pairs; carbon by the iron shot in its single form, and carbon dioxide by one iron shot combined with two of lead.

The relative weights of atoms are definitely known. That of the gas hydrogen is the least; an atom of carbon is twelve times as heavy as one of hydrogen; one of oxygen sixteen times as heavy; and nitrogen, fourteen. We may, therefore, let the weight of an atom of hydrogen be a unit; a molecule of carbon then weighs 12 of these units; a molecule of oxygen weighs 32, since there are two atoms to the molecule; and the weight of a molecule of CO_2 is,

$$12 + 16 + 16 = 44.$$

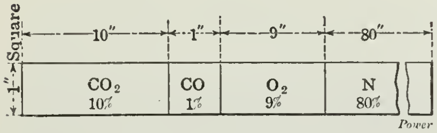


FIG. 1. PROPORTIONS OF THE GASES

Tabulating the molecular weights of flue-gas constituents we have,

$$\begin{aligned} CO_2 &= 12 + 16 + 16 = 44 \\ O_2 &= 16 + 16 = 32 \\ CO &= 12 + 16 = 28 \\ N &= 14 \end{aligned}$$

Each value means that a molecule of the gas referred to is equal in weight to that number of atoms of hydrogen. For convenience we may call the weight of the latter simply a "weight." Then CO_2 has 44 weights; O_2, 32, etc.

Now, since the gas analysis has furnished volumes and not weights, this would not do us much good if it were not for a very convenient physical law. This law states that in a given volume of any gas at a given pressure and temperature there is always the same number of molecules present.* We therefore know that in a vessel containing a mixture of gases there is the same number of molecules as though only one of those gases, or any gas, existed there alone, irrespective of the number of atoms in its molecules.

It follows that a volume of, say, two cubic inches contains twice the number of molecules that one cubic inch contains. Also, if the molecules of the larger volume are twice as heavy as those of the smaller, the weight of the larger volume is four times that of the smaller.

Fig. 1 represents a vessel, of one square inch cross-section and 100 inches long, containing flue gas. It is supposed that the constituents are separated in layers as indicated, the percentages by volume being CO_2, 10 per cent.; O_2, 9 per cent.; CO, 1 per cent.; and N, 80 per cent.

Let us suppose for the sake of argument that there is a single molecule of CO occupying the cubic inch shown. There will then be 10 molecules of CO_2, 9 of O_2, and 80 of N. The weight of the CO, as previously explained, is then 28 "weights"; that of the CO_2 is,

$$10 \times 44 = 440;$$

that of the O_2 is,

$$9 \times 32 = 288;$$

that of the N is,

$$80 \times 14 = 1120;$$

and the total weight is

$$28 + 440 + 288 + 1120 = 1876 \textit{ weights.}$$

The part by weight that the CO is of the whole is, therefore, $\frac{28}{1876}$; that of the CO_2 is $\frac{440}{1876}$, etc. It is thus clear that, knowing the parts of a flue-gas sample by volume, the parts by weight may be obtained simply by multiplying each of the former by the corresponding molecular weight and dividing by the sum of such products.

We have seen in the foregoing that the fractional part that any of the separate gases is of the entire weight may be found. Using the same method it may be determined what part any of the materials, such as oxygen or carbon, is of the whole or each other. Referring to Fig. 2, which is a rearrangement of Fig. 1, the nitrogen being omitted, the black circles represent atoms of carbon and the unshaded ones, atoms of oxygen. The figure represents a vessel each cubic inch of which contains one molecule of gas, the different gases being separated as shown. As before, the percentages by volume are 10 of CO_2, 9 of O_2 and 1 of CO. In each cubic inch of CO_2 there are two atoms of oxygen; in the cubic inch of CO there is one atom of oxygen; and in each cubic inch of O_2 there are two atoms of oxygen. Counting them, we have,

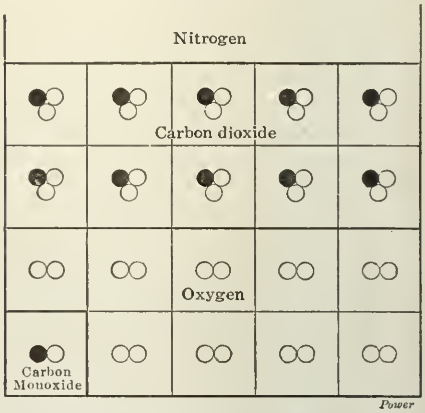


FIG. 2. DIAGRAM ILLUSTRATING COMBINATIONS OF ATOMS TO FORM MOLECULES

CO, 1 atom of oxygen.
CO_2, $2 \times 10 = 20$ atoms of oxygen.
O_2, $2 \times 9 = 18$ atoms of oxygen.
Total 39 atoms of oxygen.

Similarly, the number of atoms of carbon is,

CO 1 atom of carbon.
CO_2 10 atoms of carbon.
Total 11 atoms of carbon.

The total weight of oxygen is, therefore,

$$39 \times 16 = 624;$$

that of carbon is,

$$11 \times 12 = 132$$

weights. It thus appears that to obtain the weight of any of the materials it is only necessary to multiply the per cent. volume of each gas containing that material by the number of atoms of the material in one molecule of the gas; the sum of these gives the number of atoms of the material; multiplying this number by the atomic weight gives the result.

In the case cited the oxygen used to make the mixture is $\frac{624}{1876}$ of the whole.

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*Avogadro's law.

And, since there are 132 weights of carbon and 624 of oxygen, there are $\frac{624}{132}$ or of O_2 to each one of C.

he important calculations based analysis is that for the amount nished to the coal, expressed of air per pound of coal. This tained by the use of the prin- t given. The following ex- calculation:

e carbon in the coal, except part of it which escapes e gate, is converted into gas up the chimney. Also, all of in the flue gas comes from with the exception of a very of it which is introduced as he entering air. We may as- the oxygen existing in the flue oxygen and that in combina- the oxygen contained in the CO, equals in amount the pplied to the furnace with the air. e example given it was shown that there were about 4¾ weights of O_2 to each one of C actually existing in the chimney. According to our assumptions, then, there were 4¾ weights of O_2 actually furnished to the furnace for each weight of C burned. To produce this oxygen air must have been supplied equal in weight to,

$$4\tfrac{1}{3} \times 4\tfrac{3}{4} = 20.5,$$

since the weight of air is 4⅓ times the weight of the oxygen contained in it. The figure, 20.5, represents the weight of air furnished for each weight of carbon burned, and since the relation is between weights we may call them pounds. The result may therefore be expressed as 20.5 pounds of air per pound of carbon burned. Suppose that we had previously determined that the coal was composed of 90 per cent. carbon and 10 per cent. ash. We should require 10 per cent. less air to burn one pound of this coal than would be used to burn a pound of pure carbon. Hence,

$$20.5 \times 0.90 = 18.5$$

pounds of air required per pound of coal.

Now, this amount of air may be too much or too little to burn the coal properly. To determine the exact amount of air theoretically required for the complete combustion (that is, conversion to CO_2) of one pound of carbon, it is noted that one molecule of oxygen combines with one of carbon to make CO_2. Hence the weight of oxygen to each weight of carbon is $\frac{32}{12}$; and the number of pounds of air per pound of carbon is,

$$4\tfrac{1}{3} \times \tfrac{32}{12} = 11.5$$

It is found that unless more than the exact theoretical amount of air is supplied there will not be complete combustion; that is, part of the gas will be burned to CO instead of CO_2. It seems necessary, therefore, to furnish an excess of air and this excess may be fixed at 50 per cent. more than the theoretical amount. As before stated, the flue gases act as a vehicle for heat, and if there is more than the necessary excess of air, a greater loss of heat up the chimney results. Using the data of the example previously given, the weight of air used for each pound of carbon burned is 20.5. Assuming 50 per cent. excess of air to be the right amount, there should have been furnished

$$11\tfrac{1}{2} + \left(\tfrac{50}{100} \times 11\tfrac{1}{2}\right) = 17\tfrac{1}{4}$$

pounds. The difference between this and the air actually admitted is 3¼ pounds, and this is the unnecessary air admitted for the burning of each pound of carbon. The specific heat of air is 0.239, which means, as before explained, that if one pound of air is heated one degree of temperature, it has 0.239 B.t.u. added to it. Suppose that the excess air considered in the example is heated 500 degrees; that is, the temperature of the boiler room is, say, 70 degrees and that of the flue gas 570 degrees, the heat added to 3¼ pounds of air is then

$$3\tfrac{1}{4} \times 500 \times 0.239 = 388 \text{ B.t.u.}$$

Now, it has been found by experiment that a pound of carbon has a heat-value of 14,650 B.t.u. if completely burned to CO_2. The part of this heat which is carried up the stack by the 3¼ pounds of excess air is

$$\frac{388}{14{,}650} = 0.026, \text{ or } 2.6 \textit{ per cent.}$$

If it is desired to calculate the total heat lost up the stack (that is, the heat carried off not only by excess air but by *all* of the flue gases), an approximate value of the specific heat of these gases may be taken as 0.235. The weight of flue gas per pound of carbon must equal the sum of the weights of the air supplied and the one pound of carbon with which it combined or

$$20.5 + 1 = 21.5$$

pounds of flue gas per pound of carbon.

The heat added is therefore,

$$21.5 \times 500 \times 0.235 = 2500 \text{ B.t.u.}$$

The loss in per cent. is

$$\frac{2500}{14{,}650} \times 100 = 17 \textit{ per cent.}$$

It has been previously remarked that if CO is formed, what is called complete combustion does not take place, and not all of the heat possible is generated. If CO is the product of combustion, this product is capable of combining with more oxygen, which combination means further burning and which, since it is combustion, generates additional heat. When the carbon is changed entirely to CO_2, however, it is no longer capable of combining with oxygen and can give out no additional heat. As before stated, the heat value of a pound of pure carbon burned to CO_2 is 14,650 B.t.u. Similarly it has been determined experimentally that a pound of carbon burned to CO generates 4400 B.t.u. Clearly then, for each pound of carbon burned to CO instead of CO_2 the heat lost is

$$14{,}650 - 4400 = 10{,}250 \text{ B.t.u.}$$

Referring to the data of the example, the weight of carbon burned to CO is

$$1 \times 12 = 12$$

weights, since the volume of the CO is 1 per cent. There is, therefore, $\frac{12}{132}$ of a pound of carbon incompletely burned to each pound of carbon used. The heat lost through the incomplete combustion of this carbon is

$$\frac{12}{132} \times 10250 = 932 \textit{ B.t.u.}$$

the meaning of which is that for each pound of carbon burned there is a loss due to this cause of 932 B.t.u. Since it is possible to obtain 14,650 B.t.u. from one pound of carbon, the per cent. loss is

$$\frac{932}{14650} \times 100 = 6.4 \textit{ per cent.}$$

CONVENIENT FORM FOR FLUE-GAS CALCULATIONS.

1	2	3	4	5	6
	Per Cent., Volume.	In Each Flue Gas, the Total Number of Atoms of C.	of O.	Molecular Weight of Flue Gas.	Total Weight of Each Flue Gas.
CO_2	10	10	20	44	440
O_2	9	..	18	32	288
CO	1	1	1	28	28
N	80	..	..	14	1120
Totals	..	11	39	..	1876

Total weight of C = 11 × 12 = 132 weights.
Total weight of O_2 = 39 × 16 = 624 weights.
Total weight of C burned to CO = 1 × 12 = 12 weights.
Total weight of flue gases = 1876 weights.

When making calculations as in the foregoing, it facilitates the work to first make a table as shown herewith. Columns 1 and 2 give the results of the flue-gas analysis. Considering the line in which CO_2 appears, for example, the figure 10 under column 3 is obtained by multiplying the per cent. volume by the number of atoms of carbon in one molecule of CO_2. In column 4 the number of atoms of oxygen is obtained in the same way. The molecular weight 44 has been calculated previously; multiplying it by the per cent. volume gives the weight of CO_2. It will not be difficult to understand these figures if it is remembered that it is assumed that 1 per cent. of flue gas by volume contains one molecule and that therefore the number of molecules in 2 per cent. is two, in 3 per cent. is three, and so forth. The assumption that 1 per cent. contains one molecule is permissible because "per cent." applies to *any* quantity; the proportions are the same whether 100 cubic feet of gas are considered or 100 cubic inches.

In presenting the calculations in the way in which they are here given, it is the writer's purpose to enable the operator to get results with the use of an apparatus for flue-gas analysis only. Such measurements as may be obtained by it are the only ones necessary to the foregoing calculations, with the exception of that for heat lost up the chimney. To determine this loss it is necessary to know the stack temperature, which may be found by the use of a special device called a pyrometer. A thermometer of the ordinary construction would be destroyed by the high temperatures. In consideration of the fact that a very considerable loss may result from having the stack temperatures too high, it is advisable to have a pyrometer on hand that this loss may be determined and kept within reasonable limits.

The heat losses calculated in this article have been compared with the heat value of a pound of carbon. It must be explained, however, that this is not usually done, the heat value of the coal being used instead. Since the furnace uses coal and not pure carbon, it is more reasonable to base results upon its heat value than that of carbon. But to obtain the former quantity, a special determination must be made; and to use it, the coal must be analyzed. To avoid this extra labor, the calculations may be made as explained and the results will not vary materially from those obtained by the more correct method of calculation.

If a coal analysis is made it furnishes the per cent. of carbon in the coal. This being known, all the heat losses may be found for one pound of coal instead of carbon, as was done in our calculation for the pounds of air per pound of coal. These losses may then be expressed in per cents. of the heat value of the coal instead of carbon.

It should be noted, too, that there are some nitrogen and hydrogen in the coal which appear in the flue gas along with the carbon and the constituents of the air. The hydrogen may combine with some of the entering oxygen and form water (water being a chemical combination of these materials). The flue-gas analysis as described, however, takes no account of water. The effect of the hydrogen in the coal, then, is to make our calculation for the pounds of air per pound of coal, a trifle less than the actual amount due to the oxygen turned into water.

Coal Handling, Wood Worsted Mills

At the time of our description of the power plant of the Wood Worsted Mills at Lawrence, Mass., some three years ago, the coal was conveyed from the storage alongside the boiler room to the firing line by means of tip carts drawn by horses. With the doubling up of the boiler plant then in use it appeared that mechanical conveyance would reduce the cost, and the system shown in the accompanying engravings has been installed.

The substitution of mechanical conveyance for tip carts drawn by horses to carry the coal from storage to the firing line resulted in a big reduction in the cost of handling. The conveying apparatus has a capacity of forty tons per hour.

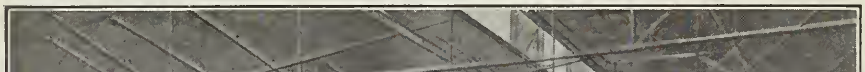


FIG. 1. COAL STORAGE SHED

Coal is brought into the storage shed, of which Fig. 1 is an inside view and Fig. 2 a plan and elevation, upon elevated spur tracks by the railroad. The trestles permit the use of side- or bottom-dumping cars, or cars which can be unloaded by the locomotive crane shown in the elevation of the storage shed in Fig. 2, and by a reproduced photograph in Fig. 5, by means of which the coal, however unloaded, may be trimmed and piled and worked over in any portion of the shed or yard.

Beneath three of the four tracks are hoppers, shown at *A A A*, in Fig. 2, and one in perspective in Fig. 3. When coal is wanted in the boiler room it is conveyed to one of these hoppers by the locomotive crane, the clam-shell bucket of which handles about a ton at a time and is discharged by the hopper upon a 48-inch apron conveyer, seen beneath the hopper discharge in Fig. 3 and indicated by *B* in Fig. 2, where it is shown extending under all three of the hoppers. This carries it to the 28x24-inch crusher, shown at *B* in the plan portion of Fig. 2, and better still in the elevation. A man watches the apron between the nearest hopper and the crusher and removes the coupling pins and other articles which frequently come along with the coal, and which the crusher might find it difficult to handle.

The crusher delivers the coal to a belt conveyer marked *D* in Fig. 2 made of five-ply rubber belt 20 inches in width, running at 210 feet per minute. This passes beneath the flag pavement of the alley separating the storage shed from the boiler house, and enters the latter

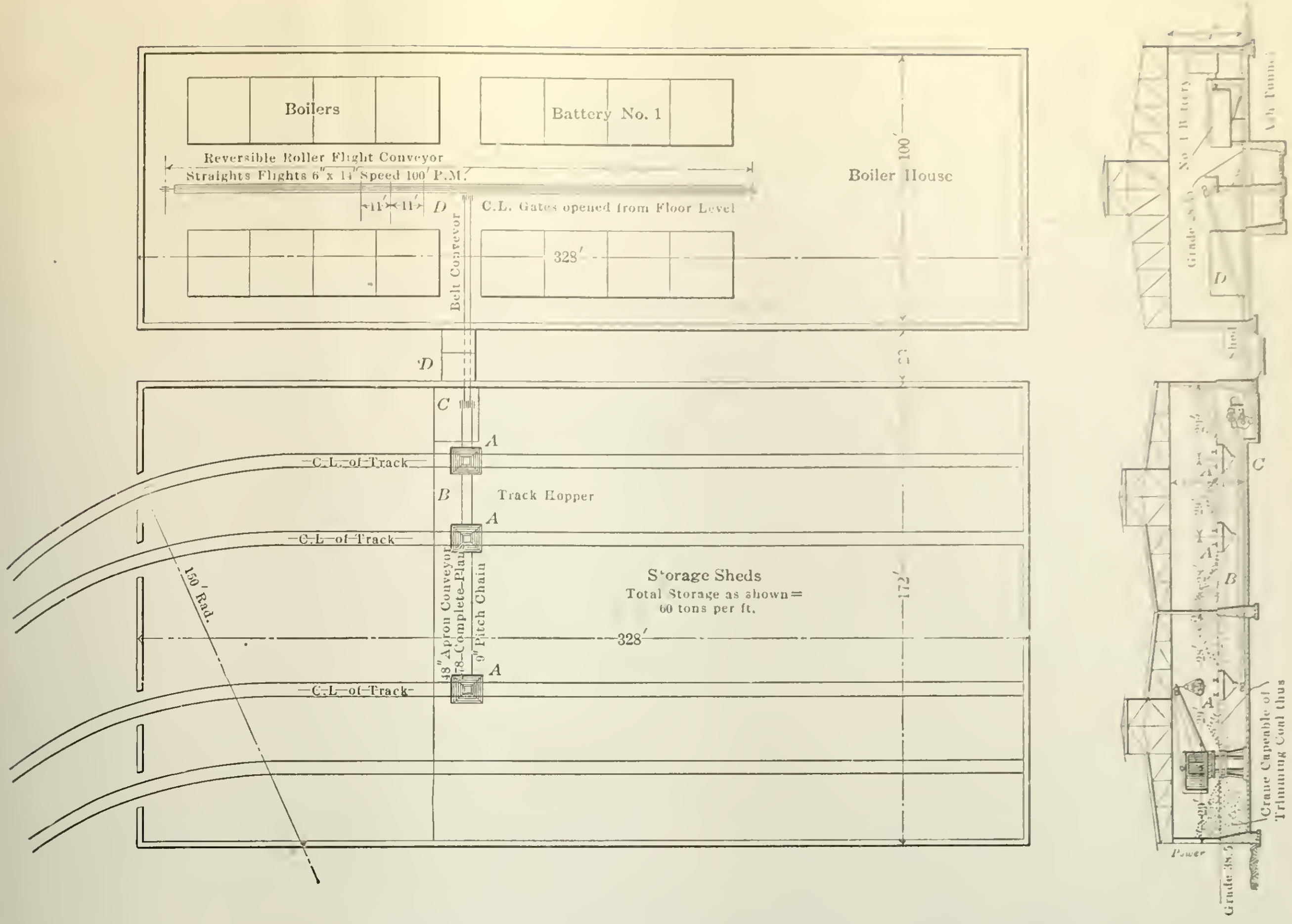


Fig. 2. Plan and Elevation of Storage Sheds

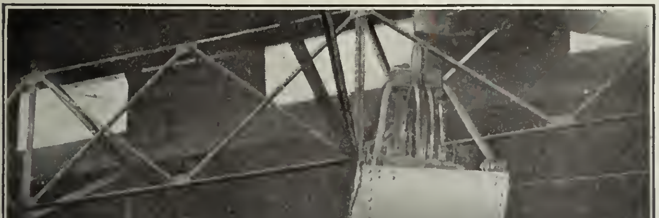


Fig. 3. Coal Hopper beneath Track

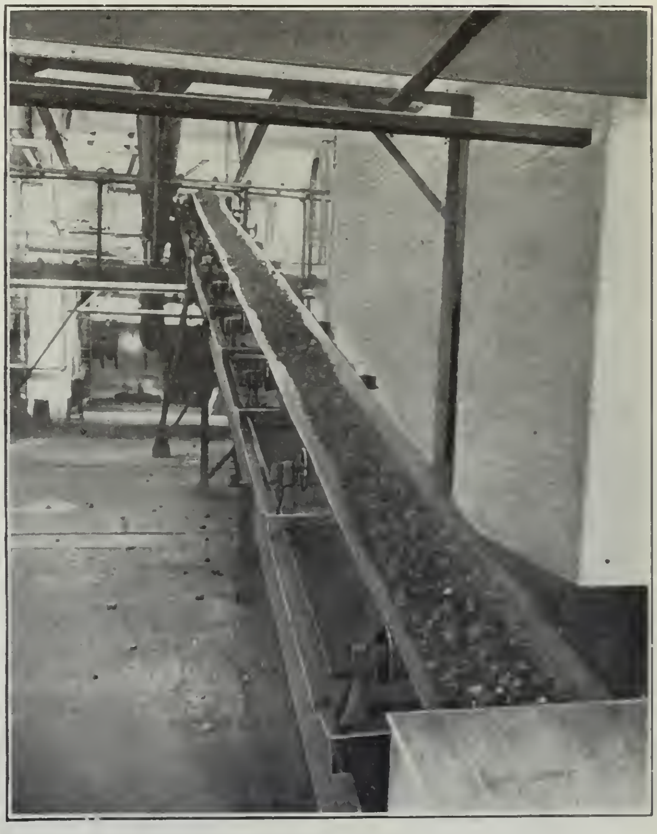


Fig. 4. Belt Conveyer Entering Boiler House

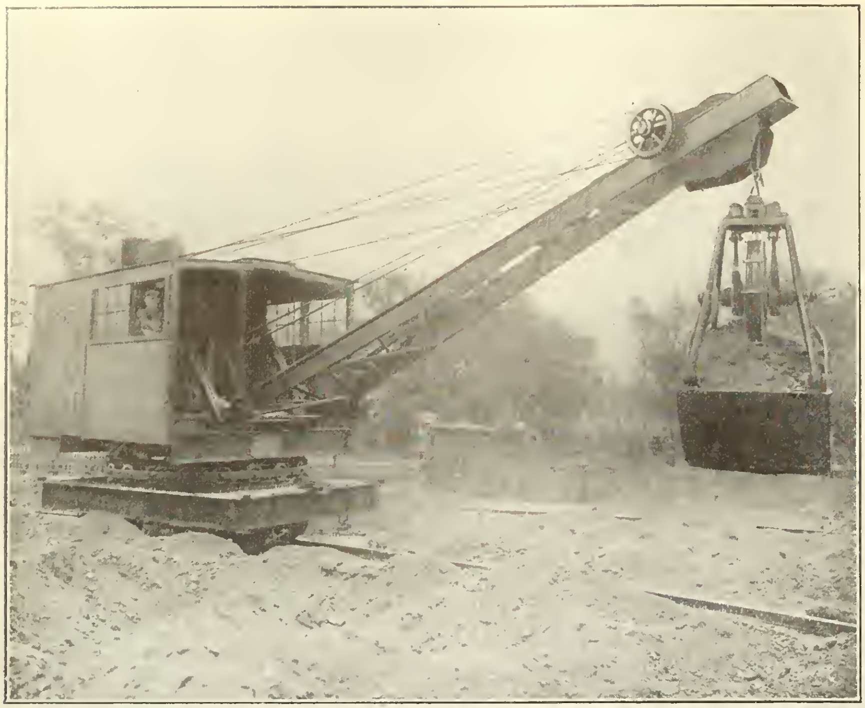


Fig. 5. Locomotive Crane

upon an incline, as shown in the elevation in Fig. 2 and in Fig. 4, delivering to the reversible roller flight conveyer extending through the center of the firing room, as shown upon the plan and in Fig. 6. This conveyer can be run to carry the coal in either direction and through gates can be set to discharge it at any point, forming a continuous bank within easy reach of the firemen of the boilers upon both sides.

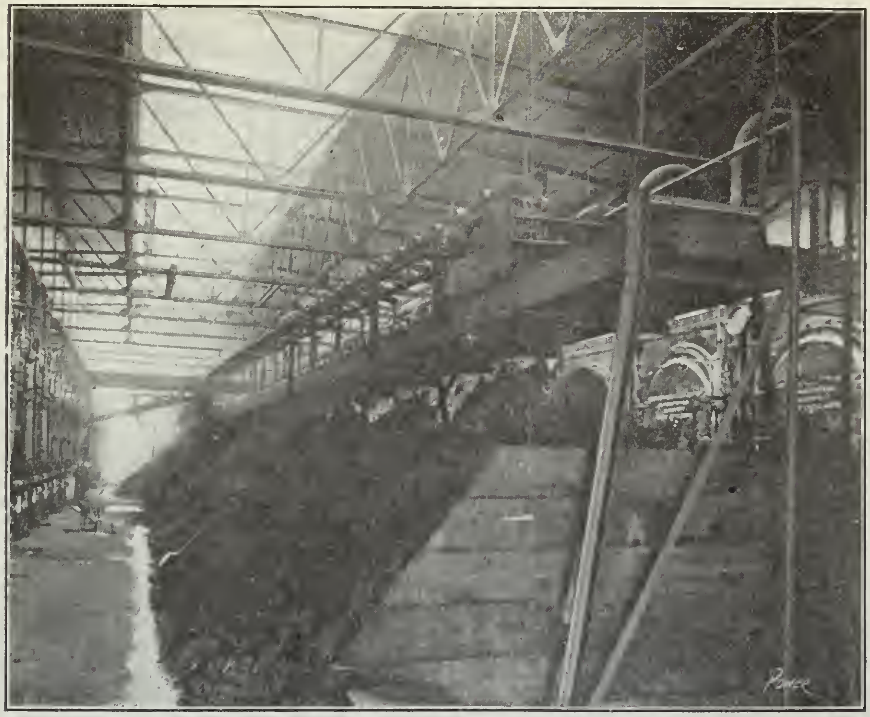


Fig. 6. Reversible Roller Flight Conveyer in Boiler Room

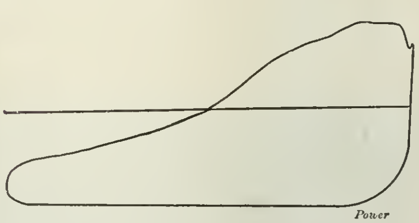
The requirements of the plant, if all of the boilers were in use at their maximum capacity, would be some 20 tons per hour. The conveying apparatus can easily handle 40. For its operation there are required three men, one on the locomotive crane, one at the apron and one on the distributing gallery in the fire room. The handling by the cart method required nine men and three carts and horses. The crane would be almost essential even if the conveyance by carts were continued for bringing in coal from the yard storage and for trimming and piling above the level at which the cars are dumped. The crane was designed and built by the J. M. Dodge Company, and the conveying machinery by the Link-Belt Company.

[Note: The credit of installing the coal-conveying system at the Wood Worsted Mills belongs to Charles A. Hardy, assistant treasurer of the mills. When I designed this plant first, having twenty boilers running, I thought horses and carts would be the cheapest thing, but after the whole plant was running I found that I was mistaken. The coal conveyer has proved a big saving to us.

George H. Diman,
Consulting Engineer.]

Faulty Cylinder Construction

The accompanying diagram shows a condition that ordinarily would cause an engineer to think overtime in an endeavor to locate the trouble. In this instance, the drop in the admission and steam line occurs on the head end of the low-pressure cylinder and on the crank end of the high-pressure cylinder of a cross-compound Corliss engine. It kept the engineer busy for a time finding out what caused the defect in the diagram, and changes in the eccentric both back and forward made practically no difference in the formation of the drop.



Not the Result of a Leaky Valve

At last the indicator cock on the head end of the low-pressure cylinder was removed while the writer happened to be present and a ½-inch rod, the same size as the hole in the cylinder, was inserted with the piston on the head-end dead center. An obstruction was found, which proved to be the piston of the engine. This solved the question in all probability, as it will doubtless be found that there is no channel cut in the head so that the indicator cock on that end of the cylinder can get full steam pressure at the beginning of the stroke. The engine is built by a well known manufacturer, and as the same defect occurs on one end of both cylinders, it does not speak well for carefulness in the little details of construction.

Many men working in an engine room class themselves as engineers, and their work as engineering. Wrong! Starting and stopping an engine does not make an engineer; neither is such work of a professional order. Guess again.

Centrifugal Pump Characteristics

The performance of a centrifugal pump is completely known when simultaneous values have been obtained for the following quantities: speed, usually in revolutions per minute; head, usually in feet of water or pounds per square inch; delivery, usually in gallons per minute or cubic feet per minute; power consumption, usually in horsepower.

Since centrifugal pumps are ordinarily designed to operate at a fixed speed, the inter-relations of the other quantities involved under conditions of constant speed only will be discussed. These relations are most conveniently shown by a chart, as in Fig. 1. The values used in preparing this chart are obtained as follows:

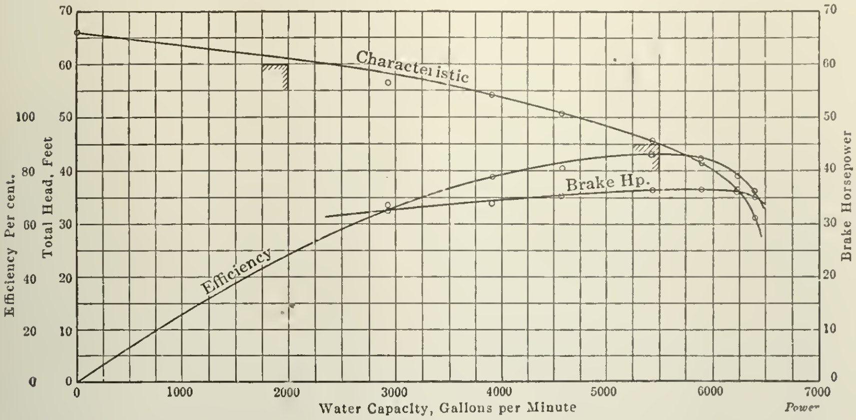


FIG. 1. CHARACTERISTIC OF PUMP SUITABLE FOR BAROMETRIC CONDENSER

The pump is first connected up as shown in the diagram in Fig. 2. The total suction head, including both the vertical lift and the friction of the piping, is measured by a suction gage connected to the suction pipe as nearly level as possible with the middle of the suction opening on the pump, while a pressure gage is likewise connected to the discharge pipe. The latter gage will indicate the total discharge pressure pumped against, including friction head as well as vertical lift.

The discharge of the pump may be measured in several ways. The standard calibrating method that has been found both most convenient and most accurate by the De Laval Steam Turbine Company, to whom we are indebted for the accompanying sketch and drawing relating to De Laval pumps, involves the use of standardized nozzles. These nozzles are made of a peculiar shape to secure the highest possible coefficient of discharge. By taking the velocity of the discharge with a Pitot tube, as shown in the diagram, the volume of the discharge may be calculated with great exactness. The vertical column connected to the Pitot tube may contain either water or mercury, according to the head to be measured.

> *A set of test figures, usually represented by a group of curves known as characteristics, shows what may be expected of a certain pump under various conditions of head and output. Characteristics are useful in selecting a pump for a given class of work.*

Inasmuch as mistakes are frequently made in calculating centrifugal-pump efficiencies, it may be well to state that the total head generated consists of the suction head measured at the entrance of the pump, that is, including the friction as well as the vertical lift, plus the pressure head measured at the discharge, plus the difference in level, if any, measured between the two gages, plus the gain in velocity head between the two points where the gages are connected, if the suction and discharge pipes do not happen to be of the same size.

The speed being kept constant throughout the test, the delivery is controlled by means of a throttle valve in the discharge pipe. When the pump is discharging water at a certain constant rate, the corresponding total head generated is read from the gages. From voltmeter and ammeter readings, taken simultaneously, and from the known efficiency of the driving motor, the horsepower consumed at the same instant in driving the pump is calculated. The rate at which mechanical work is done on the water is derived by multiplying the head generated by the rate of delivery. Dividing this product, as expressed in appropriate units, by the power delivered to the pump shaft, the mechanical efficiency is obtained.

Having these values for head, delivery, power and efficiency, a distance corresponding to the rate of delivery is measured horizontally on the scale at the bottom of the chart. Vertically above this, and according to the scale or scales at the left of the chart, points are located for the head, power and efficiency values obtained for the same instant. The opening of the valve in the discharge is then changed, changing the flow, whereupon readings are again made of all quantities as before, and the corresponding points marked on the chart. Repeating this operation a sufficient number of times, a series of points will be obtained for head, power and efficiency, from no delivery to full delivery. A smooth line drawn through the points for head is called the head-capacity curve or "the characteristic" of the pump at the given speed. A similar line drawn through the points for efficiency is known as the efficiency characteristic, while a third line drawn through the points for horsepower is called the horsepower characteristic. All three curves, taken together, are

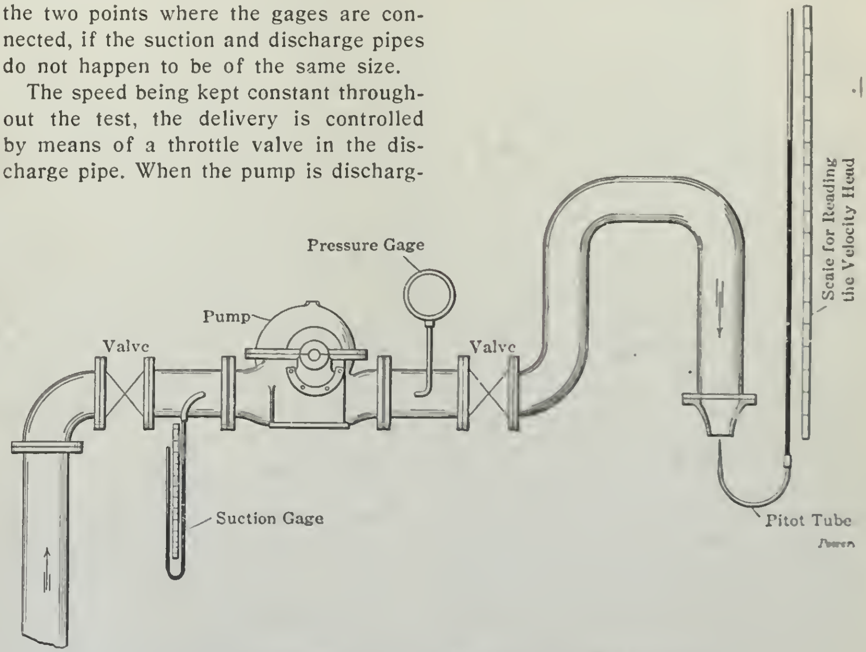


FIG. 2. ARRANGEMENT OF PUMP AND TESTING APPARATUS

known as the "test characteristics" or performance curves of a particular pump at a given speed.

From an inspection of the characteristics of a given pump, its suitability for any particular service may be determined. Suppose that we are selecting a pump to supply water from a lower level to a barometric condenser. The pump should preferably be able to start the flow of water when there is no vacuum on the condenser and will therefore be required to generate a head equivalent to the full difference in the level between the suction sump and the top of the condenser. On the other hand, when the vacuum has been established the head to be generated will be diminished by the amount of the vacuum. That is, of a total difference in level of 60 feet, the pump will need to contribute only about 35 feet of the total head when the vacuum is on and the pump should be able to handle water under both conditions with good efficiency when running at a constant speed. The characteristics of a pump especially designed for this kind of service are shown in Fig. 1.

As will be observed, the characteristic, that is, the curve showing the relation between the total head developed and the delivery in gallons per minute, has a considerable slope, starting in with about 66 feet at normal delivery and falling to about 42 feet at a delivery of 5850 gallons per minute. The design of the pump is such that the highest efficiency, namely, 83 per cent., is obtained under the normal conditions; that is, when the pump is delivering the larger quantity of water against the lower head, although the efficiency under starting conditions, when the pump must generate a 60-foot head to lift the water to the condenser without the help of the vacuum, is very fair, being about 50 per cent.

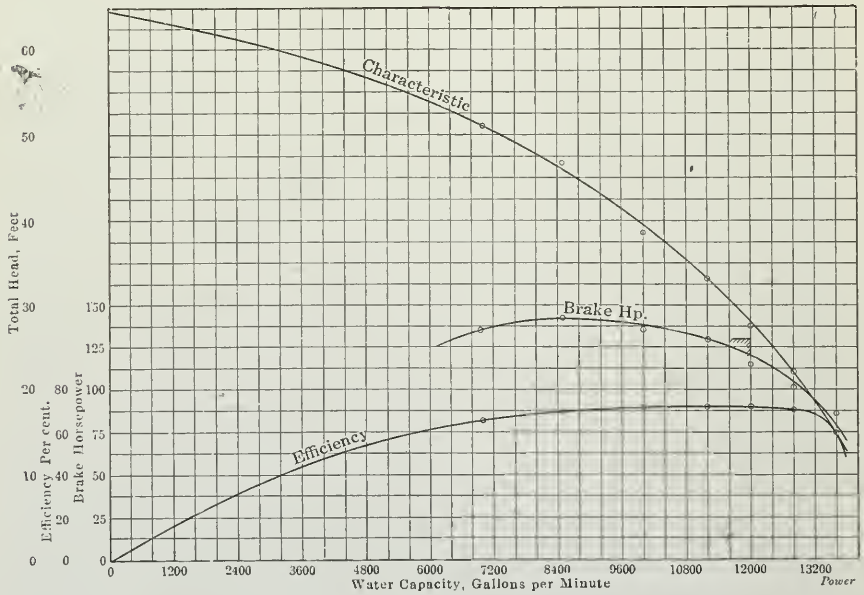


FIG. 3. CHARACTERISTICS OF PUMP FOR EMPTYING DRYDOCK

The characteristics of a pump having a still steeper head-delivery curve are shown in Fig. 3. As will be noted the head falls from about 64½ feet at no delivery to about 27 feet at a delivery of 12,000 gallons per minute, which is the rate of delivery for which the pump was designed and at which point its efficiency is 76 per cent. It will be noticed that the efficiency of this pump is remarkably uniform, being practically 70 per cent. all the way from a delivery of 7200 gallons per minute up to 13,200 gallons per minute.

Inasmuch as the efficiency is very nearly constant with increased delivery and a rapidly falling head, it will be of interest to note what is happening to the motor simultaneously. The brake horsepower consumed by some pumps increases rapidly with increased delivery; in fact, a pump may cause the motor to be burned out if pressure be removed from the discharge. In the case of this pump, however, it will be noted that the brake horsepower reaches a maximum of about 130 at a delivery of 8400 gallons per minute and thereafter decreases, falling to about 114 horsepower at a delivery corresponding to nominal rating, namely, 12,000 gallons per minute. A pump of the type just discussed would be ideal for such service as pumping out drydocks. At the start, when there would be practically no head to overcome, it would handle enormous volumes of water at high efficiency and it would continue pumping with good efficiency against a steadily increasing head until the dock were entirely emptied.

At no point would there be any danger of overloading the motor, since the horsepower consumed reaches a maximum at a delivery of about 8500 gallons per minute. After this the power taken by the pump diminishes, as above pointed out. This automatic protection to the motor is secured, not by the addition of governing apparatus or other contrivances to modify the speed or to change the form of the pump passages, but by the permanent shape of the impeller and other parts of the pump, and a pump having these qualities will, therefore, retain them under all conditions and at all times. This property of the pump makes it possible to use a smaller and less expensive motor and, further, it permits the selection of a motor having the maximum efficiency at the output coinciding with the demands of the pump under normal operating conditions. Electric motors, as now built, have only a limited overload capacity, and although it might be possible to equip a pump with a motor of sufficient size to provide for any overload, such practice is not at all economical.

In considering the efficiency of a centrifugal pump, the shape of the efficiency curve should be taken into account, as well as its maximum efficiency; that is, it hardly ever happens that a pump is at all times operated at the delivery for which it was designed or at the delivery corresponding to the maximum efficiency, and the average efficiency at the different deliveries which the pump will be called upon to meet in actual service is more important. If the efficiency curve is steep, the efficiency will fall off greatly as delivery is reduced, while, on the other hand, if the efficiency curve is flat, the efficiency will be fairly high through a wide range of load. It sometimes happens that a large pump is bought with a view to a possible increase later on in the amount of water required. This, however, will result in operation at low efficiency at the beginning and in most cases it will probably be advisable to buy a smaller pump to begin with and add another pump or replace with a larger pump when necessary.

The small shaded corners under the head-delivery curves indicate the conditions specified by the purchaser before the pump was built and the proximity of this point to the curve shows how closely the desired results were achieved by the designer. In other words, when the pump was purchased, a certain speed, head and delivery were specified and it became the designer's task to obtain the required head and delivery at the specified speed and at the same time to so design the pump that it would give the highest efficiency at this point. Accuracy in predetermining the characteristics of a pump is important from the point of view of the motor, since if the pump is to be driven by an alternating-current motor, it may prove very difficult to modify the speed after installation or at least such modification will involve diminished efficiency. On the other hand, it is poor practice to purchase a pump which gives too high head and to reduce the head by throttling, since this involves a direct loss of energy.

Application of Thermal Storage

> *By means of a tank located just above the boiler and connected by pipes to both the steam and water spaces of the latter, feed water is heated to the steam temperature during periods of light load, to be available during periods of peak load. The scheme is analogous to that of a flywheel on an engine or to a storage battery in connection with a dynamo.*

During the past few years the employment of thermal storage in connection with a number of large boiler plants in England has resulted so successfully as to insure its permanent and extended use. The following from *The Engineer* refers briefly to the general principles of thermal storage and some of the theories advanced to account for its results.

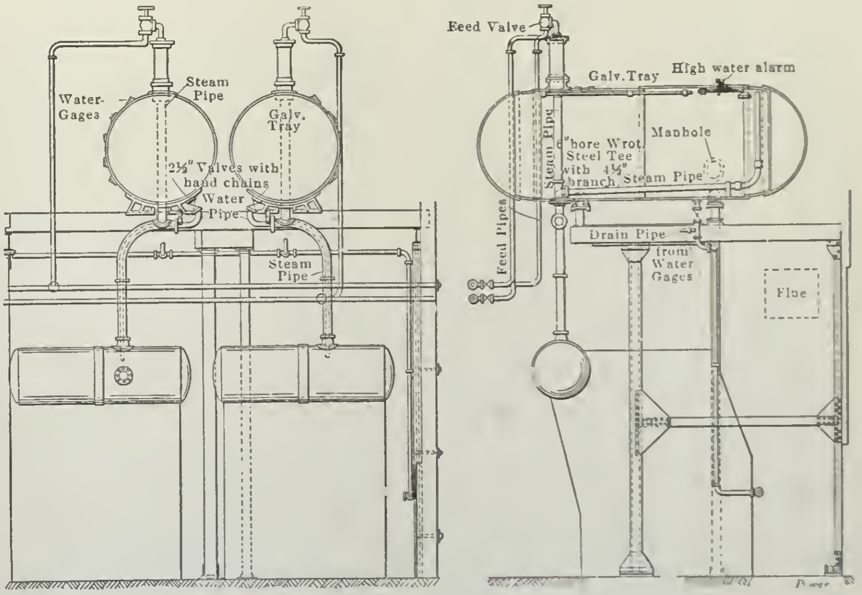
A spherical-ended cylinder is erected above the boiler in any convenient position, and is connected to it by means of a steam pipe leading from the steam space of the boiler through the lower surface of the storage cylinder, and extending into the interior of the latter to within a few inches of the inner and upper surface. The generation of steam within the boiler thus fills the storage cylinder with steam at boiler pressure. From the lower part of the storage drum a second pipe, provided with a valve, leads into the boiler and terminates below the water level. In this way, feed water introduced into the storage drum is heated by coming in contact with the live steam to the temperature corresponding to the boiler pressure, and passing through the valve on the second pipe it descends by gravity into the boiler, there mixing with the water already present. The valve on the second pipe is employed to regulate the amount of water fed to the boiler, and by operating this suitably it can be arranged so that during periods of light load the flow of water into the storage cylinder is greater than that taken from it. In this way, during such periods, a supply of hot water is gradually accumulated until the vessel is nearly full. With the arrival of the peak load, when under ordinary circumstances the boiler would have to be forced to meet the demand, the hot water already in the vessel is drawn upon for the purpose.

Thermal storage does for the boiler what a flywheel accomplishes for the engine, or storage batteries do for the dynamo. But in the mere provision of a flywheel there is nothing to cause an increase in the aggregate output of the engine. Such, however, is the case with the thermal-storage plant, for it is found that at all times, whether the boiler is fed exclusively from the storage drum, as during the time of peak loads, or under ordinary circumstances, when the water first passes through the drum on its way to the boiler, the evaporation is increased by anything from 50 to 150 per cent., as compared with what is obtained even with economizers in use.

It has been suggested that these feed reservoirs permit the boiler to be forced with satisfactory results to an extent which under ordinary circumstances would inevitably entail a prohibitive amount of priming. Again, the introduction of the feed water at the same temperature as the steam in the boiler is known to produce smoother working, and in the absence of the irregular generation of steam may be found the principal factor in the causes leading to the enhanced output.

A test recently conducted by Prof. J. D. Cormack on the maximum output from a water-tube boiler fitted with and without the storage system gave the following results:

	Without Thermal Storage.	With Thermal Storage.
Duration of test, minutes....	100	65
Draft in inches of water at chimney base............	1.04	1.08
Gage pressure, pounds per square inch..............	197	201
Evaporation per hour, pounds of steam................	14,830	26,080
	Per Cent.	Per Cent.
Output in terms of the rated capacity of the boiler (12,000 pounds steam per hour)...	123.6	217.3



Thermal Storage Applied to Water-tube Boilers

The accompanying sketch shows thermal-storage tanks applied to water-tube boilers at the plant of the St. Pancras Borough Council. It will be seen that the drums are erected on a steel gantry above the boilers, each drum being directly over the boiler which it is intended to serve. From the front steam drum of each boiler a pipe leads upward to the underside of the storage vessel, inside of which it branches into two parts, one limb rising straight upward and one passing at a slight inclination to the back of the vessel's interior and then upward. Directly above the first of these branches is placed a tower and to this point the feed water is led. Passing through a stop valve, the water descends through a spraying nozzle, and becomes mixed with the steam. The pipe leading from the water space of the storage vessel to the water space of the boiler drum is led off from the under surface of the former, and in order to avoid making an additional joint on the boiler, this pipe is made to pass within the uprising steam pipe. The storage vessels are also provided with a blowoff pipe, water gages and a high-water alarm.

An important feature of these storage vessels is that where the feed water is of a hard or impure character, the lime and other impurities are found to deposit themselves within the storage vessel rather than in the boiler proper. This being so, the accumulated deposit is readily removed by blowing off, as the absence of any direct contact between the storage vessel and the heat of the furnace prevents this deposit from being baked to a hard scale. In fact, the lime, etc., contained in the feed water passes through the blowoff valve of the storage vessel in a soft, powdery state.

Mechanical Refrigeration

By F. E. Matthews

A short review of the history of the development of the art from the time of primitive man up to present day practice.

It has already been stated that in the study of mechanical refrigeration the real entity to which we must direct our attention is heat. The two principal sources of heat are chemical reaction and solar radiation. The most important of all chemical reactions to the engineer, and in fact, to the world in general, is that of combustion; but even this source of heat has its origin in solar radiation.

Primitive man was forced to depend for his comfort on solar heat irregularly dispensed, supplemented by the combustion of the most easily available fuels. If he chanced to be fortunate in hunting he feasted; if not he fasted perforce. When the sun favored him with sufficient heat to keep him warm, his fresh meat quickly became unfit for use, for he knew not the luxury of modern cold-storage facilities. When the sun's heat became less, so that he could keep his provisions untainted, he was kept busy gathering kindling wood.

Long before the appearance of man, however, provision had been made by nature to alleviate the discomfort brought about by the irregularity of solar heat. At an early period it seems that the earth's atmosphere was very rich in carbon-dioxide, and contained much moisture; a condition favorable to a rank growth of plant life. The carbon-dioxide which is absorbed by the vegetation, is broken up and the carbon becomes a fixed part of the plant structure while the greater part of the oxygen returns to the air. The energy necessary for this chemical process is derived directly from the sun. During this period of enormous growth, vast quantities of vegetation, becoming buried beneath the surface, fell into partial decay, and finally became coal. Thus it was that there was stored up beneath the surface of the earth in these great carbon deposits, an enormous amount of latent solar energy. There it remained until man's ingenuity and industry, prompted by ever-increasing necessity, set him to burrowing in the ground to bring up the blackened remains of primeval forests, and so making the latent heat pay the price of liberty by carrying his burdens and turning the wheels of his factories.

While this source of heat was found accessible to meet the demands of variable weather, it still remained for man's ingenuity to devise some method of doing away with the alternate feasting and fasting made necessary by the heat and the exigencies of the hunt. In this he had the advantage of nature's most practical object lessons.

At first he did not stop to consider that in the freezing of natural bodies of water, large quantities of heat must pass to the air and surrounding objects, thereby virtually heating them, or that in order for the ice to melt similar quantities of heat must return to the water from the air and surrounding objects, thereby refrigerating them. He simply thought he observed that frozen water, produced by some mysterious power that the cold north wind brought from the realm of ice and snow, was nature's favorite cooling material, and as his needs for cooling effects developed, his first thought was to store away a quantity of this product each winter in the caves which nature likewise thoughtfully provided.

With certain slight refinements in the operation, this same crude method employed by our primeval ancestors is still being used to no unconsiderable extent. The fact, however, that summer requirements must be anticipated at least one season by both nature and man, has combined with a score of other elements to force the introduction of more scientific methods.

Laboratory methods of producing low temperature by means of so called "frigorific mixtures," by which a perceptible drop in temperature is produced by certain endothemic chemical reactions and solutions have been known for at least three centuries.

Probably the most common example known of a frigorific mixture is that of ice or snow and salt. As has already been stated, the addition of a foreign substance to a liquid lowers its freezing point. The effect of the addition of different amounts of common salt (Na Cl) and that of calcium chloride (CaCl) is clearly set forth in tne accompanying table. Since the addition of 10 per cent. of salt, by weight, to water lowers its freezing point to 18.7 degrees Fahrenheit and prevents its changing to the solid state till that temperature is reached, it would follow that the addition of the same percentage of salt to snow or finely divided ice would cause it to return to the liquid state at all temperatures above 18.7 degrees Fahrenheit. The result is that the ice at once begins to melt, but to do so it must absorb 144 B.t.u. of heat per pound, and in the event that this heat is not forthcoming, the temperature of the mixture will continue to fall until the freezing point corresponding to the per cent. solution is reached. At this point it will continue to exist in the solid state, and aside from the lesser intimacy of the mixture of the two constituents, ice and salt, will be the same substance as frozen brine of the same per cent. composition, and will exist under the same conditions of temperature.

EFFECT OF DIFFERENT AMOUNTS OF SALT ON FREEZING POINT OF LIQUID.

Calcium Chloride (CaCl) Brine.				Sodium Chloride (NaCl) Brine.			
Degrees Salinometer at 60° Fahr.	Specific Gravity at 60° Fahr.	Specific Heat.	Weight Pounds Per Cubic Foot.	Degrees Salinometer at 60° Fahr.	Specific Gravity at 60° Fahr.	Specific Heat.	Weight Pounds per Cubic Foot.
12	1.024	0.980	63.923	4	1.007	0.992	62.862
27	1.041	0.904	65.082	8	1.015	0.984	63.361
36	1.058	0.936	66.066	10	1.019	0.980	63.611
40	1.076	0.911	67.169	12	1.023	0.976	63.860
44	1.085	0.896	67.931	14	1.026	0.972	64.048
52	1.103	0.884	68.755	16	1.030	0.968	64.297
62	1.121	0.868	69.979	20	1,037	0.960	64.734
80	1.159	0.844	72.456	24	1.045	0.946	65.234
88	1 179	0.834	73.599	28	1.053	0.932	65.733
95	1.199	0.817	74.847	32	1.061	0.919	66.173
104	1.219	0.799	76.096	36	1.068	0.905	66.669
112	1.240	0.778	77.407	40	1.076	0.892	67.159
120	1.305	0.767	81.464	48	1.091	0.874	68.105
				60	1.115	0.855	69.603
				80	1.155	0.829	72.101
				100	1.196	0.783	74 660

The specific gravity of a substance is the raito of the weight of that substance to the weight of the same volume of pure water at its temperature of maximum density at 39 degrees Fahrenheit, at which temperature it weighs 62.425 pounds per cubic foot. The weight per cubic foot of brine given in the table is determined by multiplying 62.425 by the specific gravity as determined by a salinometer or other similar hydrometric instrument.

The greatest of all heat-absorbing changes in state of matter is that of vaporization. Since increase in pressure has the effect of raising the boiling point, or the temperature at which the liquid vaporizes, and decreasing the pressure has the effect of lowering it, it is only natural that reduction in pressure below that of the atmosphere or vacuum was first employed in attempts to cause some of the better known liquids to boil at sufficiently low temperatures to produce artificial "cold."

While there is evidence that some little experimenting was done with liquids under vacuum as early as 1755 there is no authentic information that anything of real importance was accomplished until the early part of the last century, when several independent inventors built experimental machines, none of which, however, seem to have produced any very encouraging results until Jacob Perkins, an Englishman, developed an ether-compression machine which he patented in 1834.

Perkins operated his ether-compression machine under vacuum, much as our present day ammonia-compression machines are operated when low temperatures are required. His patent of August, 1834, described his machine as being composed of the four principal parts which constitute our present machines, viz., a containing chamber or evaporator in which the refrigerating medium evaporates and through the walls of which it passes heat from the substance it is desired to refrigerate; a pump or compressor for drawing the evaporated refrigerant from its containing chamber and exerting upon it sufficient pressure to cause it to liquify when cooled; the cooler or condenser consisting of a coil of pipes submerged in water, and in which the refrigerant is cooled and liquified after compression; and a regulating or expansion valve for controlling the flow of the refrigerant liquified in the condenser as it passes under the higher condensing pressure to the evaporator maintained under a lower pressure due to the action of the compressor.

The evaporator shown in the Perkins patent consists of a circular tank made of two dished metallic disks, which receptacle was submerged in the fluid to be refrigerated. The compressor was designed to be operated by hand but otherwise the general arrangement of its working parts, except for the fact that the cylinder was invented, was very much the same as that of our present small single-acting ammonia compressors with both suction and discharge valves in the head. A simple submerged condenser, consisting of a single zigzag coil, and a hand-operated expansion valve were employed.

Both the compression and the absorption machines find their origin in the demonstrated possibility of liquifying so called gases. In 1823 Faraday announced to the world that he had succeeded in liquifying chlorine, ammonia and carbon-dioxide, as well as several other gases of less importance to the refrigerating industry.

By the same method by which Faraday first accidentally liquified chlorine, carbon-dioxide, ammonia, sulphur-dioxide, methyl-ether, Pictet fluid, sulphuric ether, ethel chloride, water, and other substances may be liquified experimentally under the proper conditions of temperature and pressure.

The first ammonia-absorption machine recognized as such, was invented by Carré about the year 1855, in which same year Harrison, an Australian, and Professor Twining, an American, are said to have independently perfected the Perkins ether machine, the latter inventor having performed the then marvelous feat of artificially freezing blocks of ice with fish inside.

Not the least among the influences that have combined to forward the development of mechanical refrigeration has been the pollution by sewage of the water from which the national ice crops are harvested. This has been especially true since it has been demonstrated that negative bacteriological tests, no matter how carefully conducted, are not insurance against typhoid, and there is accordingly well founded and rapidly growing prejudice against all ice known to be cut from sewage-bearing streams and lakes.

The breweries were among the first to adopt improved methods of cooling largely because of contamination of products through the unsanitary conditions inevitably resulting from the use of ice. This, together with the heavy labor charge which it entails, forced the development of a system of mechanical refrigeration. While the item of contamination was somewhat less important in the abattoirs than in the breweries, the enormous amounts of ice consumed and the uncertainty of the natural crop, made the adoption of mechanical refrigeration in this industry imperative from an economic standpoint.

In cold-storage work the inability of ice, without the addition of salt, to produce sufficiently low temperatures to satisfactorily preserve many perishable products, has of late years, at least, probably been the most patent factor in the combination to force the substitution of mechanical systems for ice in the cold-storage industry.

The demand for artificial means for producing refrigeration having first appeared among the larger industries above cited, the builders of refrigerating machines first set about to supply systems best adapted to their peculiar requirements. Later as the requirements of the larger consumers of cold began to show signs of having limitations, the builders began to look for other fields and quite naturally began to develop systems better adapted to the requirements of smaller consumers of ice. While the idea is still current with a certain class of consumers that food products kept in mechanically cooled compartments suffer thereby, the intelligent merchant is willing to acknowledge the superiority of mechanical cooling means in almost every case.

The chief factor which fixes the capacity limit under which it becomes commercially impracticable to install small mechanical refrigerating plants has been the cost of attendance which is practically as great for all smaller sizes as for those of from five to ten tons capacity, and this, in the general case in which steam power is employed 24 hours per day, solely for the operation of the refrigeration plant, becomes practically prohibitive. The first step toward surmounting this obstacle was to operate the plant in the daytime only in order to eliminate the expense of night attendance. In order to do this the brine-circulating system commonly called the "brine system," which will be described in detail later, was introduced. The brine system, however, usually consumes power 24 hours per day, and in order to reduce this expense the "congealing-tank" system, which will also be described in detail in a succeeding issue, was employed to some considerable extent.

Where steam is not required for other purposes and the amount of power necessary for the refrigerating plant is small, the substitution of combustion-engine power is often capable of reducing the cost of fuel as well as that of attendance; the former where cheap kerosene, crude or fuel oils of light density can be procured, and the latter where licensed attendants, insurance, and other factors work disadvantageously for the small steam plant.

The comparatively low first cost, as well as that of combustion-engine power, coupled with the slight expense for attendance now made possible by the application of certain automatic regulating and safety devices, makes this type of plant when properly installed either with or without "congealing tanks" as the case may require a most practical and satisfactory plant for the small consumers of ice. To the end of producing a small mechanical refrigerating plant capable of operating with still less attendance, so called completely automatic systems which will also be described in detail, have been developed. While it is a fallacy to suppose that these machines will operate without some attendance, they are often capable of operation with far less attendance than any other type, and were it not for the usually comparatively high cost of electric power necessary for the operation of a completely automatic plant, the advantages gained would undoubtedly more than compensate for the high first cost of such plants.

No one type of plant can be expected to be the most advantageous under all requirements, and the relative advantages and disadvantages of one over the other in first cost, operating cost, superiority of design, and safety should be carefully considered and balanced up before a decision is made.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Storage Batteries in Isolated Plants

By Norman G. Meade

It is generally recognized now that where uninterrupted 24-hour service from an isolated plant of moderate size is desired, a storage battery is practically a necessity. Usually, the engine and generator capacity in an isolated plant is sufficient for the total number of lamps connected, although they are seldom all in use at one time; consequently, the plant operates at but partial load during the total lighting hours. This means low efficiency, poor regulation and high fuel costs. The installation of a storage battery corrects this weakness by permitting the operation of the generator at the full or the most economical load for a few hours and then shutting it down entirely, the battery providing the current for the balance of the time. In many cases it can be so arranged that the generator need be operated only every second or third day, and then during the most convenient part of the day.

When on special occasions extra lighting is required, the battery may be discharged in parallel with the generator and current equal to their combined capacity may be supplied.

The principal function of a storage battery in small plants being the furnishing of current for a considerable period of time, the operation of the battery consists of cycles of charging and discharging covering practically the capacity of the battery.

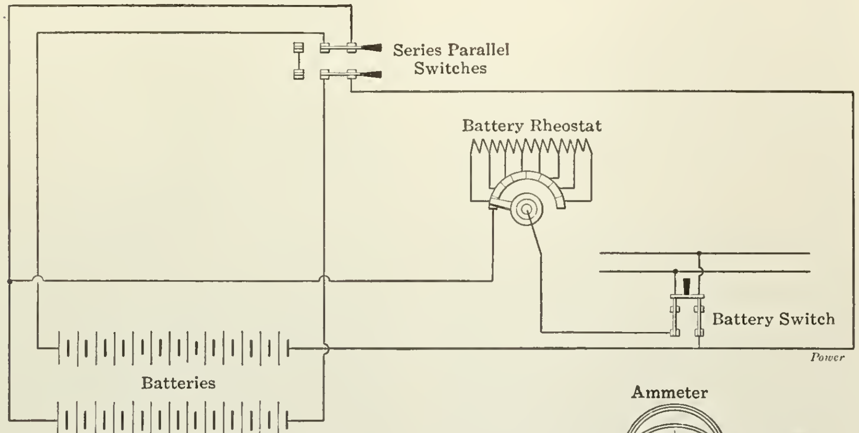


Fig. 1. Elementary Connections for Charging in Parallel and Discharging in Series through a Rheostat

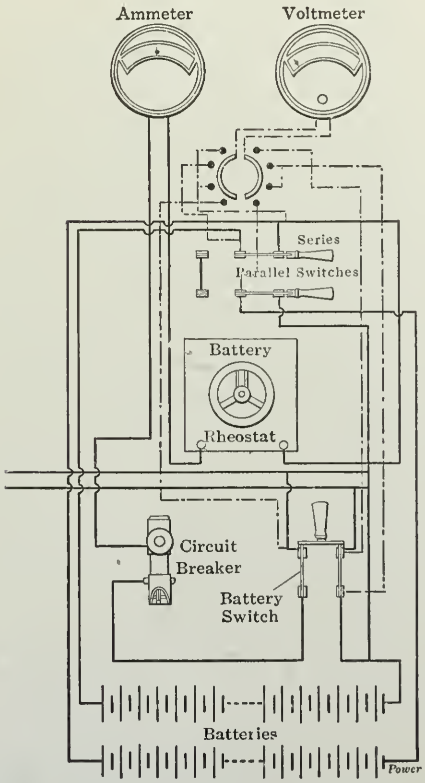


Fig. 2. Complete Switchboard Corresponding to Fig. 1

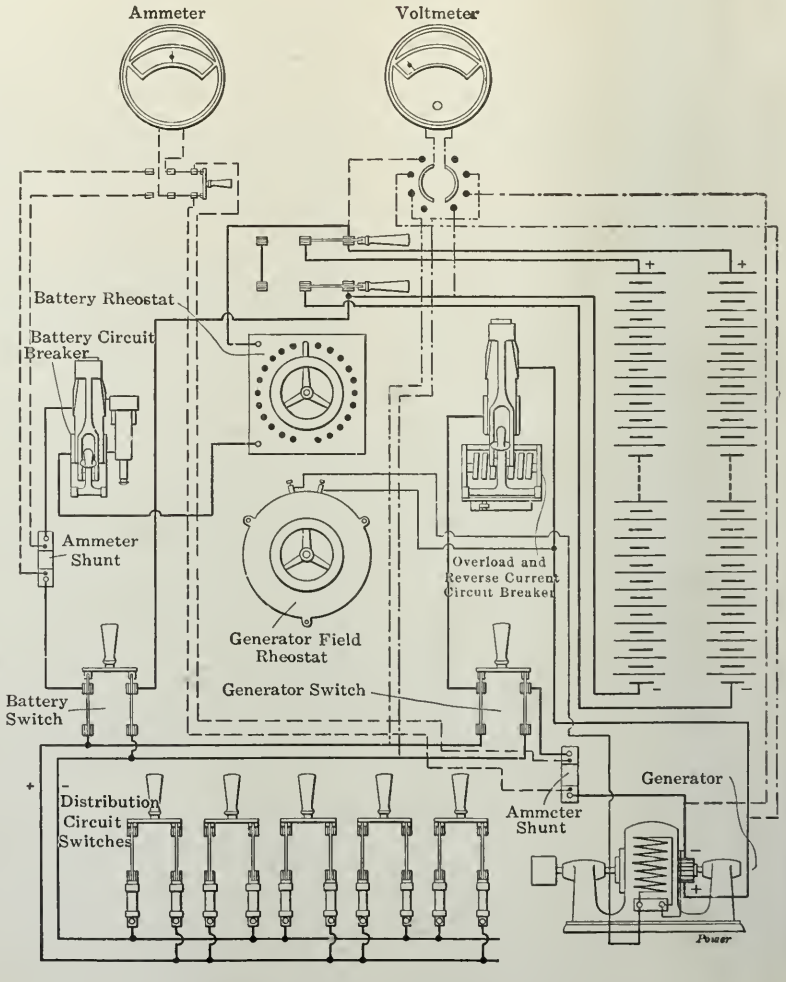


Fig. 3. Switchboard Connections for Generator, Battery and Distribution Circuits

Under ordinary operating conditions, it is desirable to maintain practically constant voltage on the lighting circuits, and as the voltage of a storage battery varies during discharge, methods have been devised to compensate for the changes in the battery voltage.

A fully charged battery which shows about 2.1 volts per cell standing idle will show, while discharging at an eight-hour rate, about 1.8 volts at the latter part of the discharge and somewhat less at higher discharge rates. To charge a battery fully it is necessary to raise the voltage, as the charge proceeds, from about 2.2 volts at the start to approximately 2.65 volts per cell at the completion of the charge. This is usually accomplished by a normal voltage charge with resistance control; a high voltage charge directly from the generator or by shunt-booster change, each of which methods will be explained later.

SIZE OF BATTERY REQUIRED

In the selection of the proper battery, the number of cells is determined by the voltage of the system and is entirely independent of the size of the individual cells. Isolated plants of the various standard voltages require batteries of the number of cells in the following table:

Voltage of System.	Number of Cells.	Voltage of System.	Number of Cells.
110	60	220	120
115	64	230	126
125	70	250	138

The data throughout this article are based upon 110 volts, which is the voltage most used.

The size of the individual cell is determined by the amount of current to be supplied and the length of time it must be supplied on one charge. For ordinary purposes it is sufficiently accurate to estimate the energy taken by a 16-candlepower lamp as 55 watts and lamps of other candlepower on a proportionate basis.

Storage batteries are rated in ampere-hours, which is the product of the discharge current and the number of hours the discharge may be continued. The capacity of a battery at the eight-hour rate is considered normal, and if the discharge current is increased above the normal eight-hour rate the ampere-hour rate decreases, as will be seen by the following example:

Time of Discharge, in Hours.	Discharge Current; Amperes.	Battery Capacity in Ampere-Hours.
8	12.5	100
5	17.5	87.5
3	25	75
1	50	50

This means that 12.5 amperes may be obtained for eight hours, giving 100 ampere-hours, but if the discharge current be 25 amperes, it can be continued for only three hours, giving 75 ampere-hours; some additional capacity is, however, available at a lower rate.

On discharge at less than the eight-hour rate, the capacity of the battery is slightly greater, but the increase is small, and for ordinary calculations it is best to consider the capacity at rates lower than the eight-hour rate to be the same as the eight-hour capacity.

The size of a 110-volt battery can be approximately determined by the method indicated in the following example, the conditions being that the battery will be charged at any time during the day convenient to operate the generator and that the battery will be able to furnish current for lamps according to the above schedule.

Time.	Number of Lamps.	Amperes.	Number of Hours.	Ampere-hours.
5 p.m. to 10 p.m.	Twenty 16 c. p.	10	5	50
10 p.m. to 6 a.m.	Two 8 c. p.	½	8	1
6 a.m. to 8 a.m.	Six 16 c. p.	3	2	6
				60

The last discharge rate is three amperes, and there will be required a battery of sufficient size to furnish 60 ampere-hours at a three-ampere rate. This being less than the eight-hour rate, a battery is required having a normal rating of 60 ampere-hours.

The above example shows a condition where the full normal capacity of the battery is used in carrying the load. Under some conditions this is not possible, that is, where the latter part of the discharge is at a high rate

Since the voltage of a cell varies from 1.8 at the end of a discharge up to about 2.65 at the end of a charge, a number of cells in series will produce a widely varying voltage across a line unless some regulating means is provided to compensate for the voltage change. The simplest method is by the use of a resistance in the battery circuit.

RHEOSTAT REGULATION

The simplest system of connections and that most frequently used in small plants is to arrange a battery of sixty cells divided into two sections of thirty each so that the sections may be charged either separately or in parallel through a rheostat. The object of this is to permit the cells to be charged with current from a 110-volt generator. Fig. 1 is an elementary diagram of this arrangement. While the battery is being charged, the series-parallel switches are kept in the position shown, with each section of the battery in parallel. When discharged the switches are thrown over to the opposite position which connects the halves in series.

The objection to this arrangement is the loss of power in the charging rheostat, it being necessary to reduce the voltage by the rheostat from 110 to about 63 at the beginning, and to 79 at the end of a charge. This loss, however, does not necessarily occur except when the current is supplied to the lamps during period of battery charge. If no lamps are required at this time, it is possible to reduce the voltage of the generator to that at which the series resistance can be en-

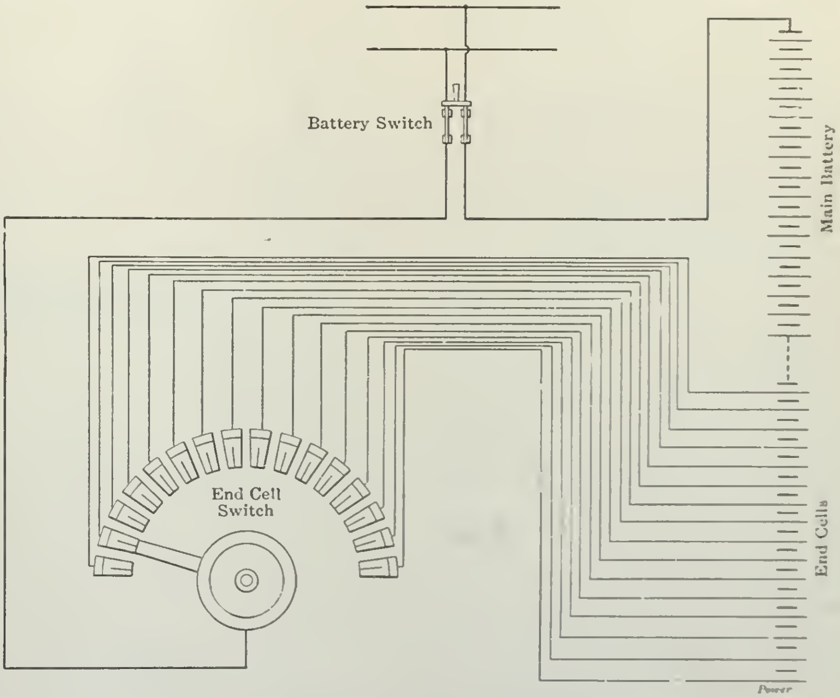


FIG. 4. END-CELL SWITCH CONNECTIONS

tirely cut out of circuit, leaving the battery connected directly to the generator; the charging current is then regulated by means of the generator field rheostat.

The voltage on discharge is regulated by means of the same series resistance that is used in charging. The loss of power in this case is comparatively small, as the maximum drop in the rheostat is never more than 12 volts with a 60-cell battery on a 110-volt system.

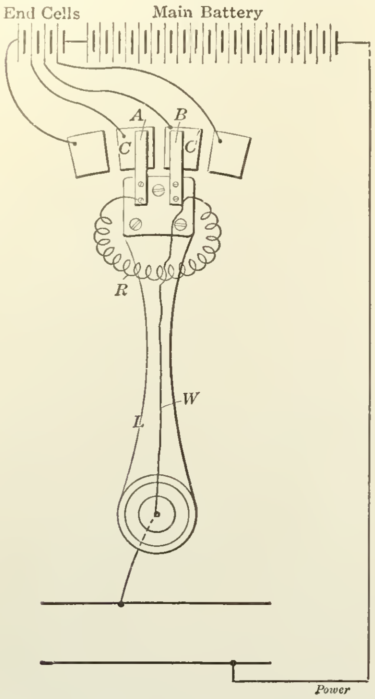


Fig. 5. Device to Prevent Short-circuiting End Cells When Cutting In or Out

Fig. 2 shows the diagram of connections of a switchboard designed for the parallel charge and series discharge with resistance regulation. This switchboard is adapted to plants where the generator is already installed and a battery is added to the equipment. The two single-pole double-throw switches permit the two sections of the battery to be connected in series for discharge and in parallel for charge or either half to be charged separately if desired.

Fig. 3 shows a similar type of switchboard for the control of both the battery and the generator. One ammeter is used to indicate both the battery and the generator currents as shown on the diagram. This is accomplished by having two shunts, one in the generator circuit and one in the battery circuit, the ammeter being connected to either shunt by means of a double-pole double-throw switch. The generator is protected by an overload circuit-breaker which has a reverse-current trip to prevent current from flowing from the battery through the generator. This switchboard, like the one shown in Fig. 2, is arranged so that the battery can deliver current to the circuit in parallel with the generator, which permits the use of a smaller generator equipment than would be possible without a storage battery.

End-cell Regulation

Another method of regulating the voltage of the batteries is to vary the number of cells connected in series. Thus, for a 110-volt circuit the number of cells required is $\frac{110}{1.8} = 61$. The number in series when the battery begins to discharge is $\frac{110}{2.5} = 44$. Therefore an arrangement must be provided whereby $61 - 44 = 17$ cells may be cut out or switched in one by one. These are called "end cells." In practice the number of end cells is less than the number obtained by calculation because the voltage of 2.5 is not maintained after the charging current is cut off. The number of end cells may be determined by the formula

$$N = \frac{E}{1.8} - \frac{E}{2.3}$$

in which E = the voltage of the supply circuit. Fig. 4 is a diagram of end cell connections.

In switching from one contact plate to the next, the circuit must not be broken, neither must the blade touch two adjacent contact plates at the same time as this would short-circuit the cell connected to these plates. All end-cell switches are therefore provided with an auxiliary contact, either on the moving blade or fixed adjacent to each main-contact plate. The main and auxiliary contacts are joined by a resistance but otherwise insulated from each other. The auxiliary contact touches one of the switch plates while the main contact touches the adjace point. The circuit is therefore not i terrupted, being completed through th resistance which has too low a value affect the line potential appreciably. Th resistance, however, is sufficiently gre to prevent short-circuiting the cell co

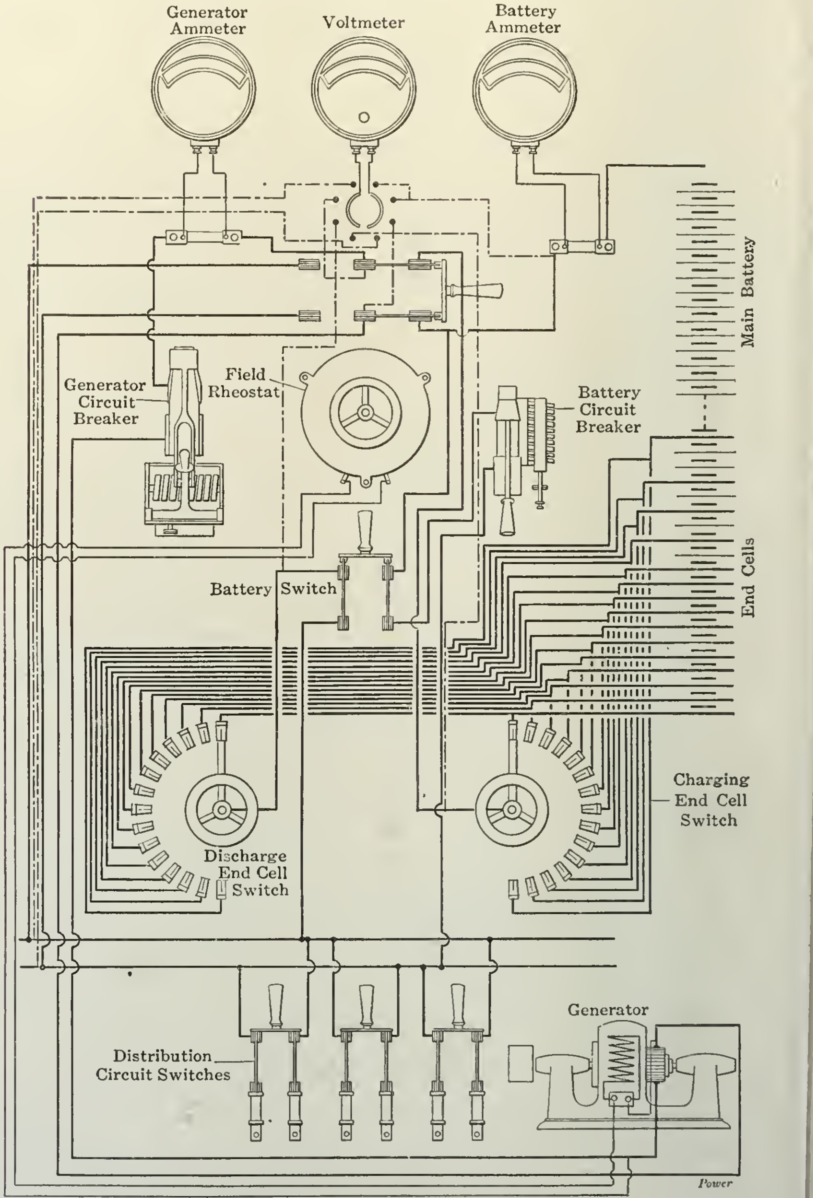


Fig. 6. Complete Switchboard for Charging and Discharging through End Cells

nected across the two plates. Fig. 5 illustrates the principle of this arrangement, though not the construction commonly used. The switch lever *L* carries two contact fingers, *A* and *B*, mounted on a block of insulating material *I* and connected to each other by a resistor *R*. The sketch shows the lever half-way between two "running" positions, the two fingers resting on different plates *C* and *C'*. At this moment the second end cell is connected to the terminals of the resistor *R* and the main circuit is connected directly to the plate *C'* by the main finger *B*. When the switch is moved far enough to the left to put both fingers on the plate *C*, the second end cell will be put in circuit and the resistor *R* cut out.

Since the end cells are gradually cut in as discharge proceeds, the main cells are discharged more than the end cells, the extreme cell being discharged least. While charging, all the cells are switched in and as the charge proceeds, the counter electromotive force of the cells increases, thereby cutting down the charging current. Consequently, it is advisable to cut out cells, one at a time, as the charge increases more and more. As the end cells were discharged less than the main cells, however, they become charged sooner, and therefore they are cut out, successively, beginning with the extreme end cell, which was discharged least. The connections of a switchboard arranged for high-voltage charge and end-cell discharge are shown in Fig. 6.

The battery is charged in one series directly from the generator, which has a voltage range to 155 volts, the charging current being controlled by the generator field rheostat. Two end-cell switches are required in order that the lighting circuits may be supplied while the battery is charging. The proper voltage for the lamps is obtained by adjusting the position of the end-cell switch connected to the lighting circuit. Two overload circuit-breakers are used, a simple one in the battery circuit and one with a reverse-current trip in the generator circuit.

Another method of voltage regulation is by means of counter electromotive force cells. These cells are connected in series with the operating cells with connections to an end-cell switch similar in every respect to the arrangement shown in Fig. 6, except that the end cells are connected so that their electromotive force opposes the battery electromotive force. As the battery voltage falls the end cells are cut out of circuit. This system is used only where the battery does not receive any attention while being charged. Since all of the battery cells are equally discharged they are all charged the same length of time, and it is not necessary to provide end cells to be gradually cut out as charge proceeds.

A 135,000 Volt Transmission System for Michigan

There is now under construction at Cook Falls, Mich., on the Au Sable river, a 9000-kilowatt hydraulic power station which is intended as the starting point of a 135,000-volt transmission system which will extend to Flint, 125 miles distant, and later to Battle Creek, 65 miles farther. The Cook Falls plant is the first of a series of four or five which, it is reported, will be established on the Au Sable river, and the available head there is about 40 feet. The generating equipment will comprise three 3000-kilowatt 60-cycle alternators mounted on horizontal shafts driven by waterwheels, with eight runners to each shaft. The voltage of these generators will be stepped up to 135,000 volts by delta-connected transformers installed at one end of the generator room.

The dam and power house foundation will be of solid concrete with molded water passages and draft tubes. The power house will be of brick above the water levels, and will have a floor space 40x110 feet. The 40-foot dam will impound a lake covering 2000 acres, which will store water to meet the daily variations in the station load. The Au Sable river has an uncommonly uniform flow, averaging 1100 cubic feet per second. The records of ten years show that at no season has the flood flow exceeded four times the minimum. This regularity of flow is attributable to the springs, emerging from the clay strata underlying the sandy surface of the northern part of the peninsula, which supply the river; there are only two insignificant tributaries to the stream in its whole length above the water-power site.

The transmission line will be a single three-phase circuit of three No. 0 copper wires carried on suspension-type insulators hung from the cross arms of 55-foot tripod steel towers. These towers are generally similar to those of the 110,000-volt transmission into Grand Rapids. Two braced bracket arms extend from one side, and one arm from the other, carrying the wires at the angles of a tipped isosceles triangle with a 12-foot base and 17-foot sides. The lowest wire will be 40 feet above the ground. The suspension insulators to be used will have eight disks linked in series, each disk having been tested to withstand continuously 75,000 volts, and subjected to 100,000 volts for a brief period. Each complete eight-disk insulator measures 52 inches from the tower hook to the line conductor.

The annual per capita consumption of coal in the United States constantly increases. In 1880 it was 1.4 tons per capita; in 1892, 2.3 tons; and in 1907 it was 5.4 tons.

LETTERS

Rotary Converter Commutators and Rings — Switchboard Operators' Salaries

Referring to the question of the care of rotary-converter commutators and collector rings, which has been discussed considerably in this department, I have found that the best lubricant to use on commutators is kerosene oil. I apply a small rag soaked in oil and follow it with a dry clean rag. By keeping the one rag well soaked with oil and wiping until no more dirt comes off, the machine will run for six to eight hours without giving trouble. This should be done when the machine is first started, and the commutator is cool. If the brushes are hard it is sometimes necessary to use a little dynamo oil or vaseline.

With the collector rings, if the proper tension is kept on the brushes, and they have plenty of contact area, one application of a heavy body oil when the machine is first started should last for a twelve-hour run; but the brushes and rings should be well cleaned after each run. Of course, with soft rings it is almost impossible to keep them from cutting a little. The main thing is to have plenty of contact area and as little tension on the brush as it will stand without sparking.

I was interested in the article concerning switchboard operators in the July 5 issue. I think the rate of pay is much too low when the education and training necessary to make a good operator, are taken into consideration.

H. C. Stiff.

Chicago, Ill.

One of the telephone journals tells about a small independent telephone exchange near Columbus, O., which has had some trouble in the operation of its lines at night, owing to the proximity of the high-potential, alternating-current circuits of the electric-light company. It is said that, owing to the induction, it is almost impossible to hear over some of the telephone lines because of the noise, but the kindness of the electric company enables the telephone people to do some business in this situation. When a night call comes in for a physician, or any call which is believed by the operator to be urgent, she telephones the lighting company to shut off the supply of electricity in the village while the conversation is being held. With this request the electric-light company obligingly complies, and the conversation is uninterrupted.

Gas Power Department

Elementary Lectures on the Gas Producer*

By Cecil P. Poole

Making Gas from Coal

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Coal consists of carbon, gases and one or two substances that will not burn and are commonly lumped under the name of "ash." If you make a hot fire and then dump fresh coal on top of the red-hot mass of burning coal, the heat will "distil" off the gases in the fresh coal and the carbon that is left will combine with the oxygen in the surrounding air, forming more gases.

This combination of carbon with oxygen is called "combustion" or burning, and if every atom of carbon unites with two atoms of oxygen, that constitutes complete combustion and the result is carbonic-acid gas, or carbon dioxide. The chemical formula for carbon dioxide is CO_2, meaning that the gas is composed of twice as many atoms of oxygen (O) as of carbon (C).

If the carbon in the coal cannot get into contact with more than equal parts of oxygen—that is, one atom of oxygen to each atom of carbon—then incomplete combustion is produced, and the result is carbon monoxide, represented by the symbols CO.

If the carbon unites with 1½ times as many atoms of oxygen as there are of carbon, then one-half of the carbon will be completely burned and the other half will be partially burned. This will produce both CO_2 and CO, but there will be 1½ times as much CO_2 as CO.

Now, if the CO_2 were passed into a bed of hot (not burning) coal, of the proper temperature, it would "pick up" another atom of carbon for each atom already in it and be changed to CO. On the other hand, since CO_2 is the result of complete combustion, it cannot be burned, and therefore will not unite chemically with oxygen.

Carbon monoxide, however, being the product of incomplete combustion, will burn, and if the CO produced in the process just described were brought into contact with more air, at a very high temperature, it would be burned and form CO_2 by uniting with the oxygen of the air.

When fresh coal is dumped on a hot fire, as described, the gases distilled out of the coal by the heat will either pass off into the atmosphere or be burned, according to the conditions immediately above the fuel bed. If the quantity of coal put on is not enough to deaden the fire and if sufficient air can get to the gases they will be burned.

All of the processes just outlined are carried on in a gas producer. Just above the grate the fuel is completely burned to CO_2; this is called the "combustion zone" (see Fig. 11). Immediately above this is a zone where two processes are supposed to go on; the excess air which was not used in the combustion zone unites with carbon to form carbon monoxide, and some of the CO_2 produced in the combustion zone also "picks up" carbon and is "unburned" to CO. This section is sometimes called the "partial combustion zone" and sometimes the "decomposition zone."

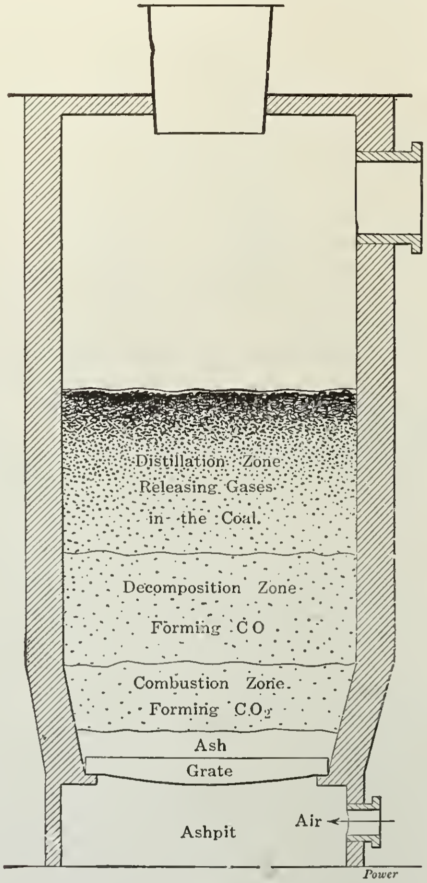


Fig. 11. Elementary Representation of Fuel Gasification, Using Air Alone

Above this is the "distillation zone" where the gases in the fresh fuel are distilled out by the heat in the gases coming up from the combustion and decomposition zones.

Fig. 11 is a diagrammatic representation of these zones and the processes which go on in them, as far as we have gone, but there is another process which goes on in most producers; that is the decomposition of steam into its component gases, hydrogen and oxygen. If air alone were admitted under the fuel bed of the generator, the combustion zone would spread to the top of the mass and the coal would simply be burned into CO_2; then the producer would be merely a stove and deliver only "dead" gases.

To prevent this, steam is admitted with the air, and the decomposition of the steam in the second active zone keeps the temperature down and prevents the combustion zone from spreading upward and burning all of the fuel completely. The oxygen of the steam combines with carbon in the second zone to form CO and the hydrogen goes on through and is taken out with the other gases.

It may be well to explain just here why "splitting up" a compound of gases causes a reduction of temperature. When carbon and oxygen unite to form either CO or CO_2, the heat in the carbon is liberated and it raises the temperature of the gases. Similarly, when hydrogen gas unites with oxygen to form steam (H_2O), the combustion liberates heat which raises the temperature. Now if either process is reversed, heat is absorbed instead of being liberated. Thus, if CO_2 is passed through hot carbon to form CO, some of the heat which was imparted to the carbon to make it "hot" is absorbed, and the temperature is reduced. Similarly, when steam is passed through hot carbon and "unburned" or decomposed into hydrogen and oxygen, as much heat is absorbed from the hot carbon as would be liberated by burning the same quantity of hydrogen. However, the oxygen combines with carbon to form CO, which liberates some heat, and the reduction of temperature due to decomposing the steam is partly neutralized.

For example, suppose that nine pounds of steam are decomposed, forming a pound of hydrogen and eight pounds of oxygen. "Burning" a pound of hydrogen into steam would liberate about 51,700 heat units; therefore "unburning" it will absorb about 51,700 heat units from the hot carbon. But the combination of the eight pounds of oxygen with six pounds of carbon to form CO will liberate about 26,700 heat units, so that the net result will be the absorption of about

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*This is the third lecture of the series.

$$51,700 - 26,700 = 25,000$$

heat units from the mass of hot coal in the second zone.

A mechanical illustration of the reversible heat process is afforded by the lifting and dropping of a weight. When a weight falls, energy is "liberated" and when it is lifted back to its original position, energy is absorbed by it. In other words, when a weight falls, work is done by the weight, and when it is lifted, work is done *on* it.

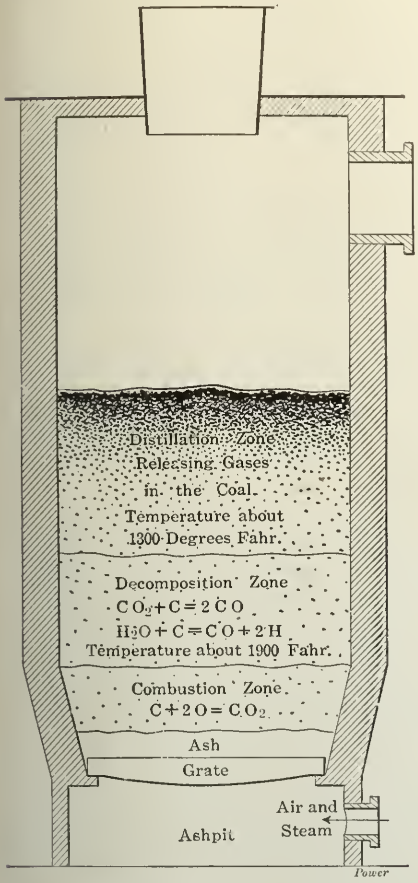


FIG. 12. APPROXIMATE CHARACTER OF THE ZONES OF A FUEL BED WHEN STEAM IS ADMITTED WITH THE AIR

Similarly, when any gas unites with oxygen, work is done *by* the gas—it raises the temperature of itself and the oxygen; when that same gas is separated from oxygen, work is done *upon* the gas—it absorbs energy in the form of heat. When carbon combines with oxygen it is necessarily in gaseous form, and is therefore considered as a gas.

Fig. 12 is an elementary diagram of the process of gasification of coal in a gas generator. It is the same as Fig. 11 with the addition of the effects produced by admitting steam with the air.

From what has been said it should be clear that while it is advantageous to send steam into the fuel bed along with the air, because it keeps the temperature down and permits continuous gasification, if too much steam is admitted the fuel bed will be cooled too much and the fire may be put out entirely. Even if that is not done, an excess of steam may reduce the temperature to such an extent that the production of carbon monoxide (CO) is greatly reduced and the operation of the generator impaired.

The question of operating temperatures will be fully discussed in a future lecture.

CORRESPONDENCE

Gas Engine Reliability and Economy

The June 28 number of POWER contained a very violent criticism by R. L. Paton of Mr. Beattie's article in a previous number. I find much in Mr. Paten's letter that is such rank injustice to the gas engine that it demands a reply.

Mr. Paton states that the gas engine is not as reliable as an electric motor but has established a reputation for unreliability. This one sentence shows that he has much to learn about a gas or gasolene engine. The charge is not true of engines built during the past ten years. It might have been true twenty years ago when this type of prime mover was in its swaddling clothes, but it has been doing more to live down its "past" than any other system ever has and at the present time can hold its own with any power source now available.

In court an expert witness has to qualify before his testimony is heard. Mr. Paton has handed out testimony without stating his experience. I will endeavor to "qualify" by stating that I have spent ten years with the gas engine as salesman, erecter and repairman with some of the largest engine makers in this country, but I am not connected with any of them now. My experience has not been confined to one town or city but spread over a large area.

I have seen many engines that were from twelve to fifteen years old doing daily work honestly and satisfactorily. I have found them working in plaster mills, sawmills, bakeries, blacksmith shops and other places where they were covered with dust, ashes or dough. The majority of them are run and cared for (?) by ignorant, unskilled laborers who do not know enough to oil a wheelbarrow when it squeaks. There are thousands of towns in this country that depend on the internal-combustion engine to supply them with water. There are hundreds of engines that are started in the morning with a measured supply of fuel and lubricating oil and left to run unattended until the supply is exhausted. This is the regular practice in many places. How many would or could pursue that method with a steam plant? I would ask Mr. Paton if he knows that there are thousands of gas engines that run from Monday morning until Saturday without a stop. In many of these plants an accidental stop would mean a loss of hundreds, and in some cases thousands, of dollars. There are engines that run three and four months night and day without stopping. Can anyone ask for any more reliable service than that?

Under equally good care, the gas engine is fully as reliable as a steam engine or an electric motor and will stand more abuse and neglect than either of them.

As regards operating cost, the gas engine will run for less than two cents per horsepower-hour, which would be equal to about 2½ cents per kilowatt-hour for electric power. As the average cost of current is from six to eight cents per kilowatt-hour, it can be easily seen that the gas engine is by far the cheapest power, even with city gas at 90 cents per 1000 feet.

H. K. WILSON.

New Bedford, Mass.

A Cure for Porous Gas Engine Cylinders

If the iron of a gas-engine cylinder is the least bit porous, more or less water is likely to seep through the water jacket wall into the cylinder, while the engine is standing idle for a considerable length of time, and this will make it very hard to start the engine. A remedy that is sometimes applied to porous cylinders is to fill the water jacket with a solution of copper sulphate or a solution of salammoniac, and allow it to stand over night. The next morning the solution is drained out and the water jacket is washed out with clean water. The copper sulphate or salammoniac, as the case may be, penetrates into the pores of the cylinder wall and causes a rusting action which fills the pores with a hard cement of iron rust, and this prevents any water from seeping through the wall into the cylinder when the engine is standing idle.

HOWARD M. NICHOLS.

Parnassus, Penn.

It is proposed in a patent issued to S. R. Bergman to improve the starting torque of a squirrel-cage induction motor by providing two end rings in parallel, one of low and the other of relatively high resistance. A laminated iron ring mounted on the rotor shaft is so arranged that it can be shifted to or from the low-resistance end ring and in this way vary the reactance of this circuit. When starting, the reactance of the low-resistance circuit is a maximum, and more current will be carried by the high-resistance ring than by the low-resistance ring, thus reducing the total current and increasing the torque.

Measuring Contents of Tanks

By F. L. Johnson

> *Simple directions for making chart and measuring stick that will show the contents of horizontal cylindrical tanks in gallons, feet or pounds.*

Because the calculation of the area of a segment of a circle involves processes which are outside the realm of pure arithmetic, many engineers balk at problems in which segmental areas are a factor. In gas works, oil refineries, and in power plants where liquid fuel is burned in the engine cylinders or under the boilers, the oil or other liquid is most commonly stored in horizontal cylindrical tanks and as an exact knowledge of the quantity on hand at any time or the amount used from hour to hour or from day to day is of no little importance to the engineer, he seeks for a method by which the contents of a tank of any diameter and length with any depth of oil may be readily and accurately calculated.

If the containing tank were rectangular or circular and stood on end, its contents could be easily ascertained by multiplying the area of the base by the depth of the liquid, and as finding the areas of rectangles and circles is a comparatively simple matter, the problem would be simple. But when the circular tank lies on its side the problem becomes an intricate one involving calculus, for the ratio of content to its depth varies with every change in depth, and unless some short cut can be made the process is one so tedious that the ordinary man shrinks from attempting it, and instead makes a rough estimate of the contents.

If it is remembered that all areas are proportional to the squares of the length of their corresponding lines, and decimals are understood, a short time spent on the table of segmental areas to be found in most engineer's pocketbooks will enable anyone to prepare a table, chart or measuring rod, which will show at once the cubical contents in pounds, gallons or cubic feet of a horizontal cylindrical tank of any dimensions.

In this process the first step is to find the area of the segment of a circle the diameter of which is unity. This area is found by the use of the accompanying table.

It will be noticed that one of the columns in the table is marked hight and the other area, which means that the segment, the hight of which corresponds to the fraction found in the hight column, will have the area found opposite it in the area column, when the diameter of the circle is unity. The fraction in the hight column is obtained by dividing the hight of the segment by the diameter of the circle of which it is a part.

To use the table for finding the area of the segment of any circle divide the hight of the segment by the diameter of the circle and multiply the area found in the table opposite the resulting fraction by the square of the diameter of the given circle.

For example, the hight of a segment is 9 inches and the diameter of the circle is 6 feet, what is the area of the segment?

Dividing 9 by 72 gives 0.125. Looking in the table the fraction 0.125 is found opposite the number 0.056664 in the area column and is the area of the segment when the diameter is unity; but the diameter of the circle is not unity but 72 inches, or 6 feet. As it has been remembered that all areas are proportional to the squares of the length of their corresponding lines, it is seen that if the number denoting the area found in the table be multiplied by the square of 72 or of 6 the area of the segment of the given circle will be found in square inches or in square feet, according to which dimension is used in the calculation.

TABLE OF AREAS OF SEGMENTS.

Hight.	Area.	Hight.	Area.
0.005	0.000471	0.255	0.157891
0.01	0.001329	0.260	0.162263
0.015	0.002438	0.265	0.166663
0.02	0.003749	0.270	0.171090
0.025	0.005231	0.275	0.175542
0.03	0.006866	0.280	0.180020
0.035	0.008638	0.285	0.184522
0.04	0.010538	0.290	0.189048
0.045	0.012555	0.295	0.193597
0.05	0.014681	0.300	0.198168
0.055	0.016912	0.305	0.202762
0.06	0.019339	0.310	0.207376
0.065	0.021660	0.315	0.212011
0.07	0.024168	0.320	0.216666
0.075	0.026761	0.325	0.221341
0.08	0.029435	0.330	0.226034
0.085	0.032186	0.335	0.230745
0.09	0.035012	0.340	0.235473
0.095	0.037909	0.345	0.240219
0.10	0.040875	0.350	0.244980
0.105	0.043908	0.355	0.249758
0.110	0.047006	0.360	0.254551
0.115	0.050165	0.365	0.259358
0.120	0.053385	0.370	0.264179
0.125	0.056664	0.375	0.269014
0.130	0.059999	0.380	0.273861
0.135	0.063389	0.385	0.278721
0.140	0.066833	0.390	0.283593
0.145	0.070329	0.395	0.288476
0.150	0.073875	0.400	0.293370
0.155	0.077470	0.405	0.298274
0.160	0.081112	0.410	0.303187
0.165	0.084801	0.415	0.308110
0.170	0.088536	0.420	0.313042
0.175	0.092314	0.425	0.317981
0.180	0.096135	0.430	0.322928
0.185	0.099997	0.435	0.327883
0.190	0.103910	0.440	0.332843
0.195	0.107843	0.445	0.337810
0.200	0.111824	0.450	0.342783
0.205	0.115842	0.455	0.347760
0.210	0.119898	0.460	0.352742
0.215	0.123988	0.465	0.357728
0.220	0.128114	0.470	0.362717
0.225	0.132273	0.475	0.367710
0.230	0.136465	0.480	0.372704
0.235	0.140689	0.485	0.377701
0.240	0.144945	0.490	0.382700
0.245	0.149231	0.495	0.387699
0.250	0.153546	0.500	0.392699

Having mastered the process of computing the areas of segments by the use of the table the utilization of this knowledge in finding the contents of partially filled tanks is an easy matter as the area of one end of any regular body multiplied by its length gives its cubical contents.

The table gives the areas of 100 segments from 1/200 to ½ of the diameter in hight. But only for certain diameters would any of the hights be found in even inches, but this is of no importance as the rod with which the hight of segment is measured may be graduated in fractions of the diameter instead of in inches, or the areas for even inches or half inches in hight may be calculated and inserted in the table.

If a measuring rod of a length equal to the diameter of the tank be marked off into 200 equal spaces and marked from both ends toward the middle with numbers corresponding with the fractions representing hight in the table, then if the tank is one-half or less full, the reading will tell the hight of the segment of liquid. If more than half full it is the hight of the empty space that will be measured.

It is not necessary in these calculations that the dimensions in inches or feet enter into them at all except when the diameter of the tank is squared.

To make a chart which will show the contents at any time, or the quantity drawn out during any period, make a column of numbers corresponding to those on the suggested measuring rod, calculating the area of each segment in square inches or square feet as desired and multiplying it by the length of the tank for the volume.

When 100 computations have been made, repeat them in inverse order.

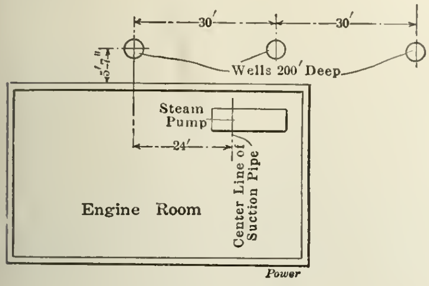
To use the chart, measure the depth of liquid in the tank with the graduated rod. If the tank is more than half full the number on the rod referred to the chart will give the contents of the empty space above the liquid, which, subtracted from the volume of the tank, will show how much remains. If the tank is less than half full the reading will be the quantity in the tank and the difference between any two readings will show how much has been taken from or added to the contents between the measurements.

On the measuring rod the chart itself may be made by putting the figures denoting the contents on the side opposite the one denoting the hight. Or the rod may be made square with the hight on one face; contents in cubic feet on the second, in gallons on the third and the fourth side may be graduated in inches, or if the specific gravity is constant the weight in pounds for each graduation may be calculated and marked on the rod instead of some other measure of capacity.

Readers with Something to Say

Pumping Layout for Criticism

I should like opinions as to the proposed plan for pumping water from artesian wells which is shown in the accompanying sketch. It is proposed to install one steam pump in the engine room to draw water from all three wells simultaneously and discharge into a tank 75 feet above the ground. We hope to draw 75 gallons per minute from each well. Will one pump do the work or must there be a separate pump for each well? Would compressed air be more satisfactory and how will compressed air compare with the steam pump for economy? How many cubic feet of free air per minute will be required to do this work, taking air at 90 pounds gage?



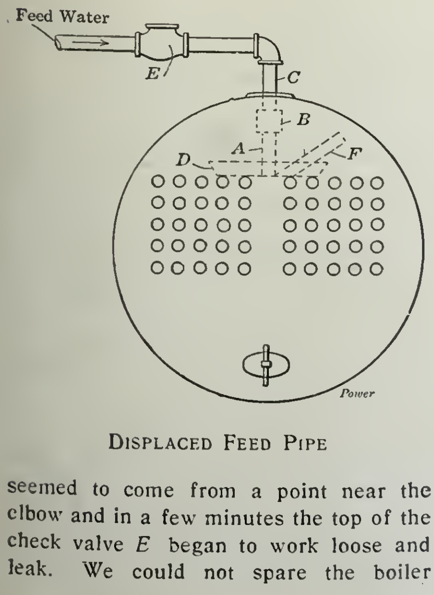
PUMPING LAYOUT

H. B. FILLMEN.

Pottstown, Penn.

Water Hammer in the Feed Line

A few days ago a very sharp water hammer suddenly developed in the feed line of one of our boilers. The sound seemed to come from a point near the elbow and in a few minutes the top of the check valve *E* began to work loose and leak. We could not spare the boiler until Sunday (two days off) so we had to keep an eye on the check valve and continue running. Saturday about 9 a.m. the nipple *C* split from the boiler shell and it was with much difficulty that we were able to make the balance of the run. On cutting out the boiler we found that the pipe *A* had worked out of the coupling *B* and had fallen over into the position *E* in the pan *D*, whose mission it is to keep the lower end of pipe *A* submerged and prevent water hammer. A new nipple was inserted and we had no more trouble.



DISPLACED FEED PIPE

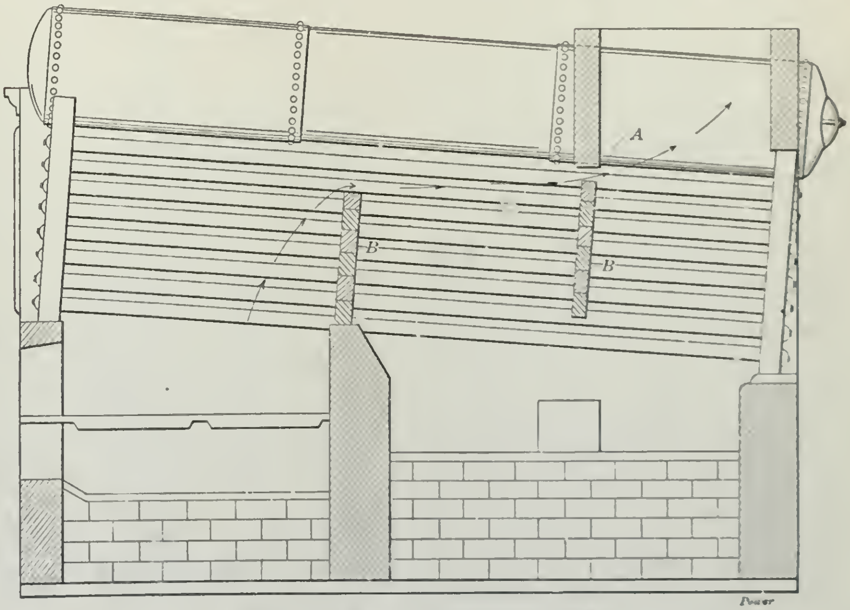
R. M. ORR.

Brantford, Ont.

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Why the Boiler Would Not Steam

The accompanying sketch shows the elevation of a water-tube boiler, two of which were installed in a plant where the writer was employed. These boilers were 300 horsepower each. When No. 1 boiler was put in commission it was found to be satisfactory in every way, but when No. 2 boiler was fired up and put in commission and No. 1 cut off, it was impossible to carry the load with No. 2 and No. 1 had to be cut in again. As the boilers were exact duplicates, No. 2 was cut out and cooled down and a search made to locate the trouble.



ROW OF BRICKS OMITTED AT "A"

It was finally located at the point marked *A*. When the boiler was erected the top row of baffle bricks *B* was not put in at *A* which allowed part of the heat to pass direct to the stack as indicated by the arrows. This small opening reduced the effective heating surface to such an extent that the boiler would not carry half of its load. After the baffle was repaired the boiler gave satisfaction and was accepted.

This was a good object lesson, and showed the importance of having all baffles properly put in, as a small opening has a greater effect upon the steaming capacity of a boiler than is generally supposed.

R. L. MOSSMAN.

Tampa, Fla.

Dismissal without Cause

Some time ago a young engineer accepted a position in a fairly large plant because the pay was better than that which he had been receiving. His watch was from 4 p.m. till midnight and an oiler came on every night at 5 p.m. and worked till 6 a.m.—a part of two engineers' shifts. The duties of this engineer were to look after four engines and generators, one blower unit and one ice machine, doing his own oiling from 4 to 8 p.m. and helping out in the boiler room whenever necessary. The conditions were such that the ice machine had to be shut down every day from about 3 to 7 p.m. to enable the generators to carry the load. Therefore, this young engineer almost always found the ice machine shut down when he went on duty.

One day after he had been there a short time, he noticed several men coming into the engine room and asking for the chief, they having seen an advertisement in a paper for an engineer at this plant. Supposing that he was to lose his job, the young engineer went to the chief and asked for explanations; he received the following reply: "You seem to know your business and you are a good hard worker, but you are just hoodooed, that is all." Of course, the assistant laughed and insisted upon a further explanation. The chief then gave it without hesitation. He said, "Well, too many things have happened to you in too short a time. First, this bearing warming up, then the square end of that suction-valve stem breaking off and also the burning of the leather cups in the stuffing box of the ice machine; all happening within 4 days. I do not blame you so much but you are just hoodooed, that is all." Not a very good reason to discharge a man for, who "knew his business and was a good hard worker."

Now I am in a position to explain these three happenings, for I am the young assistant engineer referred to.

First, the warm bearing was on a large high-speed engine at the back of which was being dumped all the material for the erection of two water-tube boiler foundations, the dust of which settled all over the engine. The bearing had run for one day and it was impossible to shut the engine down on account of the load. I discovered the warm bearing myself, cooled it and ran the engine through. It is still running.

Second, upon first taking the position, I found the square end of this valve stem twisted about one-half way around, someone having been in the habit of putting a wrench on the valve to open or close it. I started to open the valve after pumping the gas out of the machine and the square end twisted off. The engine was still turning over slowly and the low-pressure gage hand was moving, so I knew the valve was either open or was leaking badly. I let it run and found that the machine had been shut down, the discharge valve closed and the suction valve left open, a poor practice in refrigeration. I reported all conditions and what I had done to the first assistant, he being the man whom I relieved.

Third, on the day the cups were burned the ice machine was running when I went on. After feeling all the bearings in the main engine room, I went to the ice-machine room to look at conditions there. It was 4:10 p.m., just 10 minutes after starting to work, and I found the rod blue and hot. I gave the machine more gas and let up on the stuffing-box gland and remained right with it till it began to cool off, then called the attention of the first assistant.

I am not writing this with any malice toward anyone as I was lucky enough to get a position nearer my home and working six days a week at the same salary for which I had been working seven. But I should advise all engineers to make a thorough inspection of the whole plant, both upon going on and off duty. And I hope my experience will teach some chief engineers to go to the bottom of everything before discharging anyone.

R. P. Wilcox.

Chicago, Ill.

What Causes the Vibration?

We have two 72-inch by 18-foot return-tubular boilers carried on Gallus frames. The service is such as to require only one boiler to be operated at a time. No. 2 boiler gives no trouble whatever, but when No. 1 is used, violent vibrations are set up in the boiler and in the steam piping. Unless the engine is throttled so as to wiredraw the steam, the vibration is as great at no load as at full load.

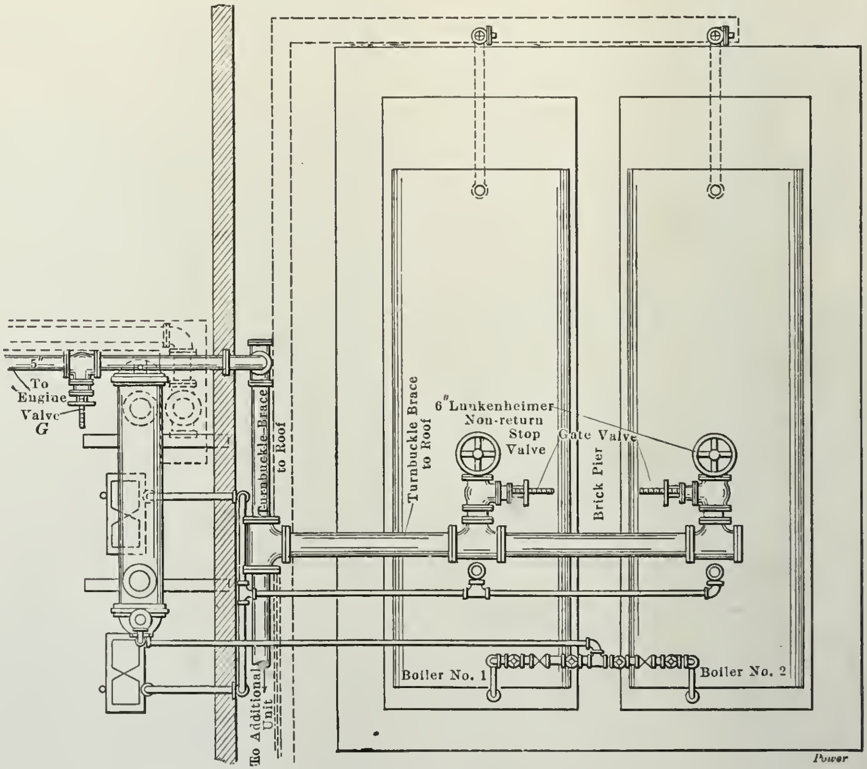


Fig. 1. Plan of Piping from Boilers

The piping is suspended, for the most part, by turn-buckle hangers secured to the roof. The engine is a 14x16-inch single-valve Fleming, running at 225 revolutions per minute.

The accompanying diagrams show the layout of piping, and the writer would be pleased to note the opinions of Power readers as to the probable cause of the trouble.

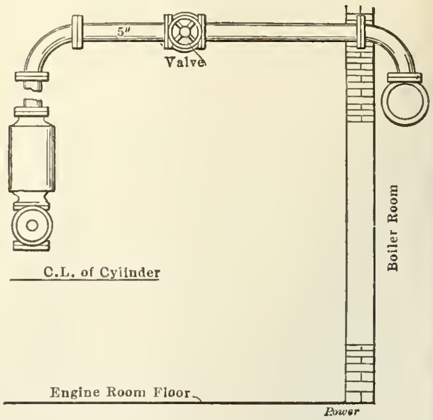


Fig. 2. Elevation of Engine-room Piping

E. S. Davis.

Logan, O.

Pump Problem

Will some reader of Power kindly solve the following problem? What will be the necessary steam pressures in the high- and low-pressure cylinders of a 10, 16 and 8 by 18-inch compound boiler-feed pump, working against 185 pounds boiler pressure at 35 strokes per minute?

W. E. Thompson.

Redondo Beach, Cal.

Finding Areas of Irregular Figures

For irregular figures where a planimeter is not available, but where a laboratory balance is, areas may be obtained with a considerable degree of accuracy by cutting the diagram out, together with a unit area of 1, 2 or 4 square inches from the same sheet of paper, and weighing the two pieces of paper thus obtained; the areas being proportional to the weights. In cases of very broad lines bounding the diagram, it is better to trace the figure on another piece of paper, following the center of the bounding line, in order to get rid of the weight of the ink.

Donald M. Liddell.

Grasselli, Ind.

Operating Mechanical Tube Cleaners

There are different methods of keeping boilers free from scale; some use filters and open heaters, thereby preventing the scale-forming matter from entering the boiler; others use compounds to nullify the scale-forming tendencies, while still others let it form and then get busy with kerosene, soda ash and a mechanical tube cleaner.

For the benefit of those using the latter method, I will relate my experience. Our boilers are of the horizontal tubular type, 6x18 feet with ninety-two 2-inch tubes. The feed water is taken from a mountain stream that runs over limestone rocks. There are no filters and the feed water is heated only to 140 degrees before entering the boilers. We run 24 hours per day and during the winter months have no boiler to spare for the purpose of cleaning. Consequently, by spring the boilers are full of scale. We tried a compound but even with frequent blowing down could remove but little. As the mud always accumulated just over the bridgewall, it was considered more dangerous than the scale; hence a mechanical cleaner was procured.

The company took the stand that the cheapest man in the plant was all right to run it, it not making any difference whether the man knew that continuous hammering will make an iron bar become thinner, and that if steam were kept on the cleaner, it was necessary to keep it moving along the tube; for if this were not done, the tube would be stretched, made thinner, and covered with small bulges, which would tend to weaken it.

So the ash shoveler was put to work cleaning the first boiler. After he had finished, the boiler was filled and steam gotten up. When it had been cut in a couple of hours, the fireman reported it to be taking too much water. I told him that was because it was clean and pointed out the economy of a clean boiler. When the load went off on Saturday night, this boiler was the first to have its fire drawn, and the fireman reported that it was taking just as much water as when making steam. I accordingly opened the combustion-chamber door and looked in. It had the appearance of a waterfall—most every tube was leaking. We had to replace seventeen out of the ninety-two tubes. Those which were taken out had bulges all over the surface.

I placed one of them across two horses and sent for the boss. When he arrived I put the mechanical cleaner in the old tube, turned on the steam and held it in one place about a second; the outside of that part of the tube immediately began to swell. The boss said, "That is enough, we will use it no more." But I told him it was not the fault of the cleaner but of the man who used it. I then proceeded to run it up and down the tube without injuring it. Since then the ash shoveler has not been allowed to run the mechanical tube cleaner but we allow him to use the scraper.

Gilbert Guèrin.

Adams, Mass.

Trouble with an Oiling System

For the benefit of those engineers who have not had experience with continuous lubrication I will relate some of my experience during the past year. The plant contains two direct-connected high-speed units, each of 100 kilowatts capacity. The engines are of a popular make and are doing satisfactory work. I took hold of them the day after they were delivered to the owners and thought I had drawn a prize. The oiling is from a gravity tank bolted on the top of the frame and feeding through sight feeds to the various bearings. After doing its work, the oil drains back to a small filter attached to the subbase and from there is pumped back to the tank. This cycle is continuous while the engine is in operation. The pump plunger is driven from the valve stem of the engine and consequently the stroke varies every time the load affects the cutoff. Right here has been all the trouble with the system. This may not appear to be of much importance, but when a person sees the oil in the tank getting continually lower, without being able to make the pump take hold, it naturally makes him anxious.

It was a cold day when I first took charge of the plant. The engine started off smoothly and the oiling system worked nicely for about an hour, then it began to fail. I tried all the things I knew and by keeping at it persistently got through the day without losing all the oil in the tank. The following day the other unit was used and worked without a hitch; it has done so ever since with absolutely no trouble, but I have had many unhappy moments with the first one. Some days it was all right, on others it was all wrong.

When the weather became warm, however, the pump worked as well as one could wish. This gave me the clue. In cold weather the temperature of the part of the room in which this engine was located had been low enough to cool the oil to a point where it would not flow fast enough to fill the pump at the speed the plunger was moving.

While all this was happening I was having trouble with the packing on the plunger. It would remain tight for a few days and then all at once would start leaking. On removing the plunger it was found to be badly worn. I had a new one made and after assembling, packed the pump with the hardest wick packing obtainable, in place of a good grade of soft wick formerly used. Since this was done there has been little trouble with the pump and the engine room can be five to ten degrees cooler than before.

G. F. Barker.

Dorchester Centre, Mass.

Rumbling in Boiler

I have charge of a battery of boilers, one of which acts very much out of the ordinary. In the morning before starting, the pressure runs up until the safety valve, which is spring loaded, begins to sizzle; immediately there is a rumbling noise inside the boiler that sounds like a jet of steam in cold water. There is a trembling of the boiler, and the steam-gage hand drops gradually from 128 to about 90 pounds and then returns to 128. It happens every morning, and I cannot account for it, and though I have had charge of steam plants for fourteen years I never before saw anything like it.

M. H. Tarbell.

Cincinnati, O.

Packing a Feed Pump

Will someone kindly advise me as to the best method and material for packing the plunger of a boiler-feed pump handling water at 100 to 195 degrees Fahrenheit against a boiler pressure of 150 pounds? I have experienced considerable trouble with our pumps, two weeks being the longest period that any packing lasted.

George Kramer.

New York City.

It is reported that the town of Windsor, Ont., has decided to purchase electric power from Niagara. The 110,000-volt line to London, Ont., will be extended over the 108 miles separating Windsor from this city. The expense of building the line to the city limits will be paid for by the Canadian government. Windsor, which is a small town, cannot use more than 2000 horsepower, but it has contracted to sell a large share to the city of Detroit, which is directly across the river.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

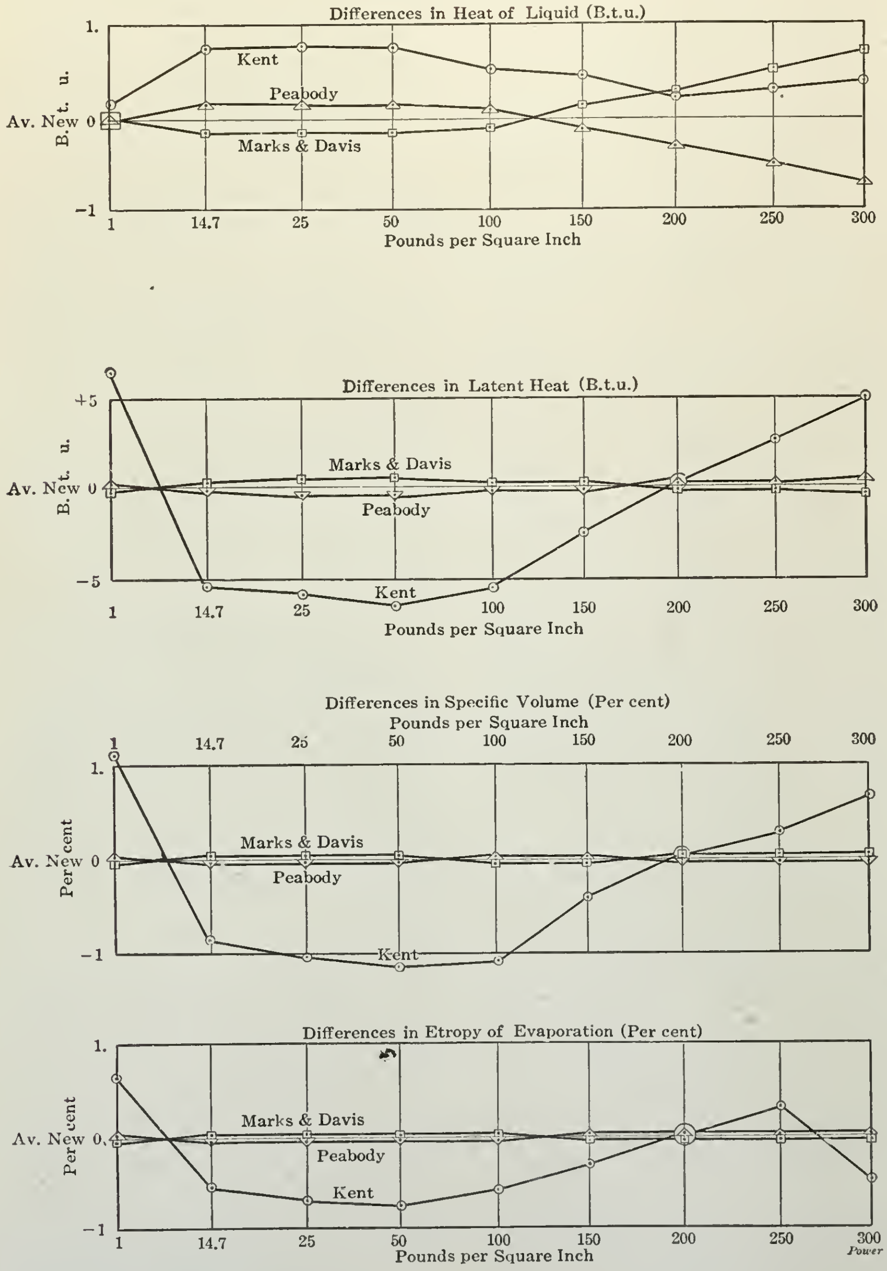
Steam Tables

The accompanying chart of relative differences of values is taken from steam tables, old and new, showing how the latest tables differ from those which have been long in use. The new tables of specific volumes that were published in a recent issue of POWER may make this of interest to some of the readers.

In each case the base line for the curves is taken as the mean of the values in the tables by Marx and Davis and by Peabody. The differences are laid off below the mean when of less value than the mean and above when greater than the mean. As is shown, the new values of the specific volumes and entropy of evaporation are substantially in agreement, while the heat of the liquid and the latent heat show a considerable divergence.

DONALD S. SAMMIS.

Stratford, Conn.



COMPARISON OF NEW STEAM TABLES (PEABODY-MARKS-DAVIS) WITH OLD STEAM TABLES (KENT-1905)

Central Station versus Isolated Plant

The never-ending discussion of Central Station versus Isolated Plant seems to be very much alive in some of the recent issues of POWER, with the general result that, "A man convinced against his will is of the same opinion still."

With no hope, therefore, of its having any other effect than to add perhaps a little fuel to the flames, I will take exception to that particular part of the argument which is based on the fact that central stations charge large consumers three or four cents less than an average price, and small consumers three or four cents more.

Electric current is a commodity which is made and sold on much the same basis as any other commodity. Those charges which go to make up more than 50 per cent. of the net cost are directly affected by the amount of current delivered to one consumer. It is exactly the same reason which makes it possible for an isolated plant of large capacity to produce current at a price per kilowatt-hour less than that for which it can be produced in a small plant.

If the large consumer is to be forced by law to pay anything like the price paid by the small consumer for electric current, then he should be forced to do likewise with every commodity which he buys, whereupon our socialistic friend will find considerable fault with his living expenses increasing somewhat more than they have already.

The statement has also been made that certain isolated plants produced current so cheaply that the central-station supply would have to be charged for at the rate of ¾ of a cent per kilowatt-hour. It would be interesting to see by what process of figuring any kind of a plant to produce electric current can be installed at a price the fixed charges on which would be as little as ¾ of a cent per kilowatt-hour.

The writer has designed and installed many isolated plants under conditions both favorable and unfavorable; but he has yet to hear of one the investment in which alone represented a fixed charge of less than ¾ of a cent per kilowatt-hour, although he has ben assured by many manufacturers of apparatus that it was very customary to find plants in which the fixed charges were less than ½ of a cent per kilowatt-hour. Persistent inquiry has failed to reveal where these plants are located. The writer is still obliged to believe that the experience of engineers in general will not substantiate any such statements.

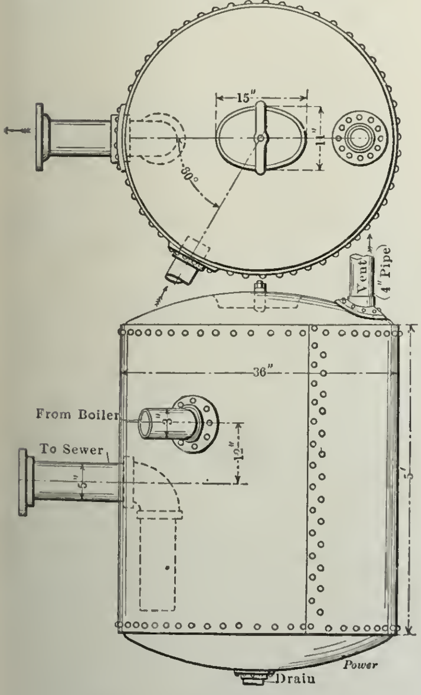
The foregoing is not intended to convey the idea that the writer does not believe that isolated plants cannot compete with central-station service in some instances; but it is intended to convey the idea that central stations approach this question with a large amount of accurate information to back up their statement, and that they are not inclined to either "bulldoze" or "hypnotize" the consumer into signing a contract which is distinctly to his disadvantage. The consumer always has the privilege of going out and employing a competent engineer to analyze central-station figures; and consequently does not need much sympathy if he does not show the same acumen in regard to this item of running expense as he does in purchasing his other supplies.

WARREN B. LEWIS.

Providence, R. I.

Accident to Blowoff Tank

In the May 17 issue of POWER, E. H. Long describes a blowoff tank which failed at a hotel in Kansas City.



SHEET-IRON BLOWOFF TANK

In my estimation the manner in which the tank was connected was responsible for the accident. His sketch shows the blowoff pipe from the boilers entering the tank near the top and continuing downward inside and the overflow to the sewer leaving at about the same level. I think that these two connections should have been reversed.

If the water in the tank is hot, due to a leaky blowoff valve or some other cause, there will be a vapor present in the tank above the water. Now, if the pressure is even slightly increased by starting to blow down, the vapor is immediately condensed and there is nothing left for the water to compress, and thus form a cushion. The result is a water hammer in the tank, which is liable to crack or break the cover.

I submit a sketch of a blowoff tank which I am about to install. This tank is special, and is to replace a tank which failed in the same way as did the one Mr. Long described and cost the life of my predecessor.

The shell is made of ⅜-inch sheet iron and the heads of ½-inch material. The cost is less than that of a cast-iron tank.

I would be glad to have POWER readers criticize my design.

R. E. COOPER.

Cleveland, O.

Perpetual Motion

Mr. Kirlin's perpetual-motion machine, described in a recent issue, calls to mind the accompanying diagram of a scheme which is submitted to the readers of POWER as being more plausible and having less gear than Mr. Kirlin's device. The basic idea is the same, viz., the getting of something for nothing, which does not occur in New York.

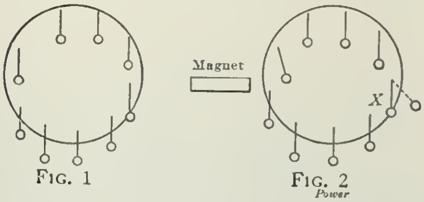


FIG. 1 FIG. 2

DIAGRAM OF MAGNETIC PERPETUAL-MOTION MACHINE

Fig. 1 illustrates a disk mounted on an arbor and having on its periphery a number of studs. Suspended from each stud is a short bar magnet made of round steel. The axes of these suspended magnets are held parallel to the axis of the disk-supporting spindle.

Fig. 2 is the same disk save that a bar magnet has been so placed that the suspended magnets are repelled as they pass in front of the stationary magnet.

Obviously the system is unbalanced and rotation seemingly would take place. Another magnet could be similarly placed at *X* so as to repel the suspended magnets away from the center.

Naturally the speed of such a device must be very low to neutralize the effect of centrifugal force. There would be some retarding effect due to eddy currents set up in the supporting disk. The appliances could be run (?) in a vacuum and thus the resistance of the air would be eliminated.

There is a fallacy somewhere but it is not visible to me.

WESLEY MCARDELL.

Brooklyn, N. Y.

Engineers' Wages

As to the suggestion of Waldo Weaver in POWER of May 17, that if engineers would furnish POWER with information concerning their wages, hours of work and pay for extra time, etc., it would add to the means of getting increases in wages, I think it is a good one.

I am in charge of a 760-kilowatt station which contains six dynamos, five engines, three pumps and three water-tube boilers. We are preparing to install a 500-kilowatt low-pressure turbine and other apparatus.

Having had a little experience with boilers and engines, and desiring more, I went into this plant as fireman to learn engineering and by keeping my eyes and ears open, and making it my business to know how and why things about the plant were done; by studying catalogs, instructions, etc., on the apparatus in the plant; by assisting to make repairs and by watching and remembering how they were made, I gradually advanced to take charge of the plant. My wages increased from time to time, from $50 to $90 per month.

On several occasions, knowing that it would be quite a proposition to get the desired increase and confident that the position was worth more, I looked around for a job that offered more than I was getting. Then having this place open, I would put it up to the "Old Man" and I always got my raise. The "Old Man" here knows what he is paying out wages for.

It is hard to show some men that their engineer is worth more money; it is hard for some engineers to approach the "Old Man" for more money, and it is hard for some men to make themselves worth more money. I knew of an engineer in a small plant who wanted a raise but would not ask for it and quit, thinking that the "Old Man" would offer him more to come back. This is a poor way to get an increase in wages.

I average 10 hours per day and get no pay for extra time. I do not lose any pay when I am off for a day or so, or am sick.

It pays one to do his work as thoroughly as if he were doing it for himself; gain the confidence of the "Old Man," be truthful and faithful.

The responsibilities of operating engineers have grown much faster than their salaries. The power-plant engineer must be continually studying to keep up with his profession. I do not know of any other class of skilled mechanics that must study, plan, think, work long hours and carry responsibilities and yet receive such small salaries as the stationary engineers.

C. E. NIGH.

Morgantown, W. Va.

The Designer Can Save Repairs

I read with interest the article in the June 7 number entitled "The Designer Can Save Repairs." It interested me particularly because I had met the same trouble in the erection of a small upright engine. The parts came disassembled and I put the flywheel and disk on the shaft while on the floor. After considerable twisting and turning I managed to slip the disk through the frame.

The designer cannot be expected to foretell, however, all the things that might result from any particular design of apparatus. We have, for example, a number of 10-inch gate valves on the main steam header. Attached to each valve is a bypass. The bypass is in the form of a gooseneck around the body of the valve as shown in Fig. 1. The bend was so short and ridged that we were continually bothered with joints blowing out. To remedy things we removed the old bypass entirely and tapped out the holes for a 1-inch extra-heavy pipe and connected up as shown in Fig. 2. This arrangement gave free expansion and we haven't been bothered since.

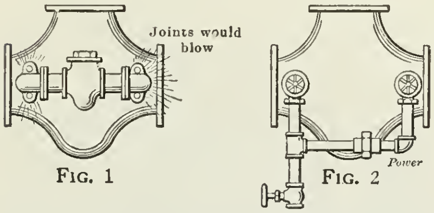


FIG. 1 FIG. 2

ORIGINAL AND IMPROVED DESIGN OF BYPASS

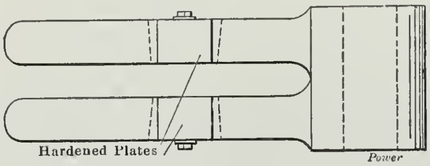


FIG. 3. CAM WITH PLATES TO RECEIVE WEAR

In another case the designer failed to foretell the possible wearing down of a cam that opens the exhaust valves, which are of the single-beat poppet type, on one of the low-pressure cylinders. This necessitated the dismantling of the whole valve casing to remove the defective cam. We now have the cams planed out and a hardened plate dovetailed in as shown in Fig. 3.

H. R. BLESSING.

Philadelphia, Penn.

Loose Crank Pin

In the June 7 issue, Mr. Forgard asks for information and advice about a loose crank pin on his engine. It is hard to assign a cause for a crank pin becoming loose, as there are so many causes. Bad valve setting or working under a high steam pressure will often loosen a pin. A continual sharp jar will sometimes start a pin. When a pin gets loose, bradding or driving pins or screws have very little merit as a remedy.

The only satisfactory method of dealing with a loose crank is to remove the pin and put in a new one.

It may be necessary to bore the pin seat so that if will receive the new pin. This can be done in a simple manner right in the engine room by the use of a hand boring bar, which is shown in Fig. 1. The bar *O* is supported by two bearings *X*, which are blocked away from the disk and clamped in the manner shown. The cutter *P* should be large enough to clean up the old bore. The bar is fed by the screw *E* and nut *D*, which must be supported from some part of the engine frame or crank disk. The bar may be turned by a hand crank applied to the square head at *C*.

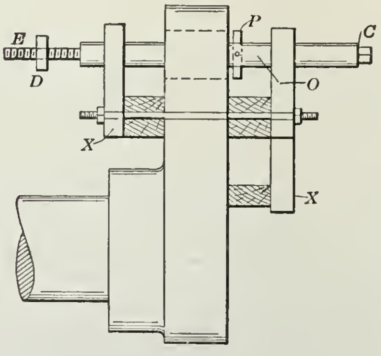


FIG. 1. ARRANGEMENT FOR BORING

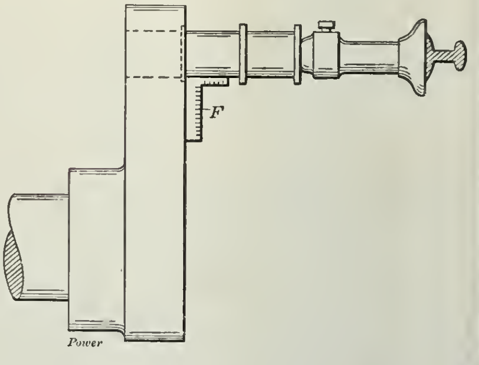


FIG. 2. JACK IN POSITION TO PRESS IN THE PIN

When the seat has been bored out the new pin can be put in place by heating the disk and inserting the pin. Another way in which the pin can be put in is by pressing it in with a hydraulic jack, as shown in Fig. 2. A yoke made of railroad iron or other strong material must be erected to support the jack. A small square *F* should be used to set the pin true before it is pressed.

A pin with a collar on it, as shown, will make a better job. The seat may be counterbored to receive the collar.

Sheffield. Ala. C. R. MCGAHEY.

Boiler Explosions

The letters recently published on this subject have been very interesting to me, perhaps more than ordinarily because I have had experience with a return-tubular boiler which was operated without any exact knowledge of where the water level was.

At the time I was only a boy and was employed to fire that boiler. I had to get steam up in the morning and keep it up as long as the inspirator would supply enough water to keep water in the boiler.

There was no license law in force at that time, and the owner of the shop was boss. I had told him that every morning the water was low and he said to "fire up easy like" until I had steam enough to start the inspirator, then just hold the steam steady until water showed in the glass and then "put the fire to her."

As I have stated, I was a boy at the time, and I felt that I must do as the boss told me.

I had an idea that this practice was not good, yet nothing unusual ever happened. When the inspirator could not keep the water in the glass, we either gave that leaky old boiler a bran mash or shut off the engine until the inspirator got a "head-start" again.

Readers may think that this was too bad a condition to last, but I know that this method of operation continued for months.

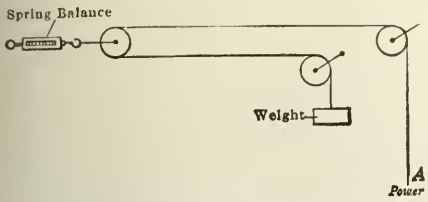
One of the shop hands told me before I came to fire the boiler, that the shop had been shut down for a few days and a son of the boss had emptied the boiler, and cleaned it out but had not filled it up again. A man from the shop attempted to get up steam and a fire was under the boiler long enough to generate plenty of steam but the gage showed that there was none. Investigation showed that the boiler was dry and red hot. When they looked around to find the leak they found that the blowoff valve was wide open.

This boiler has not exploded yet and was still being used a few years ago, carrying 85 pounds gage pressure.

Cleveland, O. H. W. BENTON.

Troubles with the Tension Carriage

The letter by T. C. Thompson bearing the above title in the July 5 number is accompanied by a sketch which makes the reason for the trouble clear. With the rope rove as shown, Mr. Thompson is right in his opinion that the tension sheave should be located between the idler sheave and the line-shaft sheave. However, it is possible by a change in the method of reeving the drive practically to cure the trouble. As shown, the rope runs from the top of the flywheel to the idler sheave, thence to the tension sheave and then to the top of the line-shaft sheave, when the drive is in operation. The relative location of the sheaves, however, seems to indicate that the drive was designed to operate with the rope running from the bottom of the line-shaft sheave to the tension sheave, thence to the idler sheave and then to the bottom of the flywheel.



EXPERIMENTAL DEMONSTRATION

There is a reason for this last manner of reeving the drive and as a substitute for the mathematical demonstration the experimental demonstration shown in the accompanying sketch is suggested. Note the difference in the reading of the spring balance when raising the weight and when lowering it by power applied to the fall *A*.

A. D. WILLIAMS.

Cleveland, O.

Faulty Reducing Motions

In the issue for June 28, A. C. Waldron in writing of faulty reducing motions shows in Fig. 1 what he calls improper and proper ways of attaching the drum cord to the reducing lever.

A description of the mental process by which he arrives at the conclusion that either of the methods shown can be called proper would make highly interesting reading, for if the diagrams shown in Fig. 2 were got from an engine with one of the reducing levers it would be impossible to get Fig. 3 by using the other.

While the sector if correctly made and attached would lead the cord to the drum in a direction more nearly right than if led from the screw eye as shown on the other lever the sector introduces other errors that are greater than would happen if the cord led from a pin in the center line of the lever and at right angles to it with the crosshead at midstroke.

The diagrams shown are clearly fakes and the conclusions he draws are erroneous, and though no engineer experienced in accurate indicator work will be deceived, some young member of the craft may be led into error by taking the article seriously.

H. M. DARLINGTON.

Midland, O.

Operation of Stop Valves and Cocks

Mr. Row, in the issue of June 28, states that because a blowoff cock is close to the boiler, the weight of water in motion will be small and the water-hammer produced by the sudden closing of the cock is negligible. This would be correct were not the water in the boiler under pressure.

When the blowoff cock of a boiler carrying steam at 150 pounds pressure is opened, the velocity of efflux of the water is approximately 156 feet per second. Even though the weight of water in motion is small, the hammer produced by the sudden stoppage of a flow at this velocity is large and liable to cause accident.

P. LANCASTER.

Monterey, Mexico.

The caption on page 1178 in the June 28 number, "Never Open or Close a Stop Valve Suddenly," gave me quite a start. But the added admonition which the article contained, "Always close a blowoff valve quickly" is enough to make an engineer's hair stand on end.

We had two boilers, each 6 feet in diameter, one 18 feet, the other, 16 feet long. These boilers were set between two partition walls with just 16 inches between. A ladder and the buck stays almost filled this space through which the fireman or engineer must pass to reach the blowoff cocks at the rear where there was another brick wall just 3 feet from the boiler settings. One night I was in a hurry to blow down the boilers, they were very dirty and on this account I did not trust this duty to the fireman. Rushing back of the boilers I opened the blowoff cock on one and closed it with moderate speed; then I opened the blowoff cock on the other, and in my haste to get away, I swung the wrench around and closed it in one swift sweep. Instantly there was a bang inside the combustion chamber. The back connection door came open at my feet nearly blinding me with hot dust. Scalding steam and water got thick around my feet. I was half blinded and bruised my head and shoulders in trying to get out of the narrow passageway.

The boiler carried 105 pounds pressure and the blowoff was made of 2½-inch extra-heavy pipe and an extra-heavy ell. The side had been blown right out of the ell by the sudden jolt caused by closing the blowoff cock so quickly.

I missed my train that night but I learned a lesson which I shall never forget.

In another plant we supplied steam to run an elevator pump in an office building about 250 feet from the boilers. Our steam line was connected to the heating system in this building near the pump by a 4-inch Y. I usually opened the stop valve at the boilers myself. A new fireman who held a second-class engineer's license asked permission to open the valves. Thinking that I could trust him, I consented after showing him the way in which I usually operated them. They were 6-inch gate valves. I would open a valve just enough to hear the steam hiss through; in about 10 minutes, I would open it a little more and so on until after about one hour the valve was half open. It could be opened the other half without waiting.

Somehow I suspected this man of opening the valves before the pipe could be warmed up, the bends expanded and the pockets freed from the water of condensation and I again cautioned him to open the valve slowly and allow plenty of time for the pipes to warm up.

One morning a gasket blew out of a 4-inch line in the building near the pump. Some time later, as the woman who cared for the offices was about to descend the stairs, a 4-inch Y just over the foot of the stairs entering the basement let go. Had she been 10 seconds earlier she could not have escaped suffocation. The fireman had opened the valve quite suddenly. The rush of water ahead of that 90 pounds of steam, together with the vacuum formed by the partial condensation, increased the velocity, and the quantity of water which did not drain out was driven hard against the two valves back of the Y. The Y, being the weakest, split from end to end. It is possible that the valves might have been broken had the volume of water not been divided and reduced.

R. A. CULTRA.

Boston, Mass.

Depreciation

I was much interested in the editorial in POWER for July 5 on depreciation. This is a subject to which I have given a great deal of study in recent years, and concerning which I have collected a vast amount of data.

Depreciation is of importance: first, in appraising the present value of plants; second, in ascertaining the actual or total cost of the product; and third, in fixing proper rates for service, as in public utilities, such as water works, electric-light plants, etc.

Depreciation in general is of three kinds: that due to physical decay; that due to obsolescence, and that due to inadequacy. All of these must be considered in any given case.

I do not, however, agree with the statement that depreciation is more rapid in the earlier years of a plant's life than later. This may be true if merely the market value is considered, but that is not the proper criterion. The unit must be considered with reference to the purpose for which it was installed, and for which it is being used. As a matter of fact, in most cases not only is there no depreciation in the earlier years, but the value may actually increase. This is particularly true of the real estate occupied by the plant. It is often true also of apparatus, due to its advanced cost in the market. More particularly, however, is it due to the fact that the unit has demonstrated its fitness for the service, and is no longer experimental. The owner of a unit which has met all expectations, and is doing entirely satisfactory work, would not trade it for an entirely new unit, within, say, the first five years of its life. Furthermore, the actual wear and tear is usually at a minimum during the earliest years. This fact is recognized by some able engineers, who recommend that there be no depreciation charge for the first years of public-utility plants, such as water works, etc.

There are many different methods of handling the depreciation account. I understand that it is common in machine shops to allow each year 10 per cent. of the preceding year's inventory value. This is objectionable because it makes the depreciation charge largest during the earlier years when, as shown above, it is actually least. Furthermore, it is during these years that many enterprises—such as water works and electric-light plants, etc., which are building up their business—can least afford a heavy depreciation charge. The most serious objection to this plan, however, is that it never entirely wipes out the account.

The plan of charging a fixed percentage every year throughout a definite period is better, but it is also open to the objection that the charge is not consistent with the actual depreciation, and is, furthermore, larger than it need be.

The plan preferred by the best authorities seems to be what is known as the sinking-fund plan. A life period is assumed for the plant, or for each of its units, and an amount is set aside each year, which amount, compounded at a reasonable rate of interest, will, at the end of the period assumed, have grown to a sum sufficient to replace the machine. Under this plan a unit with an assumed life of 20 years would not require 5 per cent. allowance each year, as its value would be repaid in the period named by an annual assignment of 3½ per cent., compounded annually at 3½ per cent. interest. Sometimes this fund is not begun until several years after the starting of the plant, in which event the annual charge must be somewhat larger on account of the remaining life period being shorter.

It is interesting also to note another way of computing depreciation, namely as a percentage of gross earnings instead of plant value. While this plan is common among large electric-light and railway properties, it does not seem to the writer to be entirely logical. It is defended, however, on the ground that the wear and tear on the plant is to a considerable extent proportional to the gross income.

WILLIAM H. BRYAN.

St. Louis, Mo.

The Selection of an Engine

I am glad to have A. V. Clarke take up, in the issue of July 12, the discussion of "The Selection of an Engine" by criticizing my statement in regard to condensation. He must have misunderstood my meaning from the answer he gives, "In the first place." My whole analysis was based on the assumption of approximately an equal cutoff for each engine, and my statement in regard to cylinder condensation, of course, rested on the same assumption. His criticism on that point is, therefore, irrelevant.

He then compares two engines, one a 20x30-inch running at 100 revolutions per minute, the other a 16x30-inch running at 160 revolutions per minute, each developing approximately 200 horsepower; and he states that the larger engine at the lower speed will be the more economical because it has less surface exposed to the steam and therefore less cylinder condensation. I do not agree with Mr. Clarke by any means and my own experience bears me out in what I here set forth.

Assume that both engines are cutting off at one-quarter stroke. The surface exposed to the steam up to this point in the 20x30-inch or low-speed engine is, then,

$330 \times 2 = 660$ square inches (cylinder head and piston)

$\pi \times 20.5 \times 7.5 = 478$ square inches (cylinder barrel)

1138 square inches, total.

For the 16x30-inch or high-speed engine it is,

$207 \times 2 = 414$ square inches (cylinder head and piston)

$\pi \times 16.25 \times 7.5 = 383$ square inches (cylinder barrel)

797 square inches, total.

The low-speed engine has,

$$\frac{1138 - 797}{797} = 0.429 = 42.9$$

per cent. more surface exposed than the high-speed engine. The speed of the two engines will not enter into this calculation, because although the small engine runs at 60 per cent. greater speed than the large engine and therefore exposes the surface a greater number of times; the steam in the large engine is exposed to the surface in its cylinder 60 per cent. longer per revolution than in the small engine.

In making my calculations I am inclined to disregard the polished surface of the cylinder barrel, and consider more particularly the surface of the cylinder heads and pistons which are not as a rule highly polished and which are far more carefully to be considered in connection with their effect on cylinder condensation because of their much greater "regenerative capacity" as I call it. In other words, these rougher surfaces have a much greater capacity for absorbing and radiating heat than have the polished walls of the cylinder barrel. Taking the surface of the cylinder heads and pistons as the more important basis of comparison we find that the low-speed engine has

$$\frac{660 - 414}{414} = 59.5$$

per cent. more exposed surface than the high-speed engine, and the latter is far more economical. My experience, both in automatic-engine work with noted Eastern builders and with the "Higher Speed" Corliss engines of the Wisconsin Engine Company, bears me out in this.

In addition to advantage in having less surface exposed to the range of temperature, the small, high-speed engine has a great advantage in respect to clearance, which is all important on the score of economy. Reduced clearance again means less surface in the ports and passages which will be exposed to the range of temperature. This surface being cored, and therefore comparatively rough, has a still greater "regenerative capacity" and it is very important for economy to reduce it to the very smallest amount.

To go back again to the original article of Mr. Miller, the only real point he made against the 16x36-inch engine running at 150 revolutions per minute was the piston speed, and he rests seemingly secure on this as he gives no reason for his stand. But outside of the fact that for years Corliss engines have been running at piston speeds of 900 t 1000 feet per minute, it would seem tha in these days of the flying machine, wireless telegraph, ocean greyhounds, mar velous machine-tool development an foundry practice, that reputable builder could be taken at their words when they offer to stake their reputations and cas in offering and building such engines The wear and tear at these speeds i properly designed engines, I know fron eight years' experience, is practically ni and the real precaution to be taken b the purchaser is to be sure that the port and passages in the cylinder have ampl area to give the engine the advantage o the full range of pressure and enable i to show a diagram mean effective pressure very close to the theoretical.

CORNELIUS T. MYERS.

Racine, Wis.

Backing Trials of Scout Cruisers

> *These tests, in which the "Chester", "Salem" and "Birmingham" participated, showed that the reciprocating engine provides better backing power than the turbine, does not require the installation of additional units and causes no drop in boiler pressure.*

In the beginning of our experience with turbines attention was naturally centered on their engineering characteristics, such as steam consumption, and details of construction and operation. With the satisfactory development of these elements, we have been able to devote more attention to the tactical features involved, as affecting the ease of handling turbine-engined vessels.

With reciprocating engines a backing power about equal to the ahead power is afforded without any increase in weight except that of the backing eccentrics, rods and links. With turbines, however, backing power requires additional turbines with a considerable increase in weight.

It is necessary therefore to restrict the backing power of turbines to that actually required by tactical considerations. The practice has been to provide a backing power estimated at about 50 per cent. of the ahead power. It has been found, however, in practice that this backing power is rather less than 40 per cent. of the ahead power.

Trials have recently been held with the three scout cruisers, "Chester," "Salem" and "Birmingham," to determine the sufficiency of the backing power provided respectively by Parsons and by Curtis turbines, and as compared with that obtained by reciprocating engines. The criterion has been taken as the distance head reached by the vessel during the interval required to bring her dead in the water from a given speed ahead. A general description of these vessels has been given in a preceding issue.

The trials were conducted with great care and thoroughness, the conditions of displacement, sea, speed, and number of boilers in use, were practically identical for all, and the results may be taken as an authoritative indication of the relative backing powers of the three types of prime mover as now installed.

The speeds selected for the trials were 10, 16, 22 and 24 knots. Three runs were made by each vessel at each of the two lower speeds, two runs at 22 knots, and one run at the high speed. At each speed only the boilers required for a sustained run at this speed were used.

There was no bottling up of steam preparatory to backing, and no speeding up of blowers during the backing interval. It was required that the boiler pressure be not reduced to such an extent as to cause priming.

The results are shown in the following table:

Time to stop from instant of signal for reversal.			Distance head reached in ship lengths		
C.	S.	B.	C	S	B.
M. S.	M. S.	M. S.			
1:52.7	1:20.1	1:10.3	2.2	1.8	1.5
2:06	1:27.3	1:21.2	1.3	3.0	2.6
2:08.5	1:37	1:03	5.4	1.2	3.0
2:51	1:40.5	1:09	6.5	1.7	3.2

It will be noted that at all speeds the reciprocating engine provides better backing power than the Curtis turbine, and that the latter is superior to the Parsons turbine.

The average time required for reversing, and the fall in boiler pressure during the backing interval were as follows:

	Average Time to Reverse in Seconds.	Average Drop in Boiler Pressure, Pounds.
Parsons turbines.......	8.2	82.4
Curtis turbines	39.2	75.8
Reciprocating engines...	5.1	none

In the case of the "Birmingham," the average boiler pressure at the end of the backing interval was 1.5 pounds greater than at the beginning of the interval.

Notes on Marine Condensing Plants

By H. E. King

> *Current practice in the condensing equipment of naval and merchant ships, including rules to determine the amount of cooling water required and the size of the air pump.*

To run a marine engine condensing means a gain of at least 15 per cent. over noncondensing, and if a surface condenser be used instead of a jet it will mean a longer life for the boilers and there will be a saving in fuel. The general arrangement of the condensing plant is fairly well stereotyped, the chief variation being in detail. A usual arrangement of condenser and air pump for merchant service is shown in Fig. 1; the condenser is formed in the back columns of the main engine, and the air pump is worked by a link from the engine crosshead *B*. The air-pump rod is extended through the guide *C* and works on the crosshead *D*. Frequently a reciprocating circulating pump is placed beside the air pump and in many cases these are flanked by the bilge and feed pumps, all worked from crosshead *D*. The arrangement is compact and cheap, but one part cannot be repaired without stopping the whole; it is, therefore, better suited to cargo boats than to warships or fast passenger ships where immunity from breakdown is vital.

On naval and fast passenger ships it is usual to arrange the plant similar to that in Fig. 2. The condenser is shown at *O* connected to the twin air pump which discharges past the air separator *B* and through the grease filter *J* to the feed tank *F*. Any water carried up by the air falls into the funnel at *A*, and passes to the feed tank. The plant shown is for a twin-screw installation and a cross connection *G* regulated by valves *D D* connects the air pumps in each engine room. The circulating-water inlet is at *I* and the discharge at *K*, the centrifugal circulating pump being driven by a small steam engine shown at *P*.

Four-cylinder triple-expansion engines are usual in large warships, a low-pressure cylinder being placed at either end, in which case each has its own condenser; the plant shown in Fig. 2, with the exception of the air pump, being duplicated.

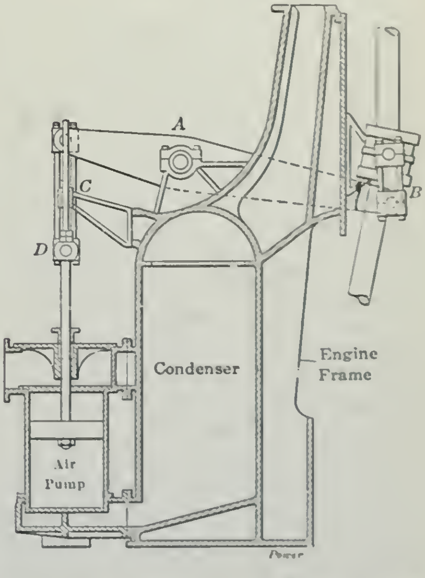


Fig. 1. Condensing Plant Worked from Main Engines

THE CONDENSER

Great care must be exercised in proportioning the condenser, for which the total steam to be condensed, the initial temperature of the circulating water and the required vacuum must be known. On a few naval and merchant vessels a small auxiliary condenser together with air and circulating pumps is provided for the exhausts from the auxiliaries, but in this case, at full power, the "closed exhaust" system is used; that is, the auxiliary exhausts are passed into the low-pressure receiver, thereby throwing the additional load on the main condenser. With the cramped engine rooms of most modern warships, the auxiliary condenser has been crowded out and the main plant is used for all purposes.

The quantity of condensing water required per hour per pound of steam may be obtained from the following:

$$W = \frac{1134 - 7\,T_2}{T_2 - (T_1 + 20)} \times (1.25 \text{ to } 1.50)$$

where,

W = Pounds of condensing water per pound of steam per hour,

T_2 = Temperature of the steam at the required vacuum,

T_1 = Initial temperature of cooling water in degrees Fahrenheit.

The above quantities are based on a cooling surface of 1 square foot per 8.5 to 10 pounds of steam per hour. Naval practice is to keep the cooling surface small with a large amount of circulating water. The condenser may be of any shape to fit the available space, but the usual form is cylindrical.

AIR PUMPS

Air pumps for marine purposes are usually of the head-, foot- and bucket-valve type, although the Edwards pump is much used. When driven from the main-engine crosshead, the length of the large lever, Fig. 1, should be adjusted so that the bucket speed does not exceed 300 to 450 feet per minute. The effective capacity of the pump may be found by either of the following rules:

1. Volume swept through by the pump piston = one-tenth to one-sixteenth of the volume swept through by the low-pressure piston of the main engine.

2. Volume handled by the pump = 7 to 8 cubic feet per pound of steam condensed.

It is advisable to fit an air separator on the discharge, the working of which has already been explained in connection with Fig. 2. Air pumps should not work against much head, and when discharging into a hotwell, if the feed tank is much above this level, a special "hotwell pump" should be fitted.

CIRCULATING PUMPS

As previously pointed out, centrifugal pumps are most suitable for the special needs of warships and fast passenger vessels; they are usually driven by small steam engines, and in this connection some special requirements should be noted. Referring to Fig. 2, it will be see that the pump has a bilge suction at and in case of accident the water can b pumped out through valve *K*. To preven the condenser tubes becoming clogged, th inlet and discharge openings are bypasse by a valve inside the condenser. Obvious ly the work done in clearing the bilges i considerable; therefore, the pump engin must be designed for this duty. Whe there are two sets of condensing pumps it is usual to arrange a cross connectio between them.

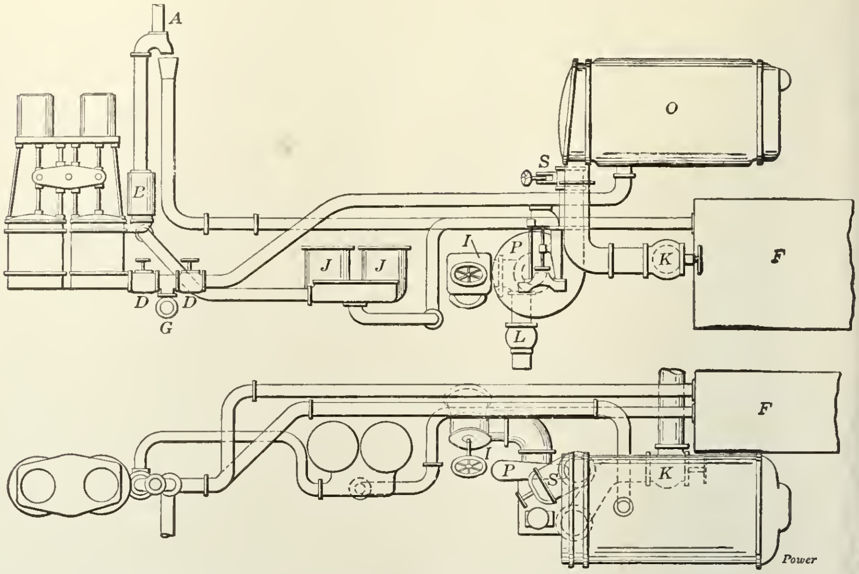


FIG. 2. CONDENSING PLANT SEPARATE FROM MAIN ENGINES

American Peat Society Conventio

The fourth annual meeting of the American Peat Society was held at Ottawa, Ont., July 25, 26 and 27. Representatives of the peat industries in various parts of Canada and the United States were present. Many interesting papers dealing with the manufacture of fuel and fertilizer from peat were contributed.

On the afternoon of July 25 the society visited the peat bog owned and operated by the Canadian government. The bog is located at Alfred, which is about 40 miles east of Ottawa. About 30 tons of air-dried peat are manufactured per day.

On the afternoon of July 26 the government fuel-testing plant in Ottawa was visited. The plant consists of a double-zone peat gas producer, wet-coke scrubber, tar filter and dry scrubber and a four-stroke cycle single-acting Koerting gas engine to which is connected a 50-kilowatt Westinghouse direct-current generator. Peat fuel manufactured at the government bog is used in the producer. Both the fuel-manufacturing plant at Alfred and the testing station at Ottawa will be fully described in a later issue.

At the banquet held on the evening of the second day among those who responded to the toasts were: Hon. Sidney Fisher, Minister of Agriculture in the Canadian Cabinet; Col. J. G. Foster, United States consul; Prof. Charles A. Davis, peat expert, United States Bureau of Mines; Robert Ranson, St. Augustine, Fla., and Dr. J. McWilliam, London, Ont.

The officers for the ensuing year are: President, Dr. Eugene Haanel, Director of Mines, Canada; vice-presidents, Joh N. Hoff, New York City, Eastern States Carl Kleinstueck, Kalamazoo, Mich., Gre Lakes and Mississippi valley; Robe Ranson, St. Augustine, Fla., Souther States; C. V. Imeson, Los Angeles, Ca Pacific States; Dr. J. McWilliam, Londo Ont., Canada; Dr. Charles T. McKenn New York City, New York section; O. Moulton, Dover, N. H., New Engla section, and Max Folty, St. Paul Min Minnesota section; secretary and trea urer, Julius Bordollo, Kingsbridge, Ne York City. The offices were held by t same men last year, all of whom we reëlected by unanimous vote.

Kalamazoo, Mich., was selected as t place at which to hold the next annu meeting. The date of the meeting w be determined later.

Power
and The Engineer

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for July, 1910, *was* 138,000.

August 2	34,000
August 9	34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Editorial

The Purchase of Coal

There are various ways of buying coal in vogue at the present time; one extreme is buying it without any knowledge of its weight or quality and trusting to providence and to the dealer for value received. The other extreme is to buy the fuel on a B.t.u. basis and subject it to a careful analysis and accurate weighing.

For many years the farmers in this country have been protected in the buying of fertilizers by State inspection so that they have a positive guarantee of the contents and value of each bag. The ordinary coal user has no such protection although he stands as much in need of it as the other. Coal dealers are probably neither more nor less honest than other people but there are many slips between the loading of the coal at the mine and the delivery on the premises of the ultimate purchaser. One rather amusing instance of such a slip occurred not many years ago in one of our large cities where coal supplied by a local dealer to an educational institution was found to be short in weight on each wagon load. The employment of detectives who traced the car from the yards to the college revealed a peculiar set of conditions. The coal was carefully weighed at the yards, but on its way to the place of delivery was hauled through a narrow street and a gentleman with a pushcart was seen to come from a neighboring alley, fill up his cart from the load on the wagon and return whence he came. An investigation showed that the coal thus pilfered was sold in a retail store in a poorer quarter by a company in which the drivers were interested, each driver receiving a certain percentage of the profits of the business.

In the purchase of slack coal, the prevailing rains have much to do with the actual amount of fuel that is received by the buyer. To weigh and test coal which is purchased in large quantities is no more reflection on the honesty of the coal dealer than an examination of a check at the bank reveals distrust of the honesty of the one presenting it. It is merely a business precaution which should be taken by anyone purchasing coal in large quantities.

Buying coal on a B.t.u. basis involves the employment of an expert chemist who will make calorimeter determinations of the heat value and ultimate analyses of the fuel itself. Such a chemist should not be in the employ of either seller or buyer but should be a disinterested expert. The expense and trouble attendant on proceedings of this kind will probably prevent their general adoption except in the case of companies buying coal in large quantities. A compromise frequently adopted is the use of a proximate analysis which can be handled by any intelligent engineer and which gives with sufficient accuracy for commercial purposes, the amount of moisture, the volatile matter, the fixed carbon and the ash. The same engineer could probably manipulate also a bomb calorimeter with sufficient accuracy for commercial determinations of the B.t.u. Too much stress, however, cannot be laid on the importance of getting fair samples in such tests. The methods of sampling coal given in the standard boiler code should be carefully followed.

A method simpler than either of the above and perhaps just as efficacious in many instances, consists in weighing the total ash and determining the scale of prices according to the percentage of ash found. If all of the coal used in a given plant is obtained from the same district, there will be little variation in the amount of volatile matter and carbon present. The variation in efficiency will be due to a greater or less percentage of ash.

Car loads of coal from the same mine frequently show a great difference in this respect owing to the particular part of the vein in which each load was mined. An examination of the report of the fuel-testing plant of the United States Geological Survey at St. Louis shows for the coals of a particular locality, a close correspondence between the percentage of ash and the heat value of the coal. Plotting points on these two quantities as coördinates gives approximately a straight line; that is, the B.t.u. in the coal will be inversely proportional to the percentage of ash. It will then only be necessary for the purchaser to weigh his coal, to make an ordinary moisture determination if he pleases so as to find if the coal is unreasonably wet, and finally to keep a record of the amount of ash from each carload. A sliding scale of prices could be arranged on this basis, a certain price being agreed upon for a sample load containing a certain proportion of ash, and then a certain number of cents per ton being added or subtracted for each 5 or 10 per cent. variaation in the ash content. This method of testing coal involves no skilled labor and no unnecessary work and as long as the coal comes from the same locality, will give an index of its calorific value which will probably be accurate enough for ordinary transactions. It will certainly be a great improvement over the usual method of buying coal without any knowledge or standard of its value.

Thermal Storage

On another page of this issue we print an excerpt from our contemporary, *The Engineer,* upon the subject of "Thermal Storage." Under this particular guise the system, of storing heat during periods of light load to be utilized during the peak load, seems to have been used extensively in England, although we are not aware of it having been used on this side of the Atlantic.

In principle the system is not new; it is analogous to the practice, followed in many well managed plants, of filling the boilers to their limit a few hours before the peak load comes on, with the result that when the peak comes on, the boiler is full of hot water at the boiling temperature ready to be transformed into steam. In this way the capacity of a boiler may be more than doubled for a short time. It is much better practice than that of cutting in additional boilers with the resulting losses.

With the English system, as described, the tests show an hourly evaporation with the thermal storage of nearly double that obtained without it. It should be borne in mind that this refers to the total evaporation per hour, not the economic evaporation, or the evaporation per pound of coal. Furthermore, the duration of the test was only 65 minutes. With a feed-water heater using exhaust steam the heat given up to the feed water represents actual gain, providing the conditions permit the use of such a heater; but with a thermal-storage tank the feed-water temperature is raised at the expense of heat units which would otherwise go into the steam. This amount of heat, however, is undoubtedly less than that which would be required to heat the water to the boiling point if the boiler were fed with comparatively cold water and forced in order to carry the peak load. While the economy is by no means increased in proportion to the increase in evaporation, still we believe there is considerable economy in the particular function which this system performs. Moreover, it permits the use of a smaller boiler than would otherwise be able to do the work, thereby decreasing the first cost of the equipment required to carry a given load, which, of course, still further enhances the economy obtained by reducing the interest charges on the investment in plant equipment.

Safety Valves

The discussion of safety valves may be made to hang on the two questions: What is a safety valve and why is a safety valve? Taking the second one first, a safety valve is put on a steam boiler because the inspector will not pass it without, and this leads to the answer of the first question, that a safety valve is something put on a boiler so that the inspector will pass it.

If this discussion should be thought too trivial even for the hot wave, when we think of steam to avoid it, it may be remembered that in the navy it is not considered good form to allow a safety valve to open or even to leak. And, seriously, a safety valve should never open except when opened to see that it is in working condition. Under some circumstances it would be proper to discharge a fireman who should allow the safety valve to blow. There appears to be only one condition under which it is good engineering practice to allow the safety valve to blow (a little, occasionally) and that is when it is necessary to give ocular proof that there is steam enough and that a failure to make schedule time is not chargeable to the boiler.

A theory of the safety valve assumes that it should be able to discharge all the steam that the boiler can make when forced, without allowing the pressure to run more than a small amount above the working pressure. The basis of this calculation for a safety valve on this theory is the well known Rankine rule for the flow of steam in pounds per second,

$$Weight = A\frac{p}{70}$$

where A is the effective area in square inches and p is the absolute pressure; that is, the gage pressure plus the atmospheric pressure. This rule will give a result 2 or 3 per cent. too large; but that is trivial in the face of the difficulty of estimating the power of the boiler to make steam when forced. The details and difficulties of that estimate and of determining the effective area of discharge may be left to a more convenient season—next January, perhaps.

It appears that some valve makers have found it convenient to insist that one way of working a safety-valve discharge more steam is to open it wider. It is evident that if a safety valve opens twice as wide it will discharge twice as much steam; or what is more to the point a valve having half the area will discharge as much steam. If only the inspector can be induced to see it that way the expense of complying with his demands will be greatly reduced.

But there has never been any trouble about making a safety valve open. Even the old lever safety valve opened promptly at the right pressure and lifted high enough to discharge all the steam the boiler could make. But when it came to closing, the lever valve was altogether too deliberate; it closed slowly, and only after a considerable reduction of the steam pressure, and paid no attention to the matter of good form, and a leaky safety valve is a nuisance anyway.

From this cause came the generation of the pop-safety valve that at once opens wide, and after a very moderate reduction of pressure shuts quite as suddenly. Now the way to stop leakage is to make the seat narrow so that the pressure per square inch is sufficient and though the valve and seat may be made of har alloy, the combination of narrow surfac and sudden closing is apt to approach th limit.

What then if the valve be made t open twice as wide? Well if the inspector does not object and if the fireman never allows the pressure to reac the popping point, nothing happens Doubtless careful training may enabl him to quietly lift the valve to see tha it is not stuck.

It is rumored that certain interests hav tried out the safety-valve question wit the assistance of a competent professional expert. Rumor never quotes technical details, but we may confidently predict that they will report with mor exactness and greater authority what i here stated concerning the discharge fro safety valves. We hope they will als prove as completely what happens to th seat of the valve when the lift is double Perhaps they will find how to do tha too, without eventually leading to th bad form of leaky safety valves.

Lest some reader may conclude tha this paper approves the abolition c safety valves, it may be wise to say tha there are occasions to be guarded agains The fireman *may* let the pressure ris too high, in which case the blowing c the safety valve should precede his discharge. Something may happen to th engine suddenly and then the steam mu be disposed of for a while. But if thes occasions were the whole story we cou afford to let the valve rise twice as high— and pay for a new seat, if necessary.

In deciding upon the size of the stea piping to be used, keep in mind two ve important factors. If the pipe be ma too large, not only will the first cost l unnecessarily increased, but the radiati will also be greatly increased; on t other hand, if the pipe be made too sma the friction will be excessive.

If you want anything done, *hire it don* If you want it done right, *do it yoursel*

The successful engineer is nev "stumped" for the lack of proper m terial or fittings; he can always fi something of use in the scrap pile.

Inquiries of General Interest

Effects of Low Water

What part of tubular boiler would be first affected in case of low water?

W. R. O.

The back head below the rear arch and as many tubes as were uncovered.

Foaming Boiler

What should be done in a case of boiler foaming?

H. A. T.

The flow of steam should be checked by closing the engine stop valve until the water settles, when, by blowing and feeding, enough of the impurities may be removed to permit the safe operation of the boiler.

Heating Surface

How do you find the heating surface of a boiler 14 feet long and a given number of tubes?

E. M. N.

Take the area of the lower half of the shell, the inside area of all the tubes and the area of one head minus twice the area of the ends of the tubes. The sum of these areas will be the heating surface of the boiler, which reduced to square feet and divided by 10 will give the horsepower.

Connecting a Boiler Under Pressure

How would you connect a boiler to a battery of two or more boilers when raising steam?

E. A. D.

When the pressure in the boiler which is to be cut into the line reaches that on the others, the connecting valve should be slowly opened. If the pressures are the same and the pipe is free from water no harm can come from opening the valve quickly, but the possible inaccuracy of the gages and the presence of water on one side or the other of the valve make caution imperative.

Factor of Evaporation

What is meant by the factor of evaporation, and how is it found?

S. O. R.

The factor of evaporation is the number by which the actual evaporation is multiplied to reduce it to the equivalent of the evaporation of water at the temperature of 212 degrees into steam at atmospheric pressure. It is found by subtracting the total heat above 32 degrees in the feed water from the total heat above 32 degrees in the steam at the observed pressure and dividing the remainder by 965.7, the number of heat units required to evaporate one pound of water at the temperature of 212 degrees into steam of the same temperature.

For example, suppose that water entering the boiler at 141 degrees is evaporated into steam at 85 pounds gage pressure. Each pound of water contains 101.9 heat units and each pound of steam 1181.8 heat units; subtracting 101.9 from 1181.8 leaves 1071.9, which divided by 965.7 gives 1.1, which is the number by which the actual evaporation is to be multiplied to reduce it to the equivalent of evaporation from and at 212 degrees.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Steeple and "Fore and Aft" Compound Engines

What is the difference between a steeple compound engine and a fore and aft compound engine?

A. W. F.

Steeple and fore and aft as applied to engines are marine terms meaning tandem and cross compound respectively.

Position of Throttle Valve

How should the steam-throttle valve be placed on an engine?

A. R. M.

It should be so placed that the steam pressure will be against the valve, tending to hold it open.

Object of a Condenser

Why is a condenser placed on the exhaust pipe of an engine?

G. A. O.

For the purpose of condensing the steam on the advancing side of the piston and thus removing the resistance necessary to force the steam out of the cylinder against the pressure of the atmosphere.

Compression

How is the amount of compression in an engine influenced by the speed?

F. R. H.

Not at all. Compression is used in engines for the twofold purpose of filling the clearance with steam and furnishing a cushion to absorb the momentum of the moving parts and bring them to rest with all of the lost motion taken up so that the centers will be passed without shock. In high-speed engines compression is usually carried much higher than in the so called slow- and medium-speed types, but it is not necessary as the piston speed is usually no higher.

Horizontal-tubular Boiler Seams

What is the better seam to use in a tubular boiler, single, double or triple riveted lap or butt and strap joint?

D. O. A.

It depends upon the purpose for which the boiler is to be used. If for low pressure, steam or hot-water heating, the double-riveted lap seam is better, because it is cheaper and is safe because the range of pressure change is not wide enough to produce what is called breathing. For any higher pressure only the butt and double-strap seam should be used.

Brake Horsepower

A prony brake has an arm 6 feet long and with the engine running at 200 revolutions per minute presses on the scale with a weight of 14 pounds. What horsepower is being developed?

L. J. K.

The brake horsepower of the engine is determined by the formula

$$H.P. = \frac{2\,\pi\,L\,N\,P}{33{,}000}$$

in which

$\pi = 3.1416$,

$N =$ Number of revolutions per minute,

$L =$ Length of lever arm in feet.

$F =$ Weight in pounds acting on scale at a distance L.

In the example given

$2\pi = 6.283$

$L = 6$

$N = 200$

$P = 14$

Substituting

$$\frac{37.69 \times 200 \times 14}{33{,}000} = 3.2$$

the horsepower developed.

The Magneto and the Electric Bell

Why does an ordinary electric bell fail to ring when connected to a telephone magneto?

G. K.

Because the bell will respond only to direct current and the magneto delivers alternating current.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Simmance - Abady Precision CO_2Mbustion Recorder

A CO_2 recorder made in London, England, by Alexander Wright & Co., Ltd., is now being made in this country by their American branch, the Precision Instrument Company, 49 Larned street, West, Detroit, Mich.

For the benefit of those who are not familiar with this type of apparatus, it might be well to state that this CO_2 recorder is an instrument which automatically, and at three-minute intervals, takes samples of flue gases from one or more boilers, analyzes them for the percentage of carbon dioxide which they contain and records graphically on a chart this percentage of carbon dioxide.

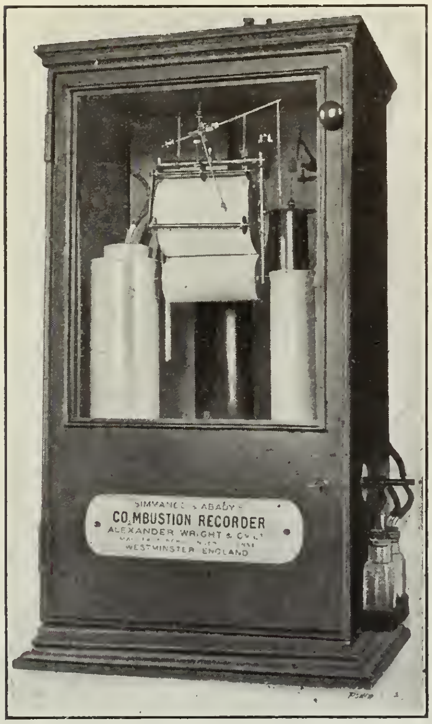


FIG. 1. THE SIMMANCE-ABADY RECORDER

The value of this knowledge to the owner, engineer or fireman, is that he can tell at a glance how complete—or, generally incomplete—the combustion of coal is on his grates, and, with this knowledge, remedy bad conditions of firing, draft, etc. It is, of course, well known, that carbon dioxide is formed by the combination of carbon with air in complete combustion; incomplete combustion produces carbon monoxide, with a loss of two-thirds of the gross available heat. Good combustion should show a certain percentage of carbon dioxide.

The recorder is a simple instrument in a case made of steel. It can, therefore, be stood in any hot or dusty or drafty boiler room, where the firemen can see it working, and is entirely unaffected by outside conditions. It is permanently made, mechanically contrived, and accurate, with little to get out of order and no glass vessels to break.

It works on a boiler or boilers with either forced, induced or natural draft; there is no "lag" in its records, and it is made so as to draw gas continuously without stopping from one year's end to another.

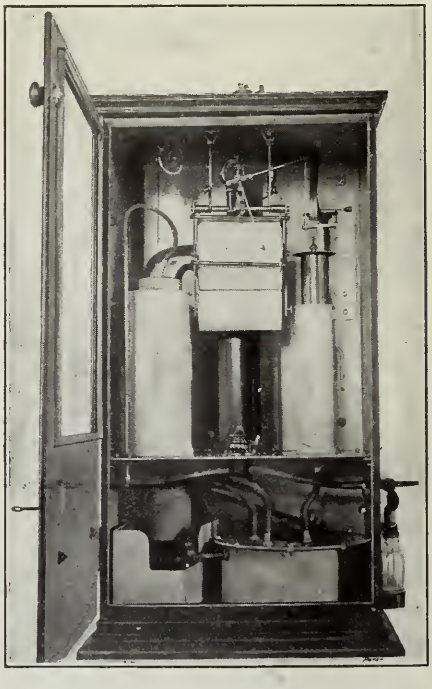


FIG. 2. RECORDER IN CASE WITH DOOR OPEN

It is made either to record the percentage of CO_2 or both the percentage of CO_2 and the flue draft on the same chart. i.e., the conditions producing the percentage. One of the half-dozen types is illustrated in Figs. 1, 2 and 3. Fig. 4 represents a reduced facsimile showing what a record of draft and CO_2 looks like. The chart for May 6 shows 6 per cent. CO_2, that for July 13, 12 per cent. or, according to the chart shown in Fig. 5, a saving of 15 per cent. in fuel.

A small stream of water is the motive power for effecting the whole of the operation which is described in the following:

Water is allowed to drop into the siphon tank *A* from a small reservoir *B* which is fitted with a safety overflow. In the tank *A* there is a float *C* which is attached by means of a chain to the bell *D* of the extractor, and this float rises and allows the bell of the extractor to fall. Assuming that the bell of the extractor is empty, then the float *C* in the tank *A* would be near the top, and when it reaches the top it engages with the drip valve *E* on the water service *F*, quickening the supply for a moment and starting an automatic siphon *G*, and also actuating the pen. The water running from the siphon actuates the balance valve *H* which opens the way from the flue to the extractor, and vents the recorder *J*. Naturally as the water siphons out of the tank *A* the float *C* inside drops, and being heavily weighted, pulls up the bell *D* of the extractor, which is suspended in a vessel filled with water, and owing to the partial vacuum created in the bell a quantity of gas from the flue is drawn into the bell.

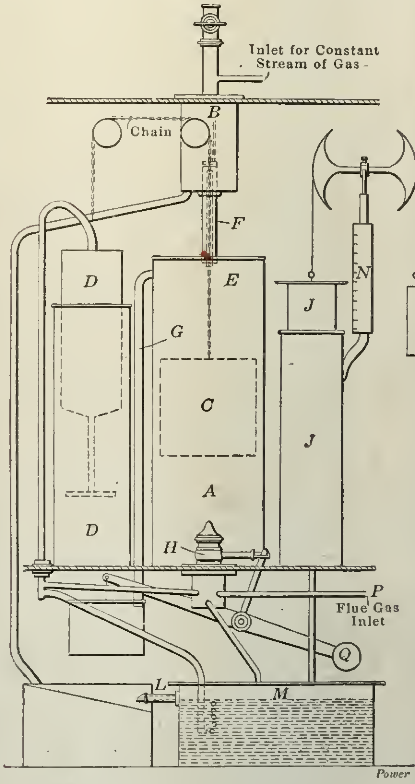


FIG 3. DIAGRAMMATIC SKETCH OF RECORDER

The siphoning being finished, the passage from the flue is automatically shut off by the balance valve *H*. The water then runs into the tank *A* lifting the float *C* and allowing the extracter bell *D* to fall. As it does so it will be seen that the contents of the bell *D*, which by the closing of the valve *H* are now uninfluenced by vacuum or conditions in the flue, are first reduced to atmospheric pressure, and then are actually under pressure, and the volume of gas is, therefore, transferred, by bubbling through the separate iron vessel *M* containing a caustic potash solution, to the recorder *J*.

The capacity of the bell *D* is such that the volume from 100 to 0 on the engraved indicating scale *N*, is transferred when the apparatus is being set by air containing practically no CO_2. When the flue gas has been turned on to the apparatus exactly the same quantity (i.e., enough to send the recorder bell up from 100 to 0) is passed from the extractor bell *D*, but on its passage the CO_2 is absorbed by caustic potash in the iron vessel *M* and, owing to such absorption, the recorder bell *J* will not rise to its full hight. It is allowed to automatically rise as far as it will, and a pen (not shown in the section) then marks on the chart its final position.

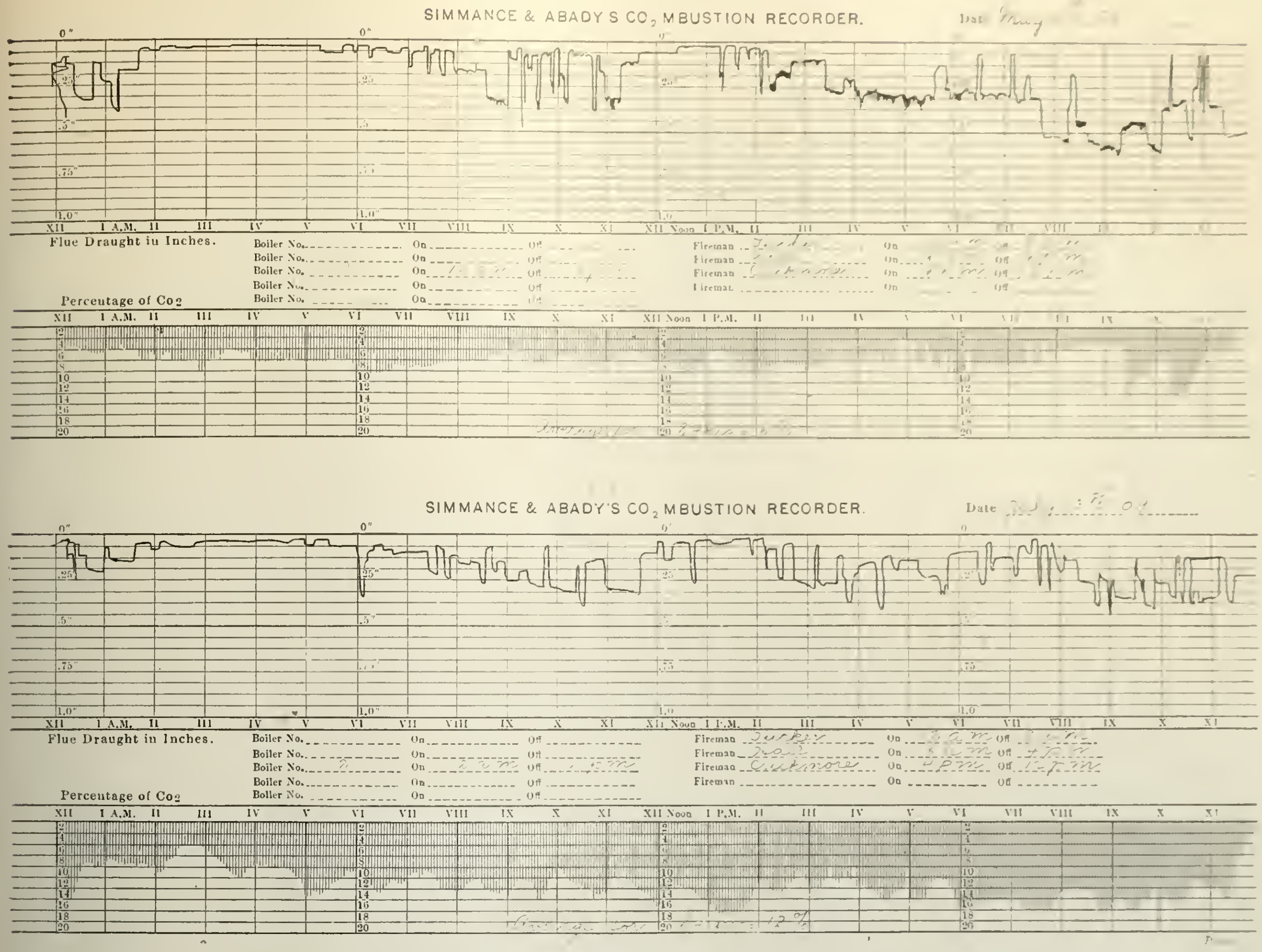


Fig. 4. CO_2 and Draft Records

The percentage of CO_2 in the gas is thus automatically recorded. This bell *J* then vents, sending out the analyzed gas by a separate channel and without mixing or coming in contact with the fresh charge of gas which is brought forward to be dealt with in exactly the same way, the whole of the operation, as well as the continuous drawing forward of the flue gas, taking place automatically by means of the dripping water.

An injector is fixed below to bring along a constant supply of gas. By its means gas is continuously exhausted from the pipes connecting the recorder to the boilers, so that the successive samples analyzed by the instrument are from the boiler flue and not stagnant gas in the connecting pipes. The injector is worked by the small stream of water, the motive power for the recorder, connected at *X* before this enters the top tank of recorder so that no extra water is used for this continuous pump. Two glass bottles, Fig. 1, are fixed in connection with the injector as safeguards. A glance at one shows whether the flue gas pipes are clear of obstructions and the other shows that the stream of water is being maintained.

Attention is called to the simple pen movement, balanced on gilded point bearings, and with adjustments in all directions.

The working of the recorder can be seen all the time, so that although the instrument is in a locked case, yet the stokers can see what effect they are producing and regulate their work accordingly.

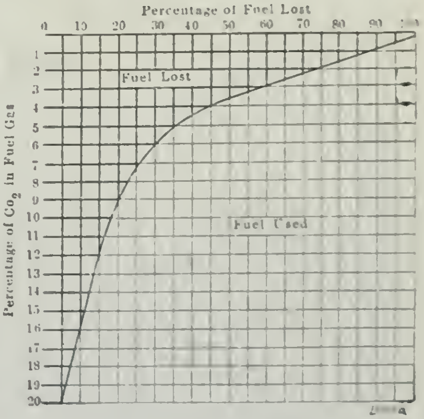


Fig. 5. Fuel Chart

Its accuracy can be checked at any moment by opening a tap and drawing in air instead of flue gas, when the apparatus will indicate zero, or no percentage of CO_2. There is no troublesome adjustment, no continuous changing of chemical solution. No glycerin, oil, or other similar liquids; but the apparatus is always accurate and unaffected by temperature, evaporation and all the dusty conditions of a boiler-house, and it contains no perishable parts.

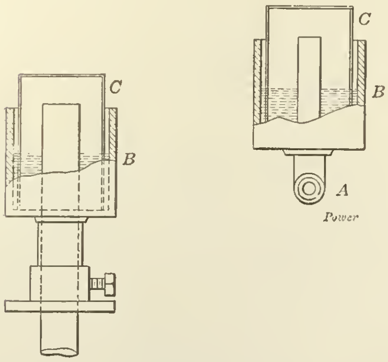


Fig. 6. Oil Cup

As the recorder continually draws forward hot and dusty gases, the pipes would become stopped up with dust and condensation if some suitable precaution were not taken to prevent it. Therefore, a system of filtering and condensing has been devised which prevents all possibility of pipe stoppage, as shown in Figs. 6 and 7.

A bus pipe or header *A*, Figs. 6 and 7, is run along the range of boilers, and opposite each boiler a tee is placed. On each of these tees is screwed a small iron oil cup *B* and a cap *C* is dropped over the inner tube, shown in Fig. 6, and a little lubricating oil is poured into the cup, thus sealing it. Into each flue is inserted a length of iron pipe (horizontally if a water-tube boiler), and in the end of each of these flue pipes is also screwed an oil cup *B*, and cap *C*, sealed with oil. This furnishes a means of making connection from any boiler to the bus pipe. A filter, Fig. 7, filled with wood wool, glass wool, cotton waste, pumice stone, as most convenient, is placed in the oil cup *B* of the boiler it is desired to test; after having removed the cap *C*, the channel of the filter is also filled with oil. A piece of muslin is placed over the wood wool or other filtering material in the filter as shown.

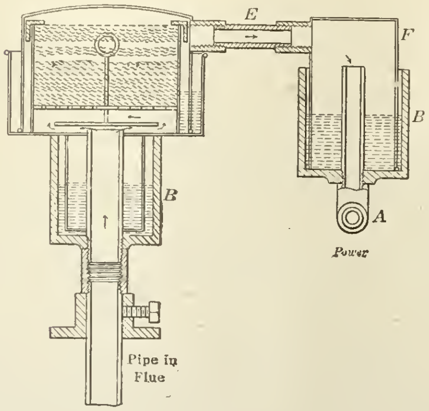


Fig. 7. Filter

Then the filter cover is dropped on and is sealed by the oil in the filter channel. Attached to the cover is a short length of pipe *E*, with elbow and cover *F*, arranged to reach the oil cup *B* opposite on the bus pipe *A*, and it is inserted in this oil cup after the cap is removed. Thus, as Fig. 3 shows, a perfect connection for the flue gas is made which can be removed in a moment to clean the filter, or can be moved on, filter and all, to the next boiler when it is desired to change over. The bus pipe must be run quite parallel to the boiler pipes if this is possible, but if not, a short length of metal-armored hose can be used to connect the filter cover with the bus pipe cover. It will be seen that one set of filter fittings will do for a whole range of boilers, that there is little restriction as to how far off the bus pipe may be from the boilers, that a set of two oil cups and caps is required for each boiler, and that the recorder can be changed from one boiler to another in a minute. There are no cocks or valves to get stopped up or become leaky.

The only attention necessary with this recorder is to put ink in the pen and tear off a chart once a day, wind the clock once a week, renew the solution once a month and put on a fresh roll of charts once each two months.

Canadian Engineers Hold Convention

The twenty-first annual convention o the Canadian Association of Stationar Engineers was held in Berlin, Ontario July 26, 27 and 28. The business session of the convention were held in Victori hall, and in a large room in the sam building, which was specially decorate and fitted up for the purpose, was hel the mechanical display under the super vision of the Canadian Exhibitors Assc ciation.

At eleven o'clock on Tuesday mornin July 26, the preliminary exercises too place. The meeting was called to ord by Executive President Charles Kell who closed his brief speech by introdu ing E. C. Fitzgerald, chairman of th local committee, who hoped that eve

Delegates and Visitors at Berlin, Ontario delegate would help toward making the convention a success. Mr. Fitzgerald then introduced Mayor C. C. Hahn, who welcomed the delegates and visitors to Berlin. W. A. Crockett, secretary, and A. M. Wickens, treasurer, responded in behalf of the engineers.

After the president had appointed the necessary committees an adjournment was taken until two o'clock in the afternoon.

At the several sessions which followed the business of the convention was disposed of with harmony and despatch.

The reports showed a steady and healthy growth, and the organization is said to be in a prosperous condition.

After some discussion many important changes were made in the constitution, the object being to have one central society in which all the provincial associations will be affiliated.

At the closing meeting on Thursday afternoon the following executive officers were elected: Charles Kelly, past president, of Chatham; J. J. Heeg, president, of Guelph; William Norris, vice-president, of London; A. M. Wickens, treasurer, of Toronto; W. A. Crockell, secretary, of Mt. Hamilton; J. A. Robertson, conductor, of Stratford; H. R. Clark, doorkeeper, of Hamilton. It was decided to hold the next annual meeting at Stratford, during the last week in July.

There were many enjoyable entertainment features. On Tuesday evening a banquet was given at the Walker house, the headquarters of the convention. When the coffee had been served, Mayor Hahn, who acted as toastmaster, introduced Dr. H. G. Lackner, member of the provincial legislature. Mr. Lackner gave a short address, which was followed by a few remarks from Executive President Charles Kelly. The company then adjourned to the Star theater, where in addition to the regular vaudeville performance Charles N. Pische, of Jenkins Brothers, and Jack Armour, of POWER, entertained.

On Wednesday morning there was an automobile ride through the city for the ladies. In the afternoon a visit was made to beautiful Victoria park, where outdoor games of all kinds were indulged in, the ladies taking a prominent part in the sports. Suitable prizes were awarded the winners of each event. Refreshments were served at the close of the festivities.

On Wednesday evening a smoker was held at Victoria hall, at which three very interesting papers were read: "Steam as a Motive Power," by A. M. Wickens; "Considerations Affecting the Choice of an Engine," by A. W. Spotten; "The Saving of Heat Units," by H. L. Peiler. The smoker wound up with an entertainment furnished by the supply-men.

The following firms comprised the exhibitors: Bundy Department, American Radiator Company, Bice Regulator Company, Canadian Manufacturers' Publishing Company, Canadian Fairbanks Company, Canadian Steam Boiler and Equipment Company, Dart Union Company, Dearborn Drug and Chemical Works, Francis Duffy, Garlock Packing Company, Goldie and McCulloch, I. G. Shantz & Co., Jenkins Brothers, Lunkenheimer Company, *Power House*, Piler and MacKenzie, Quaker City Rubber Company, Twin City Oil Company, The Strong, Carlisle and Hammond Company, POWER AND THE ENGINEER.

At a meeting of the Canadian Exhibitors Association held Thursday afternoon the following officers were elected: J. B. Goff, president; E. F. Hetherington, first vice-president; H. L. Peiler, second vice-president; Gordon E. Keith, secretary; D. O. McKinnon, assistant secretary; George O. Fischer, treasurer; W. R. Stanert, chairman of exhibit committee.

Meeting of the Ohio Electric Light Association

The sixteenth annual meeting of this growing and important organization took place July 26, 27 and 28 at "The Breakers" hotel, Cedar Point, O., the charming resort on Lake Erie. E. H. Beil, of Youngstown, vice-president, presided at the opening exercises, at which the convention was heartily welcomed by E. A. Bechstein, of Sandusky, chairman of the committee on general arrangements. Of the papers presented at the meeting one on "Low-pressure Turbines and their Operations," by W. C. Anderson, of Canton, and "Turbine Troubles," by Frank Brosius, of Columbus, were perhaps most interesting to readers of POWER. Mr. Anderson's paper ran in part as follows:

It will be agreed to by all that there is room for improvement in the coal economy of the average electric lighting plant. A coal consumption of 1½ pounds per indicated horsepower-hour, including all auxiliaries, is not unusual in steamship practice. This would be equivalent to 2.6 to 2.75 pounds of coal per kilowatt-hour, making due allowance for the grade of coal usually consumed by electric lighting plants in Ohio. The average coal consumption of the small high-speed lighting plant is about 10 pounds per kilowatt-hour, or nearly four times the amount of coal required to develop an equal amount of power aboardship. The business of supplying electric light is rapidly changing. The cheapness and perfection to which the transmission of power has been developed create an opportunity for large distributing systems, taking advantage of the economies of large units distributing current over a wide area. The small central-station plant with its inefficient means of producing electric power is, therefore, likely to be forced by public sentiment to meet rates prevailing in another territory, although such rates may be below the cost of production under its unfavorable circumstances. We must, therefore, consider in discussing how to improve the coal economy of our plants everything which bears on the situation. To discard the investment already made in generating apparatus for the adoption of more economical equipment of a later type is difficult to justify, even in view of the reduction in operating expenses likely to be realized. We are, therefore, exceptionally interested in any improvement in current-generating equipment that will harmonize with investments already made.

The low-pressure turbine is a piece of apparatus of comparatively late development that fully conforms to this requirement. The steam turbine is an apparatus that has gained a reputation for economy by using economically energy that it is impossible for the steam engine to extract from steam. Over higher ranges of pressure, i.e., from say 200 pounds to atmospheric pressure, the steam turbine is ordinarily less economical than the steam engine. It is only at pressures below the atmosphere that the steam turbine gets its greatest reputation for economy. The most economical unit for the production of power by steam is a combination of high-pressure engine and low-pressure turbine. In the plant already operating its engines condensing, any increase in capacity may be met by installing a steam turbine, either a straight high-pressure, a straight low-pressure or a mixed-pressure unit. The desirability of either one of these three units is an engineering question regarding which no general rule can be laid down. Each problem must be worked out according to the condition prevailing.

For the noncondensing plant many other problems must be met and solved. Speaking generally (from the standpoint of central-station service), the noncondensing plant exhausting into the atmosphere is likely to have a short life. The difference in economy between this kind of a plant and the plant of the average small manufacturer is too small to make profitable the sale of electric power to customers using any considerable quantity of power. The margin between the cost of power to the central station so equipped and a plant on the premises is too narrow to pay the expenses of distributing the current. A noncondensing plant therefore must, to survive, either find a market for its exhaust steam or operate condensing. If condensing water is available the low-pressure turbine offers an almost perfect solution.

Continuing, the paper brought out the flexibility of such a system in the plant of the Canton Electric Company. The installation was primarily a noncondensing-engine plant and at a later date a standard high-pressure 500-kilowatt turbine was installed, using condenser water from a well on the premises. In addition to operating in part condensing, the plant also includes an exhaust-steam heating system taking steam from the compound noncondensing engines during the winter months. The atmospheric connection to the turbine is, further, connected to the exhaust line, the steam used in this way being expanded in the lower stages of the machine.

During certain hours of the day and depending on weather conditions, there is either a surplus of exhaust steam or a shortage, and to regulate the heating pressure and take advantage of every pound of steam generated by the boilers, the turbine is floated on the line all the time, taking all the exhaust steam from the reciprocating engines that is not required for supplying the heating system or heating the feed water, thus utilizing, during a greater part of the time, steam that otherwise would be wasted, and increasing the output without additional expense at the coal pile. From midnight till morning in very cold weather or during such other hours that the load is lightest and the supply of exhaust least, the turbine is operated noncondensing in whole or in part. The regulation is altogether by electrical load, the turbine and engines feeding into the same system. This flexible arrangement is stated as highly satisfactory and insures excellent economy at all times.

Mr. Brosius' paper dealt with the minor troubles with which the operating engineer has to contend in turbine operation. In reference to the step bearing the paper stated that in the early days of the vertical machine the probability of the loss of the step was considered a serious thing, but now is not considered of more consequence than hot cranks on reciprocating engines of the high-speed type. A reserve-pump system was advocated, such as is in use at the Columbus Railway and Light Company. Here a Worthington hydraulic steam pump is so connected up as immediately to boost the pressure on the steps in case of trouble with the motor-driven pumps. A Williams pump regulator is in the line to the steps, set about five pounds lower than the pressure regularly carried on the step bearings. In case either motor-driven oil pump should drop the pressure the steam pump immediately picks it up. This system has operated perfectly since installation and is much cheaper than the accumulator system, though it requires more attention to keep in running order.

The effect of superheated steam on pipes and fittings was also discussed in this paper, the conclusion being that it is necessary to use steel castings to be on the safe side. An illustration of the effect of superheated steam on different grades of cast iron was shown by the records of two Corliss engines in the plant of the Columbus Railway and Light Company. These engines were required to use superheated steam which had been put in to operate the turbines later installed. Exactly the same attention was given to each engine, the amount of superheat and lubrication being the same in each case. No. 2 engine cylinders were entirely uninjured by the superheat, while those of No. 3 were badly worn. The deduction is drawn that superheated steam had nothing to do with the wearing in No. 3 cylinder, but that the quality of the iron was decidedly different than that in the other engine, this emphasizing the importance of high quality in any installation where superheat is to be used.

A paper entitled "Motors for Single-phase Circuits as They Are Today," by Prof. F. C. Caldwell, of Ohio State University, contained much condensed information on this type of motor, of which it said in part:

There are six more or less distinct types of motors which can be operated on single-phase current. They may be termed shunt, series, simple induction, split-phase induction, simple repulsion and repulsion induction. The shunt single-phase is easily disposed of. Since a direct-current motor will run forward no matter in which direction the current is supplied, it would continue to run if the current were rapidly reversed or if an alternating current were substituted. Differing reactance of the armature and field, however, introduce difficulties in practical operation and it seems unlikely that any extended development of this type will occur.

In the series motor, the armature and field being in series, there can be no difference in the phase of the currents. The fact that this is distinctly a railway motor and that its application on central-station power circuits is limited would not justify here an extended discussion of its characteristics. Suffice it to say that it has practically the same qualities as the series direct-current motor, though usually slightly less desirable in point of efficiency, weight and cost.

Next come the various induction types, which in the form of constant-speed stationary motors are of the greatest interest to central-station operators. In the simple single-phase induction motor we have the revolving field of the two-phase motor, but here its production is more difficult to understand. The alternating field produced by the magnetizing coils must combine with another field at right angles to it to give the necessary revolving field, but how is the other field produced? Just as in the direct-current machine we have the cross ampere turns and cross flux of the armature, so in the single-phase motor the currents set up in the rotor by the movement of its conductors through the alternating field set up a cross field, also alternating, and at right angles to the main field. We thus have the two alternating fields at right angles to each other, which are needed to produce the rotating field; but it remains to be shown that they are 90 degrees apart in phase as well as in space. This is easily understood if we remember that the rotor is an induction circuit of very low resistance and that consequently the current produced by an alternating electromotive force applied to it must lag nearly 90 degrees behind the voltage. Thus it follows that the magnetic field will be in phase with the current, will be 90 degrees behind the electromotive force in phase, and consequently 90 degrees behind the main field. We have thus the two requisites necessary for a revolving field and thus the single-phase induction motor becomes as simple in its operation as the polyphase.

The repulsion type has next to be considered. This has a simple one-circuit alternating field, but the rotor is wound like a direct-current machine and has a commutator. The brushes, however, are short circuited by a copper bar. To understand this type of machine consider a two-pole machine with the poles horizontal and the brushes set opposite each other under the middle of the poles. Then the rotor winding will act like the secondary winding of a transformer and the electromotive force set up in it will be in opposite directions at the top and bottom of the rotor. Thus a current will flow through the brushes just as in a direct-current machine. However, since the current will be in opposite directions on each side of a brush, that is, under each half of a pole face, the pull under the pole will be half in one way and half in the other and there will thus be no motion.

On the other hand, if we put the brushes vertical or midway between the poles, we shall have no current, since then the electromotive force will be in opposite directions in the two halves of the winding, and there will be no current and hence no motion. If, however, the brushes are set somewhat off from this position, the balance of voltage will be upset and current will begin to flow. Further, this current flowing under the poles will react with the field and a pull will result. Here it is to be noticed that this will occur regardless of whether the rotor is moving or stationary; hence such a motor will start itself and pull a load. This type has in general the characteristics of a direct-current series motor and in its simple form is generally unsuited to stationary requirements, though it is used by some makers as a variable-speed motor. The principle, however, is very useful in starting in connection with motors that operate as simple induction motors after attaining speed, this being accomplished by using a mechanical device with centrifugal-governor weights arranged so as to short circuit the commutator and usually lift the brushes after a predetermined speed is reached.

There remains a special type of repulsion motor or repulsion-induction motor. In this motor, besides the two short-circuited brushes of the ordinary repulsion type, there are two others, also spaced 180 electrical degrees apart and 90 degrees from the first set. The circuit by which these brushes are connected together, and which is called the compensating circuit, passes through a portion of each of the main field coils, so that a part of the voltage applied to the machine is impressed upon this part of the rotor circuit. Thus two circuits through the armature are established, the energy circuit through the short-circuited brushes and the compensating circuit through the other. By properly designing these coils, characteristics are obtained which give a very satisfactory motor. Further, by use of resistances in the compensating circuit an adjustable-speed motor is obtained, having the characteristics of a shunt direct-current motor regulated by resistances in the armature circuit. A notable characteristic of this motor is the high power factor of 96 to 99½ per cent. Thus the current required for a given load is considerably smaller than with other types.

At the close of the meeting, which was the most successful in the history of the organization, the following officers were elected: E. H. Beil, of Youngstown, president; W. O. Anderson, of Canton, vice-president; and D. L. Gaskill, of Greenville, secretary-treasurer, the latter being reëlected. Cedar Point was chosen as the place of next meeting.

New President for Westinghouse Company

On July 29, George Westinghouse was deposed from the presidency of the Westinghouse Electric and Manufacturing Company. It will be remembered that in the panic of 1907 the Westinghouse Company went into the hands of a receiver and when it was rescued by its banking creditors and reorganized fourteen months later, the direction of the company's policies was largely left in the hands of the chairman of the board, Robert Mather, who was formerly head of the Rock Island system and has represented the bankers. Friction between the organizer of the company and the chairman of the board is given as the reason for the removal of Mr. Westinghouse.

Edwin F. Atkins, of the Boston firm of E. Atkins and Co., was elected to succeed Mr. Westinghouse as president, but it is reported that the choice is only temporary, and it is believed that E. M. Herr, long associated with the Westinghouse Company and a vice-president since the company's financial troubles, will be the eventual head of the company. The new president, Mr. Atkins, has been a member of the board of directors since the company's organization and it is understood that he accepted the presidency with the distinct understanding that he was to hold office only until a permanent successor should be chosen.

At the same meeting the other retiring officers were reëlected and additional officers were elected, including G. W. Hebard and H. D. Shute, who were named acting vice-presidents.

Ex-president Westinghouse is president of the French, Italian and Russian Westinghouse companies and chairman of the board of the English company. He is also president of the Westinghouse Air-Brake Company, the Westinghouse Brake Company, Ltd., of London, the Westinghouse Lamp Company and the Westinghouse Machine Company. There are eighteen American and foreign companies which bear his name, and the stock capitalization of the American companies organized to exploit his inventions totals close to $100,000,000.

SOCIETY NOTES

A joint excursion of the American Societies of Civil and Mechanical Engineers to Panama is talked of for some time in March.

On Sunday, July 31, the members of Rhode Island Association No. 2, National Association of Stationary Engineers, held their annual outing and field-day at Palace Gardens, Narragansett bay. Out of town guests were given breakfast at the Narragansett hotel. At eleven o'clock a light lunch was served at the pavilion, after which the usual base-ball contest between the supplymen's and the local engineers' nines started. Promptly at two o'clock the game was stopped by the umpire, who announced that "Dinner is being served." Members and guests to the number of 275 filed into the dining shed and sat down to a perfectly cooked "shore dinner." After every appetite was satisfied the crowd gathered in the concert hall, where a musical and recitative program was furnished by the "Bunch." Brief speeches were made by National President Reynolds, Massachusetts State President Fuch, Massachusetts State Deputy Kearney, New York State Deputy Coe and J. D. Taylor, secretary-treasurer of the life and accident department. All who attended were presented with white silk souvenir badges, from the lower end of which was suspended in a gilt setting the photograph of Thomas P. Burke.

On July 14, President Taft withdrew several more millions of acres of coal lands in different States of the West, bringing the total of coal-land withdrawals made by him up to the enormous total of 71,518,588 acres. Something like half of this amount are new withdrawals. The areas covered are in several different States.

M. E.'s en Route for England

The steamer "Celtic" of the White Star line, sailing from New York on July 16, carried 148 members of the American Society of Mechanical Engineers and their families en route to accept the invitation of the British Institution of Mechanical Engineers to take part in their summer meeting at Birmingham and London. One hundred others had gone by earlier steamers or other routes.

With such a party an enjoyable voyage would have been assured even under a less solicitous administration and less propitious weather conditions; but with everybody connected with the boat in thorough sympathy with the occasion, and a sea which even the unaccustomed would have been willing to see a little more strenuous, the trip was one which will be memorable for opportunity and for enjoyment to those who were privileged to take part in it.

On Monday evening a reception was held by the officers and past presidents of the society, and for the first time in the history of the line the officers of the ship assisted in receiving.

On Tuesday evening Past President W. R. Warner delivered a lecture illustrated with lantern slides upon astronomy.

For Wednesday evening a musicale was arranged, most of the numbers of which were furnished by members of the party.

On Thursday evening Past President John R. Freeman gave an illustrated lecture upon the Panama canal.

On Friday evening there was a moonlight dance upon the deck. On Friday afternoon numerous amusing contests were carried on upon deck and the prizes to the winners of these as well as of the tournament games were awarded on Saturday evening. Presentations to Captain A. Hamilton and Chief Engineer C. C. Lapsley, R. N. R., were made earlier in the day.

At this posting (Saturday night) we are nearing Queenstown, and expect to reach Liverpool ahead of the schedule.

BOOKS RECEIVED

WORK, WAGES AND PROFITS. By H. L. Gantt. *The Engineering Magazine*, New York. Cloth; 194 pages, 4¾x 7½ inches. Price, $2.

STEAM TURBINES. By Walter S. Leland. American School of Correspondence, Chicago. Cloth; 135 pages, 6½x 9¾ inches; 99 illustrations; indexed. Price, $1.

ENGINEERING CHEMISTRY. By Thomas B. Stillman. Chemical Publishing Company, Easton. Cloth; 744 pages, 5¾ x9 inches; 174 illustrations; indexed. Price, $5.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

THE Singer Building in New York is the tallest in the world.

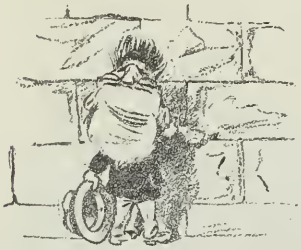
We say that today, reserving the right to retract it tomorrow.

This is an age of gone today, here tomorrow, in building enterprise. You never can tell. As Mr. Dooley puts it, " 'Tis a wonderful time we're livin' in, Hinnessy. Today they build a 20-story skyscraper and t'morrow they tear ut down t' make way fer a modern buildin'."

Today, to the best of our knowledge, the Singer Building is "it."

It rears its massive height, floor after floor, culminating in the tower that seems to pierce the sky. It makes its contemporaries look like six nickels and the Tower of Babel a pigmy.

Stand a few blocks away from it and you see it in all its majesty and impressiveness. Get close alongside of it and your cramped view reveals only a pile of stone. Your viewpoint is all wrong; it's distorted, out of focus. You are too close to your object; you only half see it; you see too much of some of it—and the magnified parts are not necessarily the important ones.



A lot of advertisers and readers of advertising are in the position of the man who stands against the Singer Building, looks up and thinks he sees it all. They are too close.

So many advertisers assume that the reader knows all about their product anyway, and that it is a waste of space and words to tell him anything in their advertisements.

So many readers of advertisements are unwilling to believe that anybody can tell them any way to save time or money in *their* business. They resent the thought.

And, therefore, many advertisements say nothing or exaggerate the unimportant and many readers close their eyes and minds to arguments and appeals that might prove of untold benefit to them.

The remedy? Get a better view-point.

Stand off a little way, get up on a hill and look at the situation.

Let the advertiser put himself into the shoes of the man he wants to use his goods. Remember that the possible buyer is not interested in the advertiser's age, personal habits or the brand of union suit he wears. The *article*, the *completed* article, that's the thing! *Why* should the possible buyer own it—get the *reasons* into the advertisement. Get them there plain and clear. Stick to words of one syllable, if possible. The average reader is not a walking dictionary, yet we know of one advertiser who describes his product with an adjective that, as he proudly says, the "dictionaries haven't got yet."

Let the reader of advertisements remember that advertised goods must make good or fail utterly. Let him remember that no fake can be successfully advertised continuously. Bear in mind that statements made in advertisements are "on record"—they're part of the contract. The advertiser must back them up. Remember, that any good device for power-plant use was designed to fulfil a need or meet a requirement. You can't tell how close it will come to fitting your case until you investigate thoroughly.

Let both advertiser and buyer get a clearer, better view of what *advertising* is.

It is simply the spreading of worth-while information among those whom the information may benefit and who may benefit the advertiser.

Some folks approach an advertisement as if they are afraid it will bite. The ad-writer is yet to be born who can get any "hypnotic" quality into his "copy." No advertiser can *force* his product on you. He simply lays his case before you and you become judge, jury, court officer and all the rest of 'em.

Advertising is a great *force*. How great, it is hard to estimate. It is for two classes—the buyer and the seller. No seller is benefited by its use any more than are the buyers of his product in the aggregate.

Which is a thought to take to heart next time you're "tempted" to investigate the merits of an advertised article.

Power
and The Engineer

Vol. 32 NEW YORK, AUGUST 16, 1910 No. 33

AMBITION to begin with and a little encouragement from others to brighten the flame has been the secret of many a man's success.

It is possible to kill ambition by withholding encouragement. The latter costs nothing, and frequently proves to be a good investment.

However, if a man has no desire to become greater, to perform better work, or to hold a more responsible position, he is not very likely to receive much encouragement, and may consider himself fortunate if he can hang on to the same old job.

In a Southern cotton mill power plant a young man, possibly eighteen or twenty years of age, holds a position as oiler. He works from five a.m. until six p.m. each work day for a wage of $8 per week.

He is not satisfied with his work, hours or wages, but, although laboring under conditions which would be obnoxious to most men, he lacks the ambition to prepare for something better.

The South holds out opportunities for engineers, but this oiler lacks the ambition to fit himself for any of them, and if someone were to force this man into a good position, he could not hold it.

It is doubtful if encouragement, and plenty of it, would have much effect in his case, because he lacks the incentive to get started in an upward direction.

He will see others come and go, prosper and obtain positions of trust and responsibility, while he is left with his oil can and greasy waste.

The ambitious man is bound to receive more or less encouragement in his work.. The last paragraph of the article entitled, "Economical Fire Room Methods," which appeared in the August 2 issue, illustrates this point.

Boys working as helpers and firemen have been taught to read and write, and were thus enabled to fill better positions.

The head fireman, because of his ambition and perseverance, has been promoted by his chief to the position of chief engineer of an up-to-date turbine plant.

Not because he was a favorite, but because he had ambition to become something better than a head fireman, which was undoubtedly stimulated by the encouragement he received from the chief.

The man without ambition is to be pitied.

The man with a burning desire to improve his position is likely to succeed.

And the man who "hands" out an encouraging word here and there exerts a silent but powerful influence over the other two.

Do not be satisfied with the position you hold while there is something better in view, and do not forget to encourage the little fellow who is struggling to gain a higher level. There is plenty of room for everybody.

Large Noncondensing Power Plant

By Warren O. Rogers

The power plant of the Birmingham Railroad, Light and Power Company is probably the largest central station in the United States running both reciprocating engines and turbines noncondensing. The exhaust steam is sold for heating purposes during the winter months. The power plant contains 31 water-tube boilers, totaling 17,200 horsepower; six Corliss engines, both double and compound, aggregating 9850 horsepower, and two 3000-kilowatt Curtis turbines. Both alternating and direct currents are generated.

Birmingham, Ala., is situated about midway between the Cahawba and Warrior rivers, each some six or seven miles distant. As a result, most of the steam plants in that city are run noncondensing, as condensing water is procurable only at a cost that practically prohibits its use without the installation of cooling towers or cooling sprays. It is for this reason that the steam-generating units of the Birmingham Railway, Light and Power Company's plant are run noncondensing.

This power plant furnishes current for the street-railway system and for commercial illuminating and power purposes throughout the city, as very few of the large office buildings are equipped with isolated plants. The plant has been remodeled, the transformation being from the old type of belt-driven unit to the modern direct-connected unit.

ENGINE ROOM

The engine room contains six Corliss engines, one of 500-horsepower, two of 1300-horsepower and three of 2250-horsepower capacity. The small engine has cylinders 20x30 inches and was originally used as a pair, but at present one side is disconnected and the two direct-current generators mounted on the engine shaft are driven by one engine. The generators are each of 2000 kilowatts capacity, at 125 volts, the armature revolving at a speed of 150 revolutions per minute. This unit carries the engine and boiler rooms lighting load, in case all the other generators are stopped, or it can be used as an exciter, when the motor-driven exciters are out of commission.

The two original 1300-horsepower engines now have 20 and 20 by 48-inch and 28 and 44 by 48-inch cylinders, one being a double and the other a cross-compound engine. Each of these engines is direct connected to an 850-kilowatt direct-current railway generator. The speed is 80 revolutions per minute. Of the three 2250-horsepower units, one is a double Corliss engine, with cylinders 36x60 inches. The other two are cross-compound Corliss engines, both having cylinders 36 and 60 by 60 inches. One of the cross-compound engines is direct connected to a 1600-kilowatt direct-current railway generator, while the other two units drive 1500-kilowatt alternating-current generators.

Besides the engine units there are two Curtis turbines driving 3000-kilowatt 60-

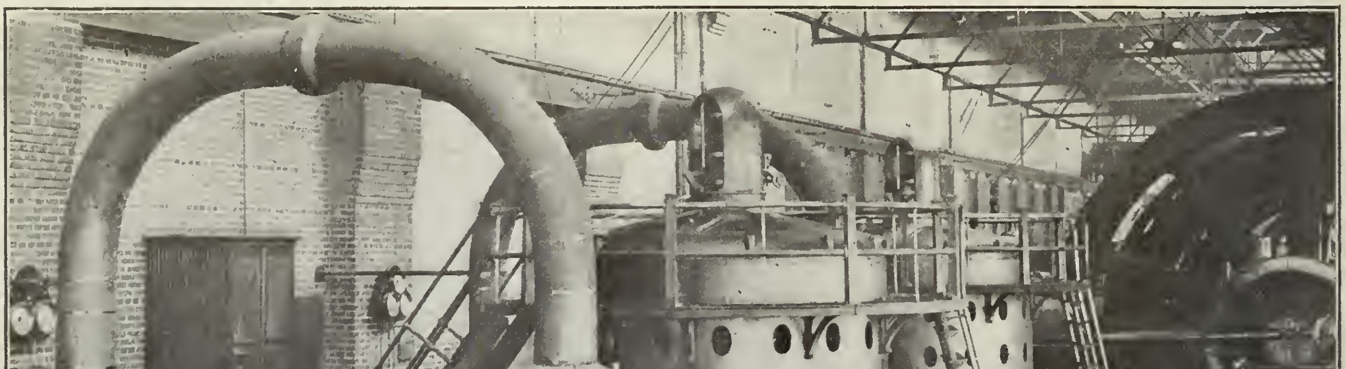


FIG. 1. VIEW OF ENGINE ROOM FROM THE TURBINE END

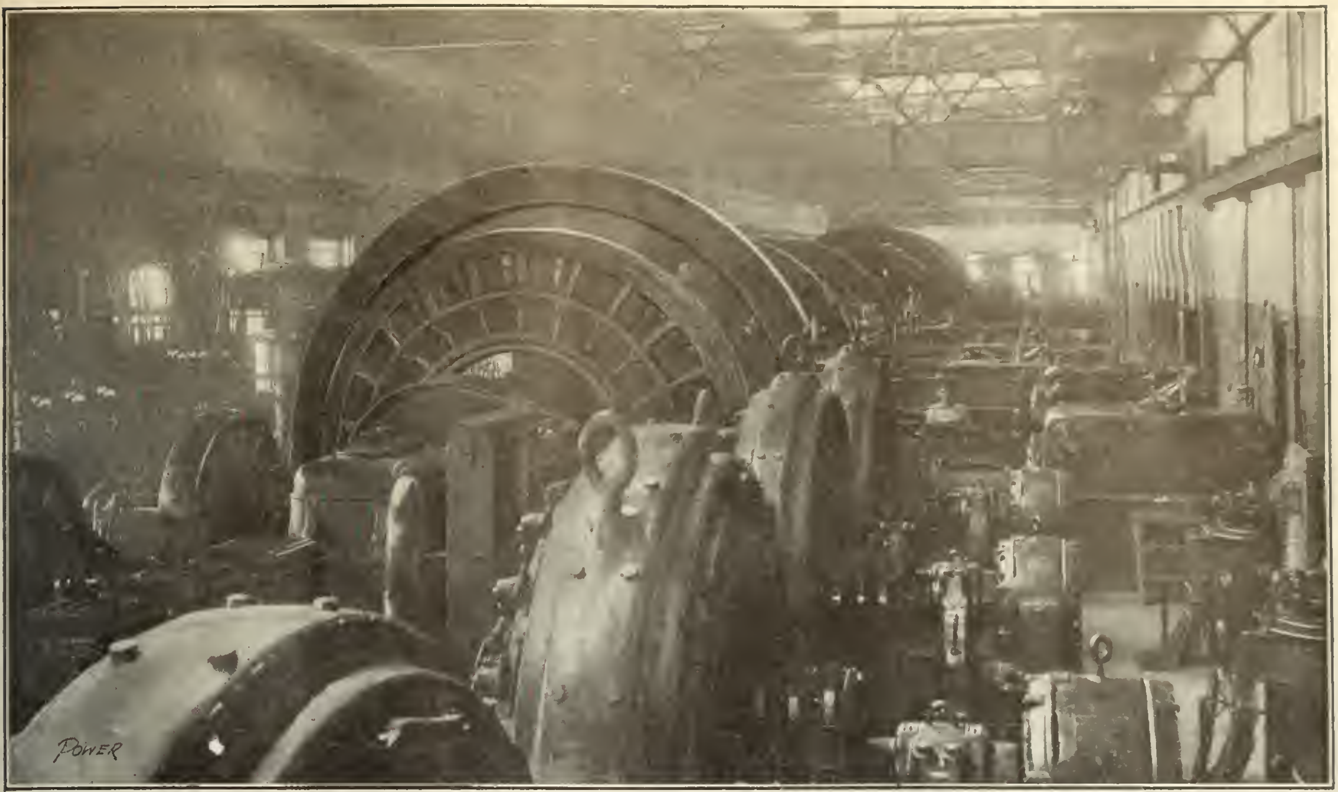


FIG. 2. VIEW OF ENGINE ROOM, SHOWING RECIPROCATING UNITS AND ROTARY CONVERTERS

cycle generators. All generators in the plant are of General Electric make. The controlling valves on the turbines are electrically operated. Fig. 1 shows the turbine units in the foreground and the reciprocating engines in the rear.

There are seven rotary converters, five being of Westinghouse make. Three of these are rated at 750 kilowatts and the other two at 1000 kilowatts.

The remaining two are General Electric machines. Two motor-driven exciter units furnish current for excitation, the smaller unit consisting of a 75-horsepower induction motor direct connected to a 125-volt generator. The larger exciter consists of a 175-horsepower induction motor driving a direct-current generator at 600 revolutions per minute.

In the pit below the engine-room flooring, and on a level with the floor of the turbine pit, is a 300-kilowatt railway booster, and just above it, on the main floor, are three 750-kilowatt transformers. At the extreme end of the plant are located nine 50-kilowatt and five 100-kilowatt transformers.

BOILER ROOM

In the boiler room, Figs. 4 and 5, are twenty-four 600-horsepower water-tube boilers, and seven 400-horsepower water-tube boilers all of Babcock & Wilcox make. They are arranged as shown in Fig. 3, and are all equipped with chain grates, with the exception of four 600-horsepower boilers. Of the total number of boilers twenty-one are fitted with the Dutch-oven furnaces, the remainder with furnaces of regulation design. The pressure carried on all of the boilers is 175 pounds per square inch.

There are three stacks, each 175 feet high, one being 10 feet in diameter and two 12 feet in diameter. The two larger

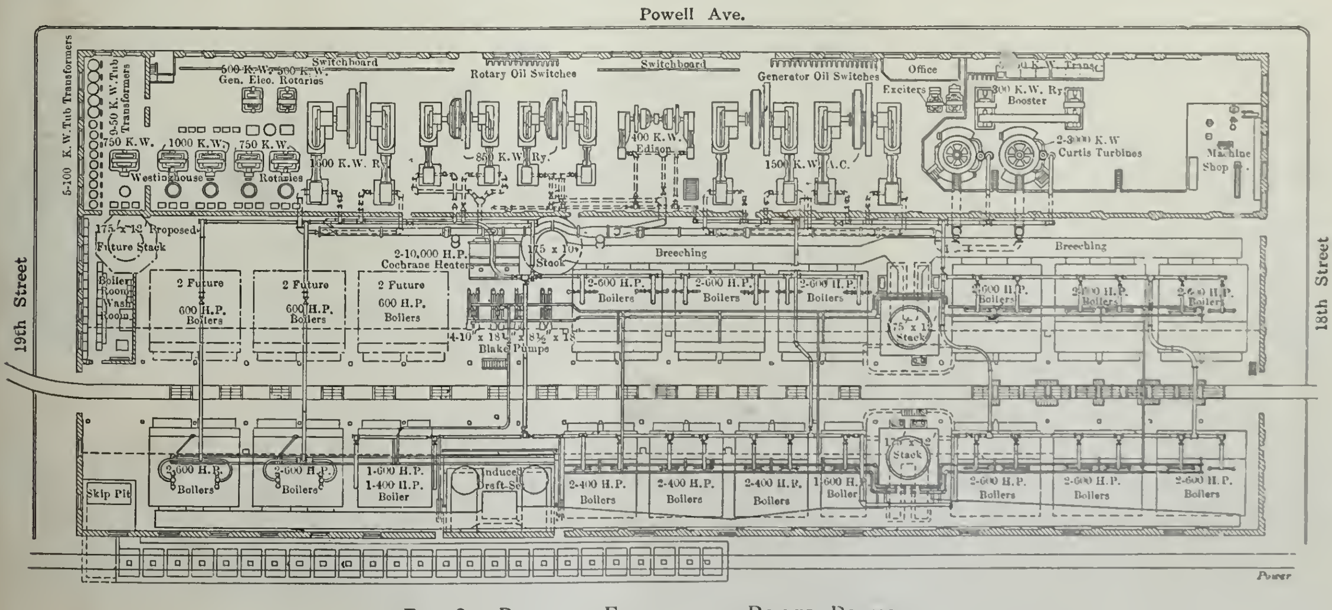


FIG. 3. PLAN OF ENGINE AND BOILER ROOMS

stacks are constructed of brick and the smaller of steel.

All stokers are driven from shafting running in front, and at the top of each row of boilers, each stoker being operated by an eccentric and rod, the latter being connected to the ratchet of the feeding device. These shafts can be driven by three 15-horsepower vertical Waches steam engines or four General Electric 7½-horsepower induction motors.

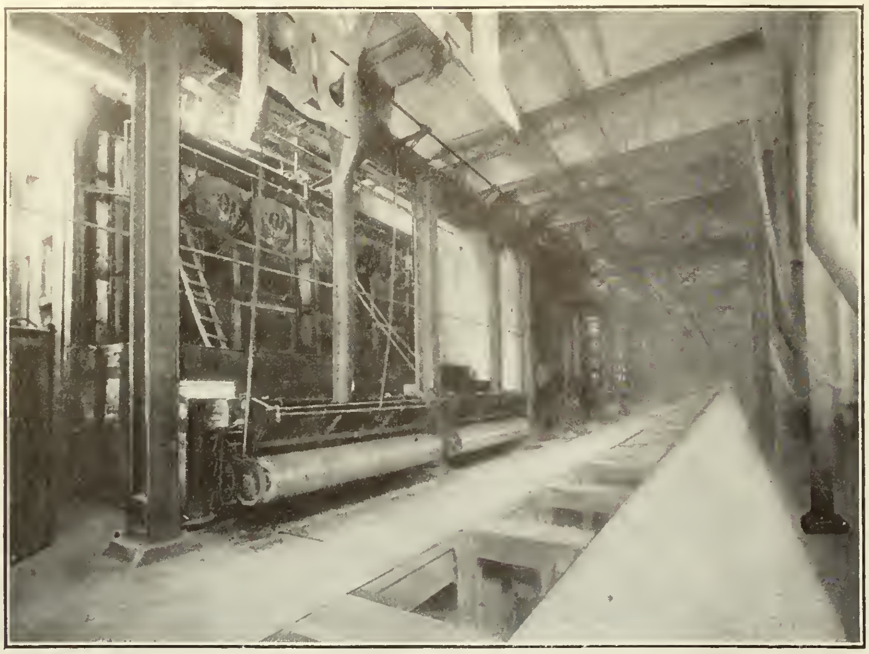


Fig. 4. Partial View of Boiler Room

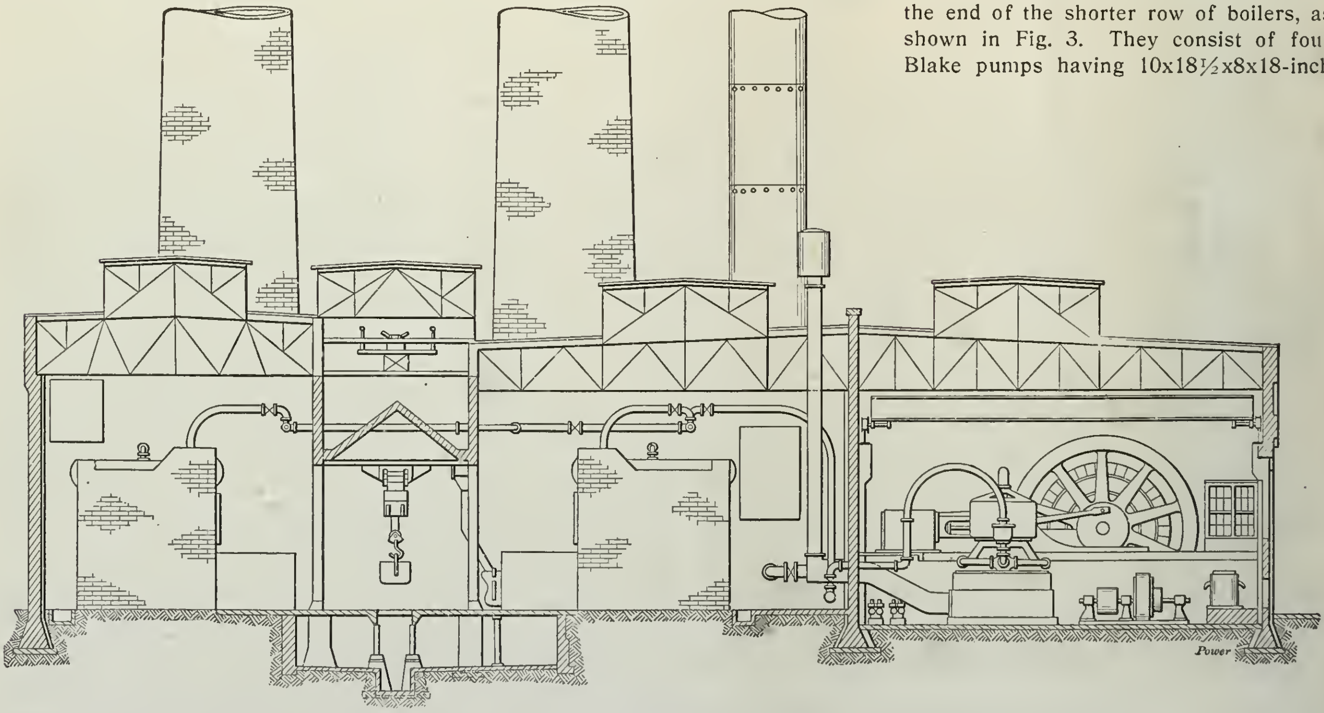


Fig. 5. Section through Turbine End of Boiler and Engine Room

Piping

A general plan of the piping arrangement for line and exhaust steam is presented in Fig. 3. The exhaust pipes from each turbine pass through the walls forming the partition between the boiler and engine rooms, and extend along the opposite side of the wall to the extreme end of the building. The engines exhaust into a common header which supplies steam for commercial heating and also for heating the boiler feed water in two 10,000-horsepower Cochrane heaters.

Exhaust steam is utilized for heating various buildings in the city during the winter months, so that operating the double engines and even the compound engines and steam turbines noncondensing has some advantages, and in this particular plant reduces the cost of the electrical energy per kilowatt-hour to a figure that compares favorably with some of the more modern and better equipped power plants.

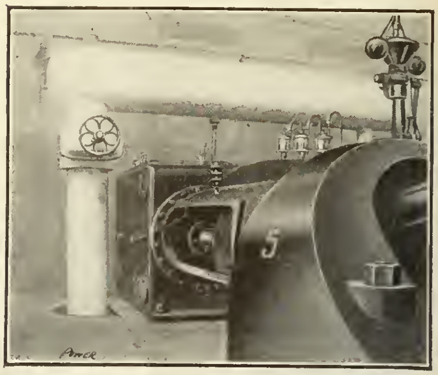


Fig. 6. Position of Receiver

In the case of the double engine the exhaust steam passes to the heating main with a back pressure of 10 pounds per square inch. The cross-compound engines have the receiver located above the cylinders, as shown in Fig. 6. The steam for the heating system is not taken from the exhaust of the low-pressure cylinders but from a tee on the high-pressure exhaust pipe connecting the cylinder and the receiver.

Fig. 7 shows the arrangement of the boiler-feed pumps which are placed at the end of the shorter row of boilers, as shown in Fig. 3. They consist of four Blake pumps having 10x18½x8x18-inch cylinders, the water end being of the outside-packed type with pot valves. The arrangement of the steam and water piping is shown to good advantage.

The method of handling the coal and ashes will be described in a separate article.

Probable Future Boiler Practice

By James E. Steely

A discussion of the heat distribution in a boiler and a plea for such design as will enable each part of the heating surface to contribute an equal share of the total evaporation.

The capacity of a boiler is a variable factor. Builders usually allow about 3½ pounds equivalent evaporation per square foot of heating surface in rating a boiler. Under the present standard of construction this is about the evaporation which is possible without forcing. However, no two sections of the boiler will do the same work; the part close to the fire will generate a large portion of the steam while that near the flue will generate very little. Some idea of the relative amounts done by the various rows of tubes was given by an instrument, used some time ago, to indicate circulation in the boilers of the Geological Survey plant at St. Louis. The bottom row of tubes were entirely inclosed in a fire-clay baffle; notwithstanding this protection from the heat, the machine made three times as many revolutions when placed in one of the bottom row of tubes as when placed in the tube immediately above it. When placed in the third row from the top it revolved very slowly. This demonstrates that the first row of tubes does many times as much work as any of the others and that the upper rows do very little. Now if the average evaporation is 3½ pounds of water per square foot of heating surface, the lower row may be evaporating 8 or 10 pounds, while the upper row is evaporating possibly only one-tenth of a pound.

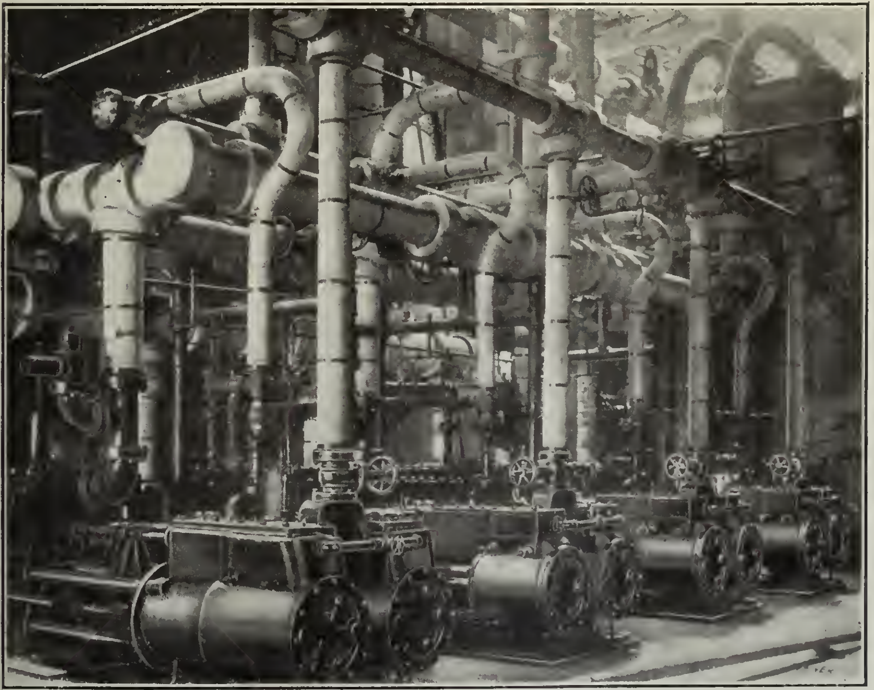


Fig 7. Arrangement of Feed Pumps and Piping

The size of the grate of the modern boiler is such that, when burning coal as rapidly as good practice and economy permit, only about 3½ to 5 pounds of water can be evaporated per square foot of heating surface. Some time ago, one of the New York power stations was equipped with a Babcock & Wilcox boiler having two Roney stokers, one placed under each end of the boiler, thereby doubling the grate surface. About 75 per cent. overrating was obtained by this arrangement without any considerable decrease in efficiency. This would indicate that the trouble with out modern standards is that the ratio of heating surface to grate surface is too large. The arguments for this practice of high evaporation are low first cost, economy of room and smaller loss due to radiation; those against it are increased maintenance cost and rapid deterioration of the boiler.

The capacity of a boiler is approximately proportional to the velocity of the gases and water. Thus the more rapid the circulation of water in the boiler or the more rapid the passage of the gases, the more rapid will be the generation of steam. Probably every engineer has seen diagrams showing the difference in ebullition between where there is circulation and where there is none. Theoretically the correct way to get the highest economy is to have the water circulating in the opposite direction to the hot gases, but the writer knows of no way in which this opposed current could be obtained and still allow free circulation of water in the boiler. If the circulation were restricted, the steam would be liberated with violence. It may be stated, however, that when a water-tube boiler of the Babcock & Wilcox or Stirling types is set with vertical baffles, this opposed current arrangement is secured to some extent. This is about as much as can be expected without sacrificing the free circulation of the water in the boiler, which is more important. The opposed flow should be secured by stages just as steam is expanded in doing work. It is out of the question to economically obtain a complete expansion of steam in one long cylinder, but the same end is arrived at with excellent results in two, three, or four stages. While there is no analogy between steam expansion and heat absorption, the same principle may be applied to obtain the best results. In other words, the boiler and economizer are better when separated than combined in one. If a two- or three-stage economizer were used, it would cool the flue gases as low as any boiler which could be built and with much less complication. Plants using noncondensing engines usually have enough exhaust steam to heat the feed water as hot as it is possible to handle it in the pump and when this hot water is further passed through a good economizer it is raised to almost the temperature of evaporation, leaving merely the work of evaporation to the boiler. If the exhaust steam were not available, a second economizer could be used in tandem, but the temperature of the flue gases would be so low as to seriously affect the chimney draft.

It has been stated that efficiencies as high as 90 per cent. are attainable. If one closely examines the heat balance of an ordinary boiler test, it will be seen that it is almost impossible to obtain any such result except perhaps with a very bulky and expensive layout. The "unaccounted for" losses and the loss due to incomplete combustion usually amount to over 10 per cent. and it would be necessary to return the burnt gases to the air at the same temperature as they were received. When temperatures are taken for calculating the chimney losses, they are taken before the gases pass through an economizer, and when the loss up the stack figures 18 to 25 per cent., it must be kept in mind that the economizer will take up considerable of this apparently lost heat. Thus while the efficiency of the boiler might be only 65 per cent., the efficiency of the boiler and economizer might be 75 per cent. The results of recent investigations show that a high furnace temperature means a high stack temperature and that a large increase in the combustion-chamber temperature is accompanied by very little increase in efficiency.

The problem of heat absorption is discussed fully in the Geological Survey bulletin No. 325 and considerable experimental data as well as conclusions drawn therefrom are set forth. Everyone interested in heat absorption should study these experiments carefully as many of them result contrary to the logical method of reasoning.

In the writer's opinion a boiler should be proportioned so that all the heating surfaces would do equal work. If this result could be attained, two or three times as much steam would be generated per unit as that which is now possible.

Corliss Engine with Two Eccentrics

By Prof. R. C. Heck

A discussion of the advantages of operating the steam and exhaust valves by separate eccentrics, by which method a greater range of cutoff is obtained.

The purpose in using two eccentrics on a Corliss engine is to obtain a greater range of cutoff under the control of the governor. With one eccentric, cutoff by release of the valve in the normal manner cannot occur much later than at about three-eighths of the stroke. With a separate eccentric for the steam valves, the governor may control cutoff to as late as three-quarters of the stroke.

In Fig. 1 is represented a typical releasing gear, with some nonessential details omitted. Piece *A* is the oscillating arm or double crank, continuously and positively driven by the wristplate. It carries the hook claw *B*, with latch arm *L* and trip arm *T*. Pin *P* is on the valve arm *D*, which is keyed fast to the valve spindle. This arm is here represented by dotted lines, since it is really in front of the other parts, and would cover up too much of them if shown in full. The knockoff cam C_1 and the safety cam C_2 are fastened to the cam ring *E*, which has a limited range of rotary movement, under the control of the governor.

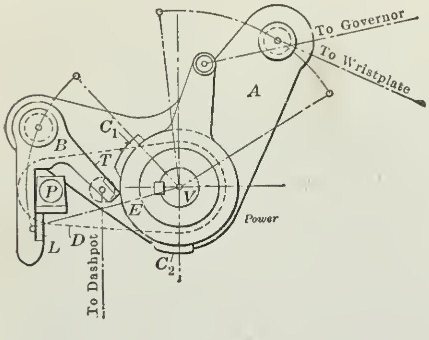


Fig. 1. Typical Releasing Gear

In order that the valve shall be released, for quick closure by the dashpot, arm *T* must strike cam C_1 before lever *A* gets to its extreme of travel toward the right. In Fig. 2 is shown the condition in which, with pin *B* on crank *A* at its right-hand limit of swing, arm *T* just touches cam C_1; then the valve will not be released, but will come back slowly with the reverse movement of the wristplate, cutting off very late in the stroke. A single-eccentric gear is usually so proportioned that this condition will exist when the governor is down on the safety pin, so that when the speed is below normal due to a heavy overload or a bad drop in boiler pressure, the valve gear will work without release.

If the safety pin or stop is out, the governor can drop still lower, ring *E* turning still farther in a clockwise direction; then arm *T* will slide up on cam C_2 when arm *B* is in the position for engaging the valve (position marked *B'* in Fig. 2), and latch *L* will be held out so that it cannot hook up, and the engine will stop.

Now consider the eccentric setting and its effect on cutoff, as set forth in Figs. 3 and 4. The former shows the position of the crank and eccentric arms with the crank on dead center. The eccentric setting, defined by the angle *C O E* or by the angle of advance *G O E*, is determined by the exhaust valve, so as to give proper release of the steam valve, and also compression. The particular angle on the diagram, $G O E = 15$ degrees, is a representative value for this type of gear, the advance being less than in a slide-valve engine.

Fig. 4 shows where the latest cutoff action by release of the valve can begin. The eccentric *O E* is still about 15 degrees above its dead center *O B*, but the wristplate is almost at the end of its travel. With the crank at 60 degrees as shown, the piston is at only one-quarter of the stroke; but the time it takes for the dashpot to pull the valve shut brings the actual valve closure to about 35 or 40 per cent. of the stroke. The range from this latest governor cutoff to the very late cutoff without release is entirely beyond the scope of utilization.

Suppose now that, as shown in Fig. 5, a separate eccentric $O E_2$ is used to operate the steam valves, leaving the exhaust valves to $O E_1$. This new eccentric can be set well back toward the crank; and when it gets near to dead-center position *O E*, Fig. 6 (the same as *O E* in Fig. 4), the crank will be at *O C*, corresponding to the position of the piston at perhaps five-eighths stroke, instead of at *O C'*. The 40 degrees, or so, of increased cutoff range is secured just where it will do the most good, for it is where the piston is covering the ground most rapidly. The angle *C' O C*, Fig. 6, gives the whole reason for adding a second eccentric to drive the steam valves, and shows this in about the clearest and most concise manner possible.

There is a limit to the amount by which the steam-valve eccentric may be set back toward the crank. If the negative angle of advance *G O E*, Fig. 7, is made too large, there will not be enough range of motion for the opening of the valve; that is, for bringing it from the rest position to the act of opening the port for admission of steam. Considering Fig. 7, we note first that the valve must be engaged or hooked up while the eccentric is near the dead-center line *O A*; and while it turns to *O E* (or to a slightly earlier position, because of the lead) the valve is moved through its lap distance. The term lap is now used to denote the overlap of the closed valve. With a range *A E* or *A F*, as against *A E'* or *A F'* with the proper setting of a single eccentric, it is evident that the steam valve must have a small closure lap; but this may not be too small, lest leakage take place.

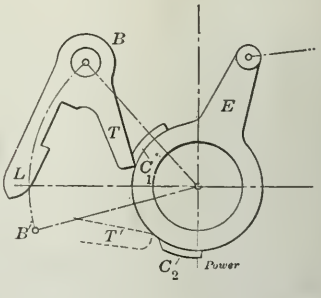


Fig. 2. Extreme Position of Hook

Giving to the word lap its usual meaning, or taking it to define the amount by which the edge of the valve overlaps the edge of the port when the eccentric is midway between its dead-center positions, we see that a steam valve driven by an eccentric set as in Figs. 5, 6 and 7 must have negative lap. This means that if the valve were kept in continual connection with the wristplate it would close the port through much less than one-half of a revolution, and leave it open the rest of the time. Consequently, it is most essential that the condition of no release, represented in Fig. 2, shall be absolutely avoided with a two-eccentric gear.

In Fig. 8 it is shown that with admission taking place when the eccentric is at *O E′* (crank on or just below the dead-center line *O C′*), cutoff by a valve in connection with the wristplate will not occur until the eccentric reaches *O E* and the crank reaches ***O C***. Since the exhaust valve is opened when the crank is above *O B*, it is evident that from this steam release down to *O C* there will be an open passage from the live-steam chamber to the exhaust pipe. With the proportions here used, live steam would pass through each end of the cylinder for about one-sixth of a revolution, or for about one-third of the total time, counting the two ends together. To make impossible this tremendously wasteful action, the requirement in terms of valve-gear design is that the cams C_1 and C_2 be set closer together on the ring, see Fig. 1, so that C_1 cannot go out of action before C_2 comes in.

In conclusion, it is appropriate to mention the scheme of increasing the cutoff range of a single-eccentric gear by giving the knockoff cam, corresponding to C_1 in Fig. 1, an oscillating movement so that even after arm *A* has begun to swing back from the position *B*, Fig. 2, the cam can follow up and overtake the trip arm *T*, releasing the valve. This idea has been applied on at least one design of Corliss engine for ordinary power service, and is in regular use on several makes of pumping engine.

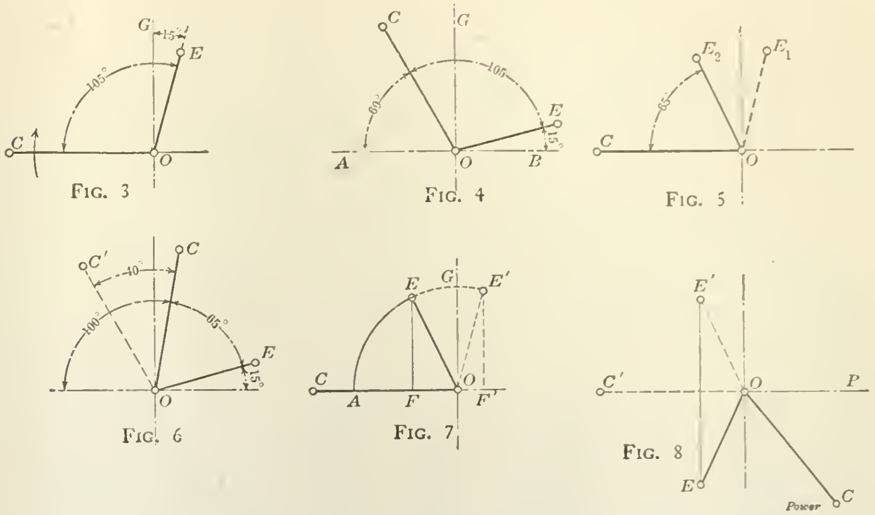


FIG. 3 FIG. 4 FIG. 5

FIG. 6 FIG. 7 FIG. 8

EFFECTS ON CUTOFF OF VARIOUS ECCENTRIC SETTINGS

The Painting of Smokestacks

By J. C. Hawkins

Frequently the engineer is called upon to paint the smokestack or at least to oversee the job, so that the suggestions given by the author on how to do the work and the kind of paint to use will not be out of place.

Painting a smokestack is not usually considered a part of the engineer's duties, but he is frequently required to do so, or at least oversee the work, and a few hints along this line may not be out of place. The methods described in the following are used by several stack painters and "steeple jacks." Take, for an example, a steel stack 5 feet in diameter and 125 feet high, with two sets of guy ropes, one set 50 feet, and the other 90 feet, from the base, without a line through the pulley or hook at the top of the stack. First it is necessary to get together a suitable rigging consisting of one set of a single and double block with about 250 feet of ⅜-inch rope, and another set of blocks with about 500 feet of ⅝-inch rope, also two or three hooks made as shown in Fig. 1.

HOISTING THE FIRST ROPE

In order to get the first line to the top of the stack, a pole, long enough to reach from the roof to the first guy band, is procured and a hook similar to *B*, Fig. 1, which is made of ½-inch iron and has a spread of 4 inches. This hook is secured to the double block of the smaller set of falls at the ring. The hook is attached to the end of the tapered pole, as shown in Fig. 4, with cord or wire. The end of the pole is tapered to permit of removing it from the loop *C*. This hook is hoisted and hung over the first guy-wire eyebolt, close to the stack, and used to hoist the double block of the large falls into place. When the heavy blocks have been hoisted to the guy, the end of the small line is made fast at the base of the stack and the seat, Fig. 2, fastened to the single block. Then, with a short piece of ⅝-inch rope to tie up to the first guy and an extra hook, the painter is ready to hoist himself up.

HOISTING THE FALLS

When the top block is reached the rope is made fast by bringing it under the seat sling and taking a hitch over the hook on the block, and then making a solid hitch with the short rope from the eyebolt to the seat sling, after which a helper can slack off on the small line. The pole is then prepared for reaching the second guy ring, as in the first instance. The painter steadies himself by pushing against the stack with his feet, and if, by any chance, he should get turned around, with his back to the stack, it is almost impossible to turn facing it again without a trip to the ground, or at best to the next lower guy.

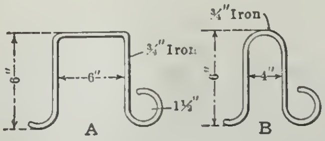


FIG. 1.

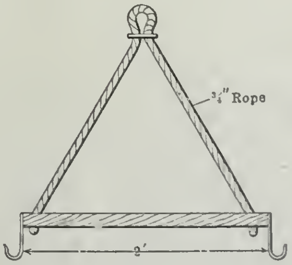


FIG. 2.

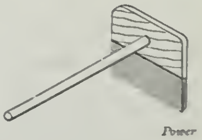


FIG. 3.

After hooking over the first guy ropes, the painter reaches the top guy by pulling the large blocks up and then pulling himself up, as before, from which position he can hook the pole over the top of the stack, using a hook like *A*, Fig. 1, and then proceed as before, to reach the top. Then by standing up on the seat and throwing one leg over the top of the stack, the double block of the large set of falls is hooked into the ring on the end of the hook as shown in Fig. 5, and the small falls are then lowered to the ground.

PAINTING THE STACK

The painter is then ready for paint and brushes, which are pulled up with a small line. A 2½-gallon galvanized-iron or other light pail for paint will be required, also a short-handled brush, about 6 inches wide, and a long-handled whitewash brush such as Fig. 3 represents, with a 4-foot handle to which a wire loop is fastened on which to hang the brush when not in use. A scraper or large putty knife will also be needed. To do a good job it will be necessary to make three trips to the top, because, when swinging around, the painter can only get a stroke or two at a time. After painting as far as the reach permits, both ways, the painter lowers himself a few feet, and in this way gradually approaches the ground.

When passing the guys care should be taken to pass on the side which is to be painted next, and when the ground is reached the hook on the top of the stack can be worked around by giving the ropes a side swish.

Having painted the second strip from top to bottom the painter performs a trick which, if done rightly, will save an extra trip to the top; that is, he prepares the double block so that from the ground it can be unhooked from the hook over the top of the stack. To do this he passes the end of the ⅜-inch line through the eye of the hook and makes it fast to the block below the hook, as shown in Fig. 6, so that when pulling down on the ⅜-inch line the block is tripped out of the hook *C* and can then be lowered to the ground. This leaves the small line through the hook at the top of the stack. Fasten a ¼-inch wire rope, or large soft copper wire, to the end of the ⅜-inch line, by winding with twine, so that it will pass easily through the eye of the hook *C*, which is left on the stack for the next painting. It should be pulled tight and fastened to keep it from chafing against the guys and breaking.

A separate hook with a pulley fastened to it can be used to trip the large block, if desired, but this leaves two hooks on the top of the stack, and the one will work just about as well.

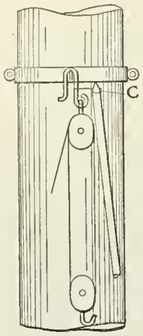


FIG. 4.

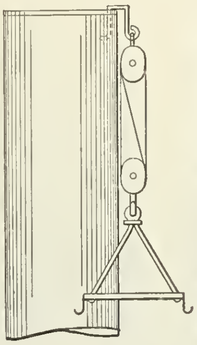


FIG. 5.

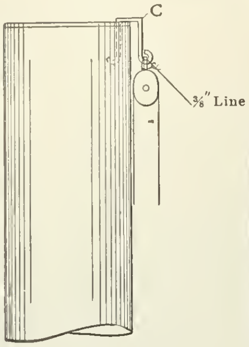


FIG. 6.

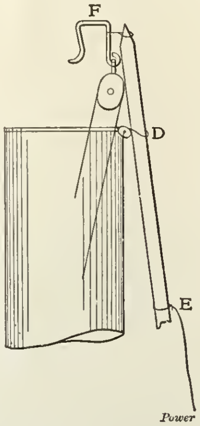


FIG. 7.

ANOTHER METHOD OF HOISTING THE BLOCKS

When there is a line through the hook at the top of the stack, the painter proceeds in a different manner to get the blocks to the top. A ¼-inch or ⅜-inch line is secured to the end of the wire rope and pulled through the hook, just the reverse of lowering the blocks. After getting the line through the hook, which may take some time and may break the wire rope or copper wire, as it will be worn thin in some places, a light pole, about 2 inches square and 12 feet long, is obtained and the ⅝-inch double block fastened to the tapered end of the pole, as shown in Fig. 7. Tie one end of the small line to the pole at *D*, about 3 or 4 feet from the top, and tie the other end to the bottom of the pole at *E*. The ⅝-inch blocks should be run out full length before hoisting. When the end of the pole, with the hook, reaches the top of the stack, a little twisting of the ropes will soon bring the hook *F* over the top of the stack, when the pole is lowered and pulled out of the loop and let down, leaving the block in place.

It would be easier to fasten the block to the end of the light line and pull it up, using the old hook and light line to work on, but it is not safe, as one cannot tell what condition the hook is in, and a good hook should always be used to work on.

SHIFTING FROM NEW TO OLD HOOK

When the top is reached, if the old hook is found to be in good condition, the blocks may be shifted to it by using the short rope to hold the seat up to the first hook; then slack off on the blocks and transfer from one hook to the other. When leaving the top for the last time the small line is adjusted to trip the block out as in the case mentioned and the wire rope or copper wire is pulled up as before. Quite often the band around the top of the stack becomes loose, caused by the top of the stack eating away. This can be repaired by driving out the old rivets and replacing them with bolts with a large washer on each side, using clamps to hold the ring in place while putting in the bolts.

As this class of work is somewhat dangerous, the painter must personally see that all knots and ropes are right before trusting them, and he should have a good helper. A good ½-inch rope would, no doubt, be strong enough to hold the painter, but a ⅝-inch rope should be used as a measure of safety.

PAINT TO USE

Many use a paint made of coal tar, which is thinned with creosote or dead oil, but there are serious objections to this paint because the creosote and tar both have a tendency to pit or corrode the iron.

Before the stack is put up it should have two coats of red lead and linseed oil on the inside and a coat of graphite paint on the outside. The latter is a better preservative of the iron than tar paint but its disadvantage is its dull color; and if put on over tar paint it will peel off, as the oil works under the tar and loosens it. If put on often enough, however, until the tar is all off, it will give good service.

A good paint may be made by mixing flake graphite, linseed oil and lamp black together, the lamp black being used only as a coloring matter and the graphite as the body of the paint.

A good lubricant for hoisting ropes, according to *Mines and Minerals*, is made by mixing one bushel of freshly slaked lime to a barrel of coal tar, or a mixture of pure tar and tallow can be used. When pine tar which contains no acid is used as a base lime is unnecessary, as tar is solution proof to ordinary mine water. Another good mixture contains tar, summer oil, axle grease and a little pulverized mica, mixed to a consistency that will penetrate between the wires to the core and will not dry nor strip off. The lubricant should not be so thick as to prevent inspection of the rope, and after the first application should be used sparingly, so that the rope may be kept clean and free from grit. Graphite mixed with grease is another lubricating mixture that is used successfully, and sometimes pulverized asbestos is used instead of graphite.

The Design of Rope Drives

By Francis H. Davies

The influences of length and angle of drive, form of groove, ratio of pulley diameters, speed, size and character of rope, are outlined. Special types of drive, such as the cross, half-cross and angle, are considered.

LENGTH AND ANGLE OF DRIVE

Of the many points to take into account in designing a rope drive the questions of length and angle are of high importance as both have a material influence upon the efficiency of the system. Upon the arc of contact which the ropes make with the pulleys depends the amount of power which it is possible to transmit. Failure to appreciate this fact is bound to lead to unsatisfactory results. Under any given set of conditions the longer the drive the greater will be the arc of contact, and in the case of horizontal working it is usually productive of the best results to have a distance between the shafts of at least three times the sum of the radii of the driving and driven pulleys. This will give an arc of contact of a reasonable number of degrees and will allow the ropes to be worked at their proper horsepower. A smaller arc will only deal effectively with a lower power. A drive which is too long is subject to serious jumping of the ropes under any sudden alteration of load, and to reduce this to a minimum it is necessary to increase the tension to a degree which, from the point of wear, both of the ropes and the bearings, is not desirable. Matters may be improved very considerably by the use of a grid or comb through which the ropes run, but, obviously, it is better practice to dispense with such adjuncts, if possible, and keep the length of the drive within reasonable limits.

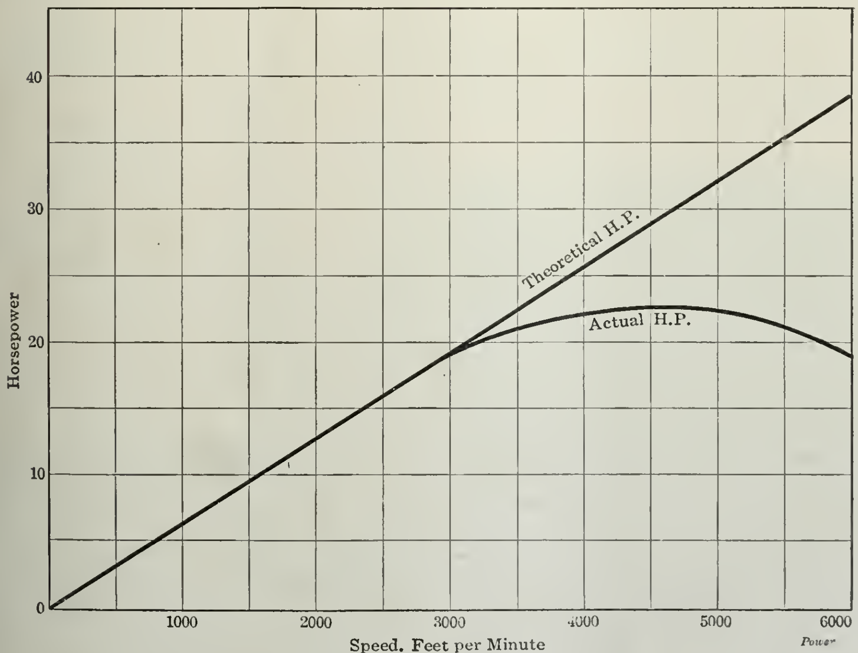


FIG. 1. EFFECT OF CENTRIFUGAL FORCE ON POWER TRANSMITTED BY 1¼-INCH ROPE

It is in horizontal running or at angles to the horizontal up to approximately 45 degrees, that rope driving gives the best results. In a vertical drive or one in which the angle is more obtuse than 45 degrees, the arc of contact becomes too low to give a good grip on the pulleys, and hence for any given power a greater number of ropes must be used. In a vertical arrangement, the power it is possible to transmit will be reduced about 25 per cent. from this cause. Where it is impossible to avoid the vertical or a very obtuse angle it is generally productive of better results if the bottom pulley acts as the driver, and under these conditions the single- or continuous-rope system is often preferred to the multiple drive as it is essential that the rope should be kept tight to prevent it hanging off the bottom pulley and the tension gear employed with a single rope is particularly useful in this respect.

TOP OR BOTTOM DRIVE

As a general rule it is preferable that the bottom should be the driving side, as under this arrangement the sag of the top side considerably amplifies the arc of contact. However, the reverse is sometimes purposely employed where, owing to a rapidly varying load, the ropes have a tendency to surge or jump. With the slack side at the bottom there is less likelihood of the ropes leaving the grooves, but they will not, of course, transmit the same power and a greater number must therefore be used.

PULLEYS

Under ordinary conditions the pulleys should not be less in size than thirty times the diameter of the ropes. This is a general and safe rule, but circumstances occasionally necessitate some deviation from it. Where the pulley has to be of a smaller ratio the amount of power transmitted per rope must be reduced by the addition of other ropes to offset the excessive wear which takes place under such conditions. The more a rope is bent in passing over a pulley the greater is the internal friction set up between the strands and fibers, and the effect of this soon becomes apparent in the shape of wear. As the less bending a rope is subjected to the longer it will wear, it is always advisable to err on the side of large pulleys.

GROOVES

The groove of which the sides form an angle with one another of 45 degrees has now become standard practice as it has been found to give the best results. It combines a maximum of frictional adhesion when the rope is on the pulley with a minimum of resistance when it leaves it, and does not tend unduly to convert the natural circular shape of the rope into that of a V. A more acute angle, while securing a better grip, wears the rope as it leaves the groove and very shortly alters its shape in such a manner that its efficiency drops. An obtuse angle of 30 degrees or thereabout has little wearing effect, but it has been found by experience that it does not suffice to give the necessary grip. Some engineers prefer an angle of 40 degrees as there is then less tendency on the part of the rope to rotate around its center and so wear the surface; but the 45-degree groove with perfectly flat sides is by far the most usual and is, on the whole, considered the better. It is very difficult to assign a reason for the revolving of ropes as the phenomenon appears to follow no regular law. Generally speaking, it is deprecated, but there are not wanting engineers who prefer a certain amount of rotation. They contend that with rotation the wear is made more even.

It should be borne in mind that the rope should under no circumstances be allowed to touch the bottom of the groove as this implies slip owing to the fact that under such conditions the rope cannot be wedged in the groove as it should be. Partly for this reason it is often advisable when ordering ropes to send a templet of the grooves to the rope maker, thus throwing upon him the responsibility for the success of that part of the drive. This is particularly the case where the grooves on the two pulleys differ in any respect as may at times be the case. So long as the difference is not enough to cause the rope to ride too high above the teeth in the one case and too near to the bottom of the groove in the other there is no difficulty in working under such conditions.

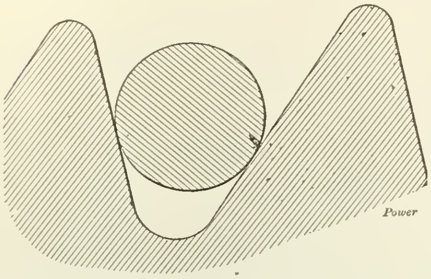


Fig. 2. Groove Suitable for Out-of-line Drive

Number and Size of Ropes

Numerous tables have been published from time to time giving the horsepower that ropes will transmit at various speeds, and for ordinary conditions they are doubtless reliable. It must not be forgotten, however, that relative pulley diameter, length and angle of drive have an important bearing upon the power it is possible to transmit, and as before stated, it may be necessary under certain conditions to increase the number of ropes by 25 per cent. or more to insure freedom from trouble and undue wear.

In diameter ropes are seldom less than 1 inch, and owing to the resistance offered to bending and the consequent waste of power, a diameter of 2 inches may be taken as the other extreme. The most usual sizes are between 1¼ and 1¾ inches. These constitute a happy mean which gives a minimum of power loss by bending together with a maximum of weight for the production of the necessary amount of sag. Where the multiple-rope drive is in use, it is not correct practice to employ ropes of varying sizes on the same pulleys, as there is always a tendency on the part of the smaller ropes to run the pulley at a different speed, particularly where there is much difference between the diameters of the pulleys. This necessarily sets up friction and the smaller ropes will be found to have worn more quickly than the others.

There should always be plenty of clearance around a rope drive, not only to allow for a fair amount of sag but also for convenience in carrying out repairs. It is usual to assume that a horizontal drive under ordinary conditions will sag from 8 per cent. to 10 per cent. of the distance between the two shafts, and if the sag be greater than this, the ropes are running too slack.

Speed and Horsepower of Ropes

Most authorities agree that for main drives a speed ranging between 3600 and 5000 feet per minute is the best and most economical. It is quite possible to carry the speed much higher than this with safety, but, while some circumstances may justify it, as a general rule it is not good practice. Theoretically, the power it is possible to transmit with any given rope drive should increase in direct proportion to the speed at which it is run, but in practice this is by no means the case, as centrifugal force comes into play. This is demonstrated in Fig. 1. The straight line shows the power theoretically possible to transmit. The curve shows the manner in which the horsepower falls off above a certain critical speed between 4000 and 5000 feet per minute. In addition to this important consideration it is very necessary to remember that excessively high rope speed is inefficient in other directions. The life is materially shortened by the undue amount of bending which takes place, and air resistance to the movement of the ropes and pulleys becomes an appreciable factor. The horsepower which a rope will transmit with safety at ordinary speeds is very much in excess of that given in the average table, but it is not advisable greatly to exceed a load which gives a working strain of 200 pounds per square inch of sectional area as this means a correspondingly shortened life.

Out-of-line Drives

It happens occasionally that it is necessary to connect two shafts which are not strictly parallel. With ordinary pulleys there is an angle, depending to some extent upon speed, power, length of drive and relative diameters, at which the ropes will not keep in the grooves unless forcibly retained by means of jockey pulleys. This is usually put at anything over three degrees. It is unsafe, however, to place too great reliance upon a figure which may be a long way out under peculiar conditions. If such a drive cannot be avoided it is better practice to install pulleys having specially shaped grooves. A pulley groove which has been found very satisfactory is that shown in Fig. 2. While this groove retains the angular relation of 45 degrees between the two faces its center is several degrees out of the vertical, and the extent of this latter angle must depend upon the amount the two shafts are out of parallel. It is obvious that such an arrangement permits the rope to run on and off the pulleys without undue friction on one side and entirely obviates all risk of the rope mounting the groove.

Cross and Half-cross Drives

The cross drive is best avoided where possible, but if it is installed in the right way it will, in most cases, be found satisfactory. The usual method is to employ a multiple-rope drive working in pairs, between each of which it is desirable to leave an empty groove. The ropes of each pair are crossed respectively right and left handedly so as to make the two center ones the driving side. The result is that as both are moving in the same direction there is no friction between them. The two ropes forming the outer or sagging side, having little strain upon them, are easily forced aside by the inner pair, and friction between the two although appreciable is not important. If it be desired to use one rope lapped twice around the pulleys instead of two separate ones, the tight side should be arranged to run on the outside, the sagging ropes crossing each other in the center.

In the half-crossed drive where one pulley is at right angles with the other it should be arranged so that the slack side runs on the straight and the tension side on the skew. The former will not enter the groove if the conditions are reversed, whereas the tight side does so readily.

Right-angle Drive

The right-angle drive is the least desirable of any and should be avoided if possible. The loss of power is necessarily very considerable, and in all but the simplest form of right-angle drives difficulties are apt to arise with the guide pulleys. Fig. 3 illustrates a method which will obviously give little trouble as the grooves of all the pulleys are in line with one another. However, it is not always possible to employ such a simple arrangement, and something similar to the more complicated drive shown in Fig. 4 may be essential. A moment's consideration will show that under these conditions the grooves of the guide pulleys cannot all be in line with those of the driver and the driven; there is, therefore, a ten-

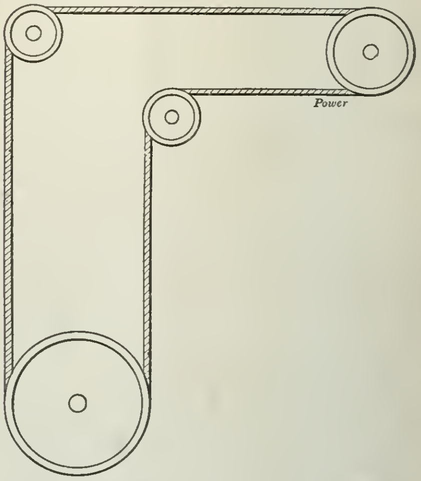


Fig. 3. Right-angle Drive with Parallel Shafts

dency, on the part of the ropes to leave the grooves, and most careful design is necessary to secure satisfactory running. The design of the guide or jockey pulley will materially affect the operation of such a drive. Speaking generally, the grooves should be well rounded at the bottom and wide, but in complicated drives, particularly those of the right-angle class, it is often necessary to employ a special form which can only be arrived at after mature consideration of the particular problem. Grooveless guides, similar to a cotton reel in shape, have been tried and are as a matter of fact still frequently employed, but experience has shown that on the whole they are not desirable although there are certain simple cases which may be negotiated by such means.

ROPES

There is a certain amount of divergency of opinion as to whether the three- or four-strand rope is the better. It is claimed for the latter that it has four points of contact with the pulley as against three in the former, and for this reason it is supposed to give a better grip. Advocates of the three-strand rope advance the theory that the triangular form fits itself better to the groove, and a good deal of the preference shown for it is due to the fact that unlike the four-strand rope it possesses no core. Where first-class ropes are concerned it is possible to overestimate the importance of this feature. There is no denying the fact that in a badly made four-strand rope the core is very apt to take more than its fair share of the load and to fail prematurely; but it is certainly not fair to assume that all makes are alike in this respect, and looking at the matter from an impartial point of view there is probably little to choose between the two types of rope.

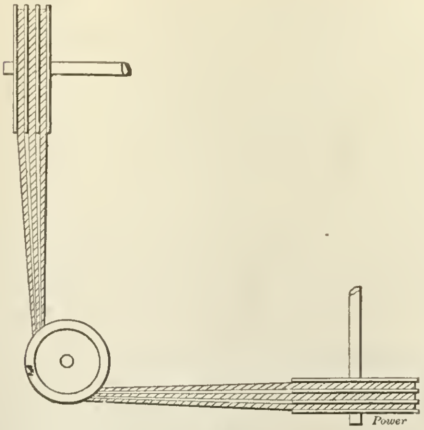


FIG. 4. RIGHT-ANGLE DRIVE WITH SHAFTS AT RIGHT ANGLES

With regard to material, there can be no question that the cotton rope is better than the hempen. Owing partially to lower internal friction the life may be put at twice that of the latter, and there is less loss of power in bending, less stretch, and less noise. The cost is higher, but as a general rule it is worth incurring.

It is perhaps not generally known that in some branches of the rope trade it is customary to treat the rope with black lead or oil adulterants with a view of increasing the weight. This is sometimes carried to the extent of an addition in weight of 25 per cent. or even more, and since ropes are sold by weight, it is well to be alert to detect the practice.

A Convenient Form of Prony Brake

The prony brake shown in the accompanying photograph is used in the testing laboratories of Columbia University and has been found convenient for testing high-speed engines of relatively small powers; that is, up to forty or fifty horsepower.

The brake is very simple in construction, being made up of a coil of rope, a set of chainfalls, some lengths of pipe, a wooden beam and a platform scales. The pulley used in this case is carried in a ball-bearing hanger and is also double flanged, the outer flange keeping the rope from running off and the inner flange serving to hold the cooling water. It would not be practicable to use this style of brake with a plain pulley, as the rope would run off.

The chainfalls are mounted directly over the center of the pulley and the distance between the holes *H*, in the beam, through which the rope passes, is equal to the diameter of the pulley. The beam is suspended from the chainfalls by a knife-edge *K*, which allows free movement of the beam. The tension on the rear end of the rope balances the weight of the brake arm, thus making the reading on the scales direct, and avoiding the necessity of subtracting the weight of the brake arm.

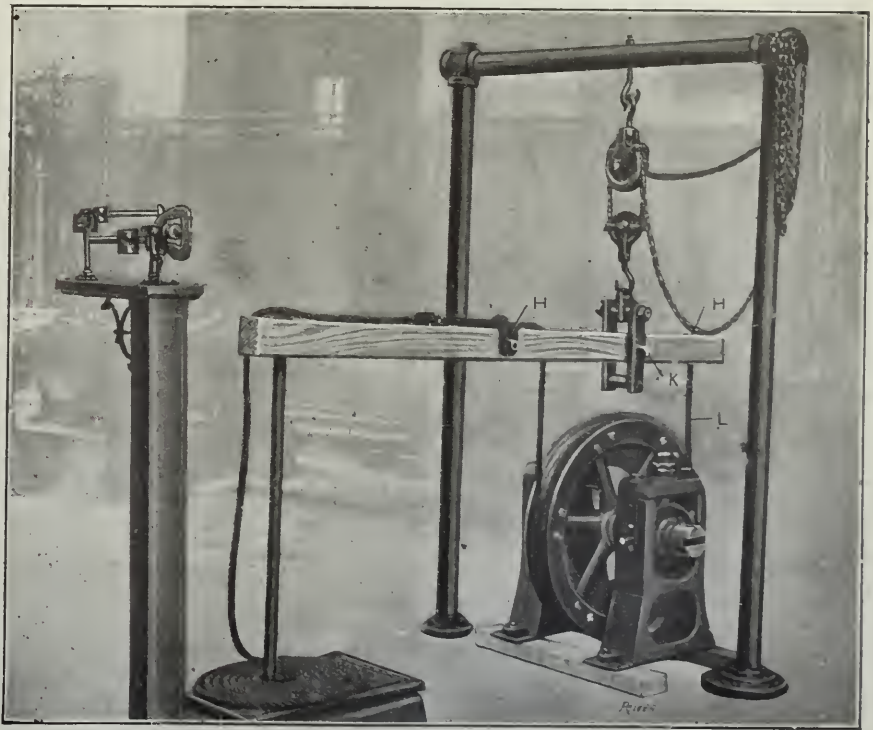
As the pulley is of the ball-bearing type, the pressure upon the bearings caused by taking up on the chain is negligible. After this tension has been adjusted approximately the scale beam may be kept floating by a slight tension applied to the rope through a piece of string, one end of which is tied to the rope at *L* and the other end of which is in the hands of the person making the test. The coupling at the end of the shaft serves to connect the apparatus with the engine to be tested.

The power is determined in the usual way, that is,

$$B.H.P. = \frac{2\pi \times W \times L \times N}{33{,}000}$$

where, $\pi = 3.1416$, W = scale reading, L = length of brake arm, in this case 4 feet, N = revolutions per minute. Hence, the formula may be expressed

$$B.H.P. = 0.0007616 \times W \times N$$



PRONY BRAKE RIGGED FOR TESTING

Measuring the Flow of Large Streams

By Franklin Van Winkle

Of any stream the rate of flow is determined by multiplying the cross-sectional area by the average velocity of the water. It is not a difficult matter to compute the area, but the determination of the average velocity of current passing a given cross-section requires care and accuracy. The author illustrates plainly how to obtain the area and also the velocity of flow by means of surface, subsurface and rod floats.

In considering the value of a natural stream as a source of power or water supply, the volumes and variation in the flow available will be questioned with greater or less exactness. In estimating such a value it is unwise to form more than general opinion from average run-off conditions based upon drainage area tributary to the stream, or upon the record of other streams in the same locality, though a knowledge of those conditions may be very desirable. Direct observations should be made of the stream under consideration. The flow should be determined on the best information obtainable, and not only as to quantity of water available under maximum and minimum conditions, but also as to duration and extent of fluctuations between these limits.

Although average and extreme low-water conditions generally control or limit the extent and usefulness of a plant, a knowledge of the volume of flood flow is of importance in arrangement and construction of dams, designing capacities of spillways and in making provision against floor damage to buildings and other works. These considerations of flow of a given stream are only determinable from actual tests.

It is not expected, however, that one should delay his decision as to character and extent of a development until a series of tests have actually been made covering every stage of flow worthy of his attention. Inspection of the banks, high- and low-water marks, investigation of the uses made of waters of the same stream above and below the site in question, and testimony of the younger as well as that of "the oldest inhabitants" of the locality, will serve as valuable checks on opinions formed from actual tests and estimates of flow made at different stages of stream flow.

In a previous article* it was stated that under most circumstances, the measurement of a quantity of water could be determined by ascertaining the velocity of its discharge through some kind of passageway or orifice whose dimensions were ascertainable. For most practical purposes, measurement of the flow of streams can be satisfactorily performed by direct determination of velocity and cross-sectional area of the natural channel, and for measurement of flow of large streams it is usually the case that no other methods are practicable.

As rate of flow consists of the product of a given cross-sectional area and the average of velocities of all different portions of the stream as it passes the given section, the value of a computation of flow depends directly upon the care and accuracy with which both of these factors have been obtained. Measurement of cross-sectional area may present no unusual difficulties, but determination of average velocity of current passing a given cross-section is difficult from the fact that the velocity of the stream at every point of the cross-section is continually changing.

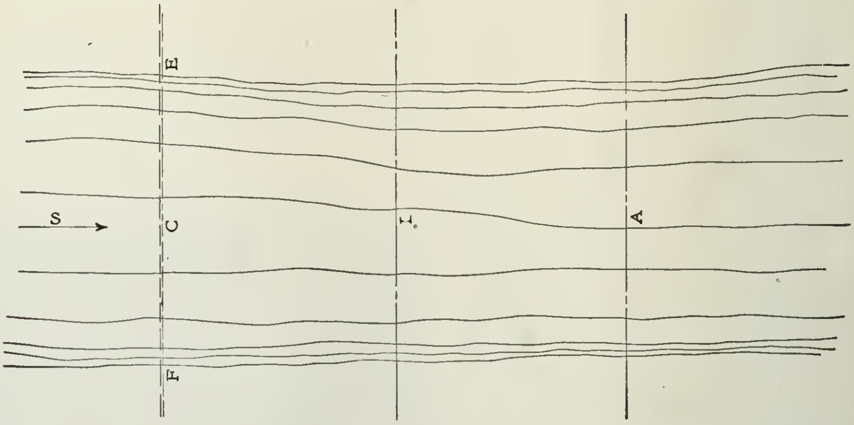


FIG. 1

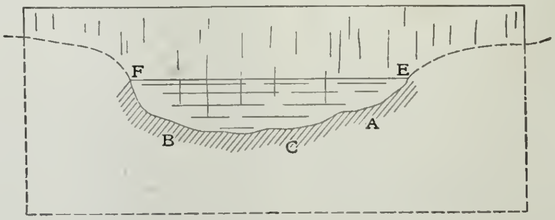


FIG. 2

Power, N.Y.

Erratic Motions of Flowing Water

The irregular rolling, surging and boiling motions of mountain streams, so clearly indicative of continuous changes in direction and velocity at every point of a cross-section, continue, in some degree, even after the waters reach a point down stream where the flow is smooth, noiseless and with velocity that is scarcely perceptible. The motions of flowing water are little less erratic than those of the wind. By releasing a small quantity of heavy oil, liquid dye or a handful of bran some distance below the surface of a quietly running stream of clear water, these irregularities of currents and velocities are made apparent by their drawing out the discolorations in a confusion of intertwining figures, and great irregularities of flow exist even when the rate or discharge of the stream may be practically constant. They seem to be largely dependent upon the alinement, length, shape and roughness of the channel, but are present to greater or less degree in streams flowing in all kinds of channels. Open-stream measurements should, therefore, be made with due regard to the existence of continuous fluctuations in direction and of velocity of currents, and methods should be employed which will result in elimination of errors compatible with the purposes of the gagings.

Methods of Measuring Stream Flow

When a stream is large, or for other

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*Power, August 25, 1908, page 313.

reasons employment of weir measurements would be impracticable, recourse may be had to other direct measurements of flow which, in many instances, are capable of being employed with as much exactness as the occasion will demand, and without being attended by the loss of time, trouble and expense incident to installation of a weir.

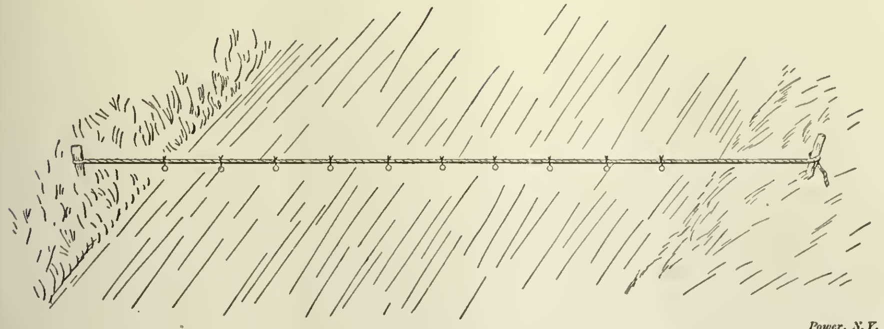

Power, N.Y.

FIG. 3

In considering the merits and adaptability of any method the fact should not be lost sight of that employment of natural cross-section of the stream is common to all; that the measurement of cross-section and of velocities constitutes the main factors in computation of stream flow, and of these two factors, accurate measurement of velocity is the more difficult to obtain. Any of the methods which may be employed require accurate determination of cross-sectional area of the stream at one or more places, and any measurements made for that purpose should be taken with great care. The means employed must be adapted to the size of stream and the facilities afforded for the work. The detail of processes employed by the engineer or surveyor may differ from those used by the mechanic, but each having the same object in view should arrive at the same result.

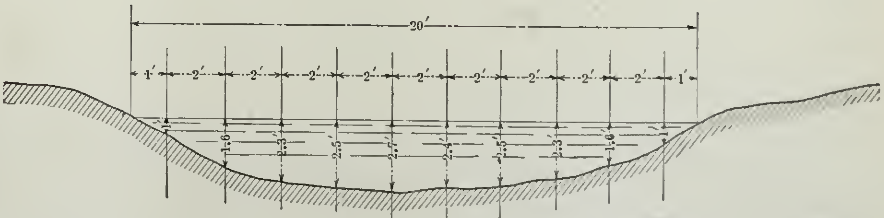
CROSS-SECTION OF THE STREAM

Ordinarily the cross-section is understood to be in a vertical plane at right angles with the thread or main direction of the stream as it passes the place where the section is taken. Supposing that cross-sections are to be taken of the stream at places *A*, *B* or *C*, Fig. 1, a representation of the cross-section taken at any point, as *C*, would present the same appearance as though the waters of the stream, without changing their hight, had suddenly come to rest, and then cutting the stream crosswise by insertion of a vertical plate of glass like a dam. The outline of water would present an appearance similar to Fig. 2 in which the section of the surface of the stream from *F* across to *E* in Fig. 1 would be represented by a horizontal straight line, like *FE* in Fig. 2, and the irregular shaded line *FBCAE*, Fig. 2, would represent the cross-section of the channel; and the whole figure which is contained between those lines, would be the cross-sectional area of the stream taken at the point *C*, Fig. 1.

The purpose of taking cross-sectional measurements is to determine the form and area of cross-section of the stream, for if this area, found in square feet, is multiplied by the average velocity in feet per second at which all parts of the stream flow through such a section, the product will be the whole volume of stream flow in cubic feet per second. Determination of the form and magnitude of a section will require measurement of breadth of the stream, and a number of measurements of its depth, taken at stated intervals, across the whole width at the place where the section is taken. The closer the intervals of depth measurements, the more accurately will the section be obtained, but for convenience in computation of the sectional area it will be best to take depth measurements at equal intervals.

OBTAINING THE CROSS-SECTIONAL AREA

For obtaining sectional areas of ordinary mill streams, good results are to be obtained by employment of very simple instruments. Depth measurements can be taken with any convenient form of measuring rod, taking care, however, that it is not pushed down into the bottom when taking the readings. To prevent the rod from penetrating the bottom, its lower end should be provided with a plate or disk several inches in diameter. A length of 3/4-inch iron pipe makes an excellent measuring rod if provided with a half-flange fitting screwed on the lower end, and for extensive measurements it will pay to lay off the rod in feet, with each alternate space, painted red and white like a surveyor's range pole. Intermediate divisions of inches or tenths of feet can also be laid off, or measurements of the fractional parts of feet can be measured as taken with a 2-foot rule. For measuring depths up to 3 or 4 feet, the writer prefers a newly planed rod of white pine, about 7/8x3 inches, on which to mark each depth and its number with a lead pencil on the side of the rod, subsequently measuring and recording results. By this method measurements are more quickly made, and the same person making the measurements may record them with much greater convenience than when continuously handling a rod in and out of water. For obtaining proper spacings of depth measurements across a stream, one end of a tape line can be secured to one bank while an assistant draws the tape to the other bank, and then by wading along the down-stream side of the tape line, or rowed in a boat by another assistant, measurements of depth can be taken at such intervals as may be desired. In the absence of means for holding a tape line in place, the intervals can be laid off with small tags spaced at equal distances along a stout cord or small rope when drawn taut, and this can be stretched across the stream suspended from trees or stakes, as shown in Fig. 3; but in any event exact measurement should be made of the whole width of the stream.



Power, N.Y.

FIG. 4

No matter what system of recording dimensions may be adopted, before leaving the site a good free-hand sketch should be made of the section with all dimensions inserted, as shown by Fig. 4. Paper with square rulings or ordinary section paper is best for this purpose, as by their aid free-hand sketches may be drawn very accurately to scale and checked before leaving the location. For most practical purposes, depth measurements taken at intervals of about 2 feet apart will be satisfactory, but in any case, to avoid confusion, the intervals should be equal. When any number of equally spaced depths have been taken, the area of the section is readily found by mutiplying the breadth by the average depth, but when the section of the bottom is very irregular, as shown in Fig. 5, it will be more accurate to make computation of the whole area from a careful platting of the section.

The cross-sections of large and rapid streams are obtained by soundings taken in the manner illustrated in Fig. 6. Supposing *A B* to be the line on which soundings are required, a base line *C D* is measured off along shore, preferably, though not necessarily, at right angles to *C D* and range poles are set at *C* and *B*. A boat is allowed to drop down stream from a point above *E* and when it comes in line with the range poles *C* and *B*, at a signal from a person stationed at *A* the lead is dropped and a person stationed at *D* observes the angle *C D E* with a compass, transit or theodolite. The sounding line may have a weight of about 14 pounds of lead, and if the boat drops down stream slowly, the weight may be allowed to hang near the bottom, so that there is no loss of time in taking the sounding when it arrives over the line *C B*. In place of using a theodolite or other angle-measuring instrument, the observer at *D* may be provided with a "plane table" and simple form of alidade, as shown in Fig. 7, with which to observe and lay off different angles. In extensive surveys of large rivers, observers with theodolites are stationed at both points *C* and *D*, the instrument at *C* being directed across the stream on the line *A B*, while that at *D* is kept trained on the boat. When the boat arrives on the line *A B*, the observer at *C* signals, the sounding line is dropped and the observer at *D* reads off the angle *C D E*. By setting up range poles or other conspicuous markers at points *A*, *C* and *D*, the use of an angle measurement made by a person stationed at *D* and another stationed on the line at *C* can be dispensed with by taking readings of angles subtended at *C E D* with a box sextant carried by a single observer in the boat.

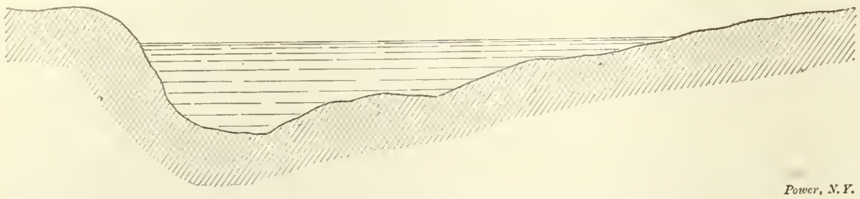


FIG. 5

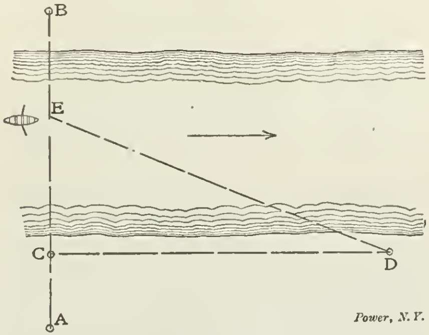


FIG. 6

By repeating observations taken at different distances from shore, the location of different soundings can be platted in their proper position, and the form of the river bed drawn by connecting the extremes of the lines from which the average section is readily determined.

MEASURING VELOCITY

As velocities at different points of a cross-section are variable, average velocity is only an imaginary one which if it existed in every point of the cross-section, the discharge would be the same as that resulting from the aggregation of filaments with uniform velocity. The least error in obtaining average velocity by any method may therefore be expected when gagings are made in channels which are straight, smooth and uniform in cross-section and the flow is free from swirls and eddies.

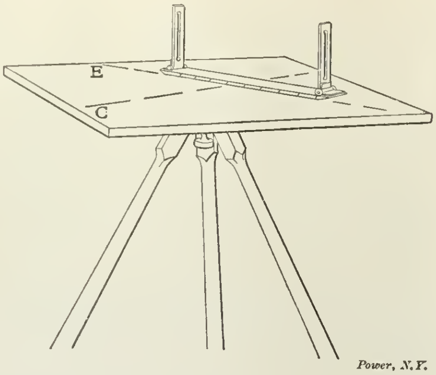


FIG. 7

Methods of measuring velocities may be divided into three general classes:

(1) By speed of floats passing over a measured length of channel.

(2) By instruments for measuring velocities of different points of a given section.

(3) By inference of velocity from temperature due motion; or measurement of inclination of the surface of a stream, combined with dimensions of its cross-section and assumed roughness of the channel.

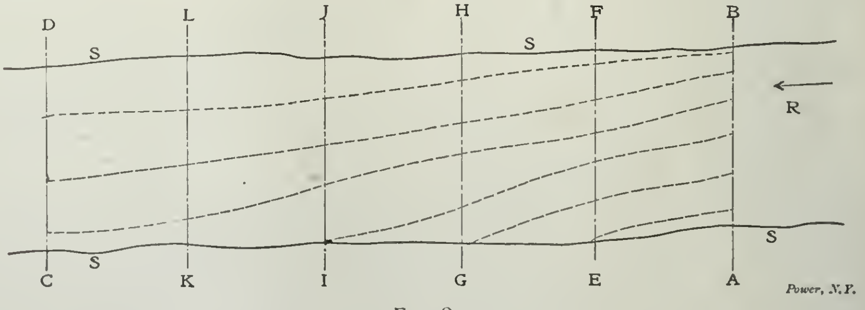


FIG. 8

Measurements made by floats are the simplest, requiring only the observation of the time it takes for a floating body to move over a measured course; while measurement of velocity by instruments requires that a constant for the instrument must first be found by experiment. Methods coming under the third head are almost wholly theoretical and will hereafter be referred to only for the sake of greater completeness in treating this subject. Under favorable circumstances, measurements by floats and by instruments can be made to yield results which for accuracy closely rival those obtained by weir measurements. Each has its advantages and disadvantages.

When measurement of flow is based on the velocity of floats passing over a measured distance, then it is necessary to arrive at an average value of cross-sectional area of the stream for the spaces passed over by the floats. For this purpose three or more equally spaced cross-sections must be obtained of the measured course, as at *E F*, *G H*, etc., Fig. 8; and for obtaining the flow in cubic feet per second, the average area of cross-section in square feet is to be multiplied by the average velocity of the stream in feet per second over the measured distance.

By the float method the mean velocity of only comparatively few particles of water is obtained for the time required for the float to move from one point of observation to the other, i.e., a float is acted upon only by the particles near it during the time of its passage over the measured distance, for it is carried along by a single impulse of the water and is of no service in indicating velocity due to succeeding impulses, which may be greater or less than the impulse which moves the float. On the other hand, instruments stationed in a given cross-section receive and measure the velocity of succeeding particles that strike against them during the observation. If there were no irregularities in velocities and particles followed one another past a given point with uniform velocity, then, in absence of wind or other disturbing elements tending to divert a float from its proper course, the float method of measurement would be all that could be desired. But since these irregularities of velocity exist in all moving waters it is believed that measurement of velocities of streams, wherever practicable, by employment of velocimeters and measurement of a single cross-section is preferable to float measurements of velocities combined with the average of several sections.

SURFACE FLOATS

Any small light bodies, such as small pieces of wood, wax or other material which will float and can be readily observed from the banks can be used as surface floats for measuring the surface velocity of a stream. Small dependence can be placed on floats, however, unless the course selected is over a fairly straight portion of channel, uniform in breadth and depth, and there is no disturbance from wind, eddies or other causes tending to separate the float from the same surouding particles of water in its passage from one end of the measured course to the other. The measured course should be 50 to 200 feet long. The floats should be dropped in the stream some considerable distance above the beginning of the measured distance so as to take up the velocity of the stream before traversing any portion of the measured distance.

In Fig. 8, *S S S S* representing the banks of a stream flowing in the direction indicated by the arrow *R*, suppose the distance *A C* to equal 200 feet, to be laid off on the left bank of the stream. If the stream is narrow enough some kind of lines bearing equal-spaced tags, Fig. 3, or equally divided rods, should be stretched from bank to bank at *A B* and *C D*, dividing the breadth at the beginning and end of the course into five or more equal spaces. The floats released at *R* should be paced at such distance from the banks as to cause the passage of one after another under different divisions of the line *A B*. An observer stationed at *A* should note the space under which each float crosses under the line *A B*, noting the time. To avoid confusion, a second float should not be released until the preceding one has passed under the line *C D*. The time required for passage and the space of *C D* passed under should both be noted. Intermediate lines *E F*, *G H*, *I J* and *K L*, equally spaced apart and corresponding with *A B* and *C D* in divisions of tags, will serve in obtaining a record of intermediate velocities and of the courses taken by the floats. It very commonly happens that many of the floats started under the beginning line *A B* are stranded at one side or the other of the channel before reaching the end of the measured course, and if their paths were to be platted the courses of the floats would be represented by the dotted lines shown in Figs. 8 and 9. In such cases, little more can be done than to take into consideration a shorter length of measured course, first taking into account only the time of passage of equally spaced paths of floats which have traversed the measured distance which is taken, and then, assuming that the average distance traveled by the others is only one-half as far in the same time, the time required for their passage over the whole measured distance may be regarded as twice the average time required for passage of those which do pass over the whole measured distance. Suppose, for instance, that the measured distance taken is *A C*, Fig. 9. The paths of floats 1*p*, 2*p* and 3*p* are terminated before reaching the section of the stream *C D*. If the average time required for passage of floats by the remaining paths 4*p*, 5*p*, etc., over the whole measured distance *A C* was 40 seconds, then in making up the whole average time, the time attributable to floats describing paths 1*p*, 2*p* and 3*p* should be assumed as 80 seconds.

Dividing the measured distance *A C* in feet by the average time in seconds thus found, gives the average surface velocity in feet per second. Having obtained the average surface velocity, the mean velocity of the stream must be determined from its relation to surface velocity. This is by no means uniform, nor can it be said that it is well established. Mean velocities are frequenty assumed to be about 80 per cent. of surface velocities, but careful experiments have shown that the relation between surface and mean velocity is only a chance one, depending on proportions of length, breadth, depth and shape of channel and is constantly varying. Inspection of Fig. 9 with respect to such paths as 1*p*, 2*p* and 3*p* suggests that the surface velocities along those paths have been arrested but must exert reactions on the mean velocity which are impossible of analysis.

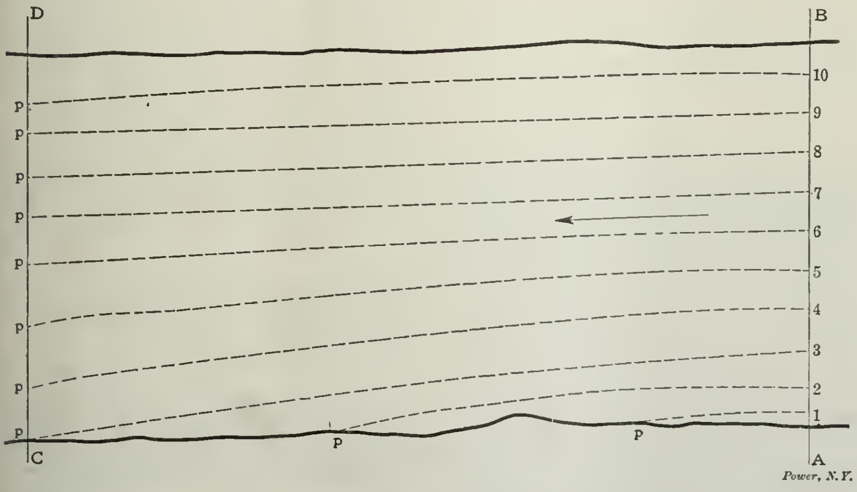


FIG. 9

In his "Lowell Hydraulic Experiments," Mr. Francis reports that surface floats of wax, about 2 inches in diameter, passing down the center of a long rectangular flume 10 feet wide by 8 feet deep moved about 6 per cent. slower than a tin tube 2 inches in diameter which extended a few inches above the surface and was loaded so as to stand nearly vertical, reaching within a few inches of the bottom of the flume. Also that the velocity of the tube was less than the mean velocity of the water in the flume. In another rectangular flume, which was 20 feet wide by 8 feet deep, he reports that the velocities of the tubes were about ½ per cent. greater than the average velocity of the entire body of water; and in a flume 29 feet wide by 8 feet deep the velocities of the float proved to be about 10 per cent. greater than that of the body of water. Charles Ellet, Jr., in tests of velocity of the Mississippi, concluded that the mean velocity was greater than the surface velocity by about 2 per cent., but in shallower streams he always found surface floats to travel faster than subsurface floats.

These examples are quoted to show the difficulty in assigning any particular relation between surface velocity and mean velocity, and although the surface float has frequently been relied upon in important gagings, its employment would appear to be misleading rather than useful.

SUBSURFACE FLOATS

The defects of surface floats in indicating only surface velocity are largely overcome by employment of submerged floats, though the latter cannot be employed in places where grass, weeds or other obstructions occupy portions of a channel where it is desired to obtain velocities. For measurement of flow in large open channels, subsurface floats have been used with excellent results. The same general methods are employed for obtaining the mean velocity as with surface floats, excepting that the indicated average velocity is assumed to be the mean velocity from the fact that the largest portion of the float is well below the surface, and the float as a whole is assumed to partake of the mean velocity of adjacent vertical films of water by which it is carried down stream.

The double float consists of a light surface float and a submerged float, the latter somewhat heavier than water and usually connected to the surface float by a cord or wire, the purpose of the upper float being to support enough weight of the lower float to keep the latter at a constant depth and indicate its position. The connecting cord can be lengthened at will so as to permit the lower float to be carried along by the velocity of the water at different depths.

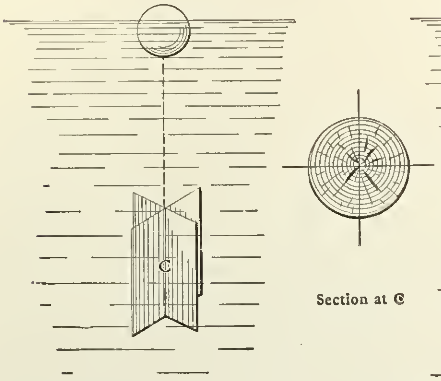


Fig. 10

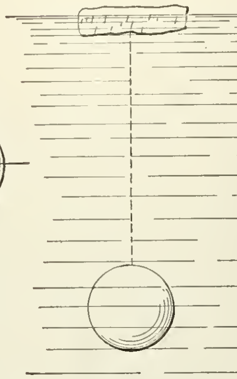


Fig. 11

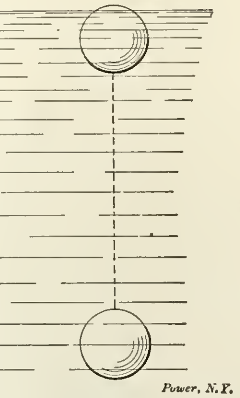


Fig. 12

Fig. 10 shows a form of subsurface float which has frequently been employed and recommends itself for economy of construction. It consists of two tin plates bent at right angles and joined together at the angle by soldering, and connected by a cord to a small wooden ball for a surface float. In place of soldering the parts of the lower float, they may, to good advantage, be secured to a piece of wood set in the angle as shown in the sectional view. Fig. 11 shows a form of submerged float consisting of a hollow metal ball connected to a piece of cork which acts as a surface float. Fig. 12 illustrates twin floats consisting of equal-sized hollow spheres, the upper one being a little lighter and the lower one a little heavier than water. According to Captain Cunningham, who made extensive gagings of natural streams for the British Government, the twin float gave better results than ordinary forms of subsurface floats, being less disturbed by wind and waves; but being almost completely submerged they are more difficult to locate.

Fig. 13 is an illustration of a subsurface float such as used by T. E. Ellis in the Connecticut river survey (1874). The submerged portion *A* consisted of a hollow annulus of tin plate 8½ inches in diameter by 8½ inches high and 7½ inches inside diameter, which, sealed at both ends, provided a ½-inch air space all around the cylindrical portion. Two brass wires were soldered across the lower end at right angles to each other, and to these a 28-ounce lead sinker was attached; two other wires were soldered at right angles to each other at the center, coming together near the upper end of the annulus, and to these the connecting cord was attached. The surface float *B* was a hollow ellipsoid made of tin plate, 6 inches in diameter and 1½ inches deep, with a cork in the top to hold a small flag and a small eye in the bottom for attaching the connecting cord.

Double floats of the same general description were used by Humphreys and Abbott in gaging the Mississippi river (1851), and the same method of gaging that river and its tributaries was used almost exclusively by engineers of the Federal Government from 1851 to 1881, but since the latter time the use of floats has been largely superseded by employment of improved forms of current meters.

Employment of double floats was found by Messrs. Humphreys and Abbott to be the best method available to them for obtaining reliable results and as their gagings were conducted with great care, brief description of their manner of using double floats, illustrated by Fig. 14, should be of interest. A suitable place was selected on the river, a base line *A B* 200 feet long was measured off on one bank parallel to the axis of the current, and sections *A A′* and *B B′* at each end of this base were marked out at right angles to it. The time consumed by each float in passing between these sections and the position of each float when it passed them were noted by four men, two at each end of the base, each party having a stop watch and a theodolite. At a signal of the engineer in charge, the floats were placed in the river from a boat about 100 feet above the upper section *A A′* and after passage over the measured course were picked up by a man in a boat some distance below the lower section.

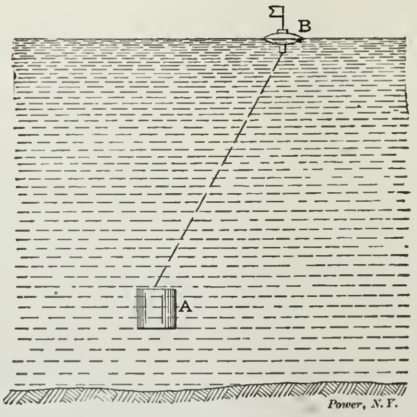


Fig. 13

At the instant a float passed the upper section a signal was given, the watches were started, and the angular position of the float was read with both theodolites. The instant the float passed the lower section a signal was again given, the watches were stopped, and the angular position was again read with both theodolites. These readings gave the distance of the float from the base line when it passed the sections and two independent measurements of the time consumed by it in passing between the sections. Soundings were taken at the end sections and at one or more intermediate sections, from which the area of the mean section was computed.

With this method base lines over 300 feet in length have been employed. In deep rivers, or in streams carrying weeds and grass, measurement of velocity by means of floats is about the only one which is practicable. It is impossible, however, to determine the position or depth of the lower float. Its position is approximately indicated by that of the upper float, but its relative position varies with the direction and velocity of the wind and the length of cord connecting the floats, and on account of side pressure on a long connecting cord combined with the vertical upward or downward action of currents on the lower float, it is difficult to determine the depth of the lower float with precision. The upper float may drag the lower one or *vice versa*. Employment of the double float is, however, very superior to that of the single surface float in obtaining mean velocity.

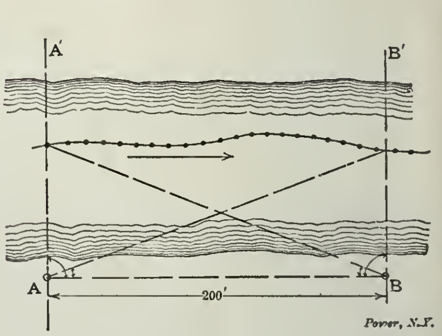


Fig. 14

Rod Floats

Rod floats have been employed for measuring mean velocity of flowing streams for many centuries, the earliest experiments recorded of their use for this purpose being those of Professor Cabeo, in 1646. They may be made of wood or thin metallic tubing, from 1 to 2 inches in diameter, weighted at their lower end so as to float nearly vertical. The lower end should nearly touch the bottom and the upper end should rise a few inches above the surface and may be provided with a small flag or tuft of cotton at the upper end so as to be readily located. One of the best forms of float rod consists, as shown in Fig. 14, of a wooden rod with a metal cap at the lower end in which to place different amounts of shot for sinking the rod to proper depth. The rod can be made in several sections of length to be screwed together according to depth of stream where it is employed. For measuring velocities of raceways 2 to 3 feet deep, an ordinary broom handle, loaded at one end, makes a very good rod float.

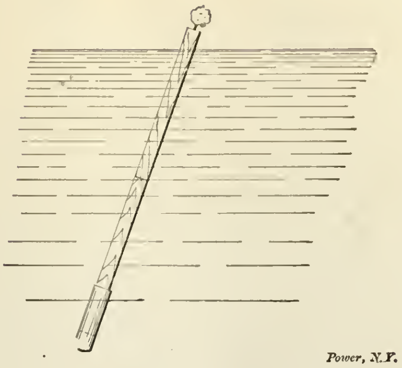


Fig. 15

Rod floats are to be employed for obtaining average velocity in the same general manner as that for using double floats. In obtaining the flow, a measured distance should be carefully measured off, the average cross-sectional area of the stream for the measured distance should be obtained with accuracy, and this combined with the average velocity of a number of floats released at equal distances across the breadth of the stream from bank to bank. For canals and raceways of moderate and uniform depth and for gaging streams containing floating grass and weeds, this is undoubtedly the best method that can be used. While float rods cannot be used for obtaining average velocity in very deep streams or in streams of rough or irregular bed, and may be slightly affected by wind, they interfere but very little with the natural motion of the water, they measure velocity direct, are not affected by silt and weeds, they measure forward velocity, are readily obtained and are inexpensive.

Quick Repair of Compressor Cylinder

By A. J. Dixon

A refrigerating apparatus of the absorption type, used to cool drinking water for a public building, was put out of service in the midst of a spell of torrid weather by an unusually severe leak in the piston-rod stuffing box on the ammonia cylinder of the pump which returned the liquid from the absorber to the generator. Inspection showed that the leak was caused by a crack extending longitudinally through the wall of the stuffing box on its under side, for about 8 inches from the end, as shown in Fig. 1.

Midway of the length of the stuffing box, which was about 14 inches overall, a cast-iron gland was placed between the rings of fibrous packing to entrap any leakage from the cylinder past the inner section of the packing and to return it through a duct on the under side of the stuffing box to the suction chamber. This suction duct was formed by drilling two holes at right angles through a lug cast on the barrel of the stuffing box, the holes being closed on the outside with ⅛-inch pipe plugs. These details are all shown in Fig. 2.

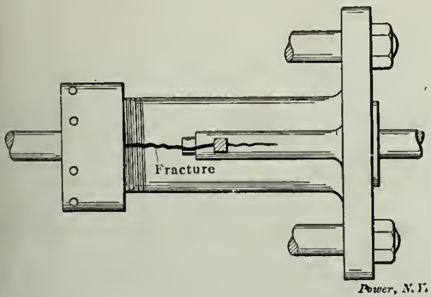


Fig. 1.

Regarding the origin of the fracture, the most plausible explanation seemed to be that in operating the pump the stuffing-box nut and gland had been drawn up so tightly as to compress the packing longitudinally and expand it radially, creating an excessive tensile stress in the wall of the stuffing box, thus starting a small crack which ended in complete rupture through the line of drilled holes, this being the weakest longitudinal section of the stuffing-box wall.

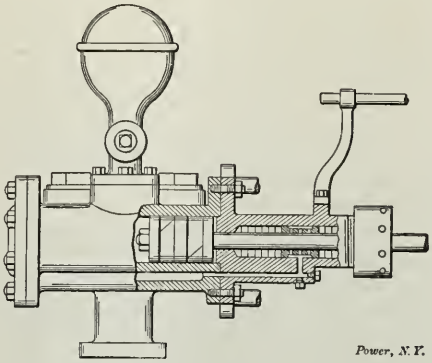


Fig. 2.

The substitution of a new cylinder head and stuffing box would have necessitated suspending cooling operations for several days, but this being intolerable, considering the weather, the engineer was constrained to devise an immediate repair. Several schemes were considered, but the plan finally adopted is shown in Fig. 3.

The cylinder head, with which the stuffing box formed a single casting, was first clamped to the faceplate of a lathe and the thread for receiving the packing nut was chased back to the boss which formed the support for the valve-stem bracket. An extra-heavy blind pipe flange, 10 inches in diameter, was then bored and threaded to a force fit upon the stuffing-box thread, and with a 12-foot lever and three men, whose weight aggregated 580 pounds, the flange was run onto the position indicated in the sketch. This accomplished, the pump was reassembled and again put to work under the usual discharge pressure of 150 pounds per square inch, without showing the semblance of a leak.

The condition of the fracture indicated that during its development, particles of the rod packing had been gradually squeezed into it, and these fibers serving as packing for the joint, assured an absolutely tight job when the edges of the fissure were drawn together under the pressure of the band formed by the pipe flange.

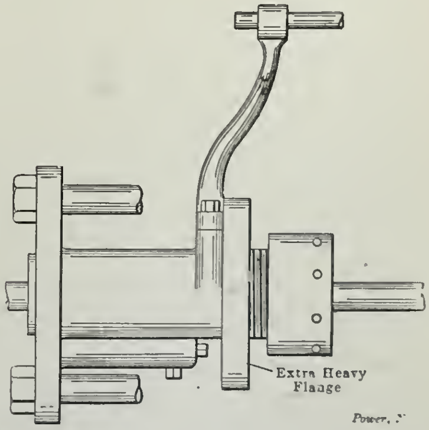


Fig. 3.

The Saving of Heat Units*

By H. L. Peiler

> *Small leaks, which individually appear insignificant, in the aggregate make a sum total at the end of the year that is surprising. Discharging condensation to the sewer is a leak of no small importance. All steam-using machinery should be amply and quickly drained, and the water of condensation kept under pressure and returned direct to the boilers without loss of its heat units.*

On going over a certain plant with the owner, I asked him where a particular return line led that originated in a dry room. He replied that it emptied into the drain. I asked how much water he was losing, and he stated that it could not be much and that the water was cheap. However, I managed to arouse his curiosity sufficiently to have the pipe uncovered and the water measured. To his surprise he found that a 50-gallon barrel was filled every 40 minutes and that the temperature of the water near the outlet of this large drier and while still under pressure was 306 degrees Fahrenheit. His engineer calculated for him the approximate cost, in tons of coal, of heating nearly 250,000 gallons of water from 50 to 300 degrees Fahrenheit, and this manufacturer came to the conclusion that he was not sufficiently wealthy to continue that system, when at relatively low cost the same water could be returned to the boilers direct and without loss of heat units. This plant had almost every practical device for making steam cheaply. High-class boilers, modern engines, approved equipment everywhere, including ball-bearing shafting and everything to reduce friction loss to a minimum, and yet this was only one of a series of "leaks" that were discovered during a careful investigation, and eventually he found that leaks, which cost him $1800 to stop, had for years previously been trickling $2600 annually into the drains. Everything possible had apparently been done to make the whole plant modern and uptodate, which is equivalent to saying "economical to operate," but the men in charge had neglected to "look into the furrows." This is only one of a series of similar incidents which I can recall in my own experience.

For purpose of comparison, it is necessary for me to state briefly the conditions that are most frequently met with in steam-using plants in this country. They are, of course, quite familiar to all engineers. The steam is trapped at the point of discharge at the apparatus in in which it is used. The water is then usually run by gravity to some low point, gathered in an open heater, vented receiver or hotwell, and after the make-up water has been added, the boiler is fed from this central reservoir. Where the steam is used under very low pressure (zero to five pounds) and where exhaust steam from the engines is mixed with this low-pressure live steam, the temperature of this water of condensation upon reaching the receiver is frequently under the boiling point and the addition of the cold make-up water further reduces it to a point where it can readily be handled by a good hot-water pump. Under these conditions, it is not uneconomical to handle the returns in this manner. But here I would like to point out that in drawing hot water from an open heater or receiver, and passing it through a pump, the temperature is still further reduced, and many engineers who have thermometers on the heater or receiver overlook the important fact that in most cases they lose from 8 to 12 degrees between the heater and the point where the feed water enters the boilers, and that this means a direct loss of from ⅔ to over 1 per cent. of the total fuel consumption.

However, there are very few plants that answer to this description. In nearly every industrial process in which steam is used for heating, drying or boiling, it is more economical to use a higher pressure in order to obtain the greater rapidity of result from the higher temperature Even in paper mills and the finishing departments of textile mills it is usual to find pressures of from five to ten pounds used in drying In laundries, in breweries, in confectionery boiling, in wool and cotton drying and many other lines of work, 60, 80 and even 100 pounds pressure are usually employed, and in all these cases the pump and receiver method is wasteful. It must be remembered that in a dry room, for instance, on which 80 pounds pressure is carried, the water of condensation as it leaves the outlet, and while still under pressure, has almost the same temperature as the steam from which it originates, that is to say about 320 degrees Fahrenheit. When this water reaches atmospheric pressure it at once cools by expansion to the boiling point and throws off 108 degrees of its temperature and a considerable portion of its own bulk in the form of expansion vapor, some of which is a dead loss. It is manifest to the most inexperienced new hand that if this water can be carried back direct to the boilers without being allowed to expand there will be an absolute saving of the amount of coal necessary to reheat this water by the 108 degrees that it has lost. When one recalls the fact that steam or hot water under even five pounds pressure has a temperature 16 degrees; at 10 pounds, 28 degrees; at 30 pounds, 62 degrees, and at 60 pounds, 95 degrees above boiling point at atmospheric pressure, and that between 10 and 12 degrees increase in the temperature of the feed water means a theoretical saving of 1 per cent. in the fuel account, it follows as a natural consequence that it is wrong in theory as well as in practice to cool off water and then spend good money in replacing a loss of heat that can readily be retained and used over again.

If the water can be returned into the boilers above the evaporation point, at atmospheric pressure, the gain is very much larger in proportion than if the boiler feed is below the boiling point. The reason for this is that the working capacity of the boiler has been changed, and has been brought to a state where the boiler is more efficient in heat units. This difference of efficiency must be also credited to this method of feeding, and the change will naturally show greater total results than if the saving had been computed from changes due to the difference in the temperature of the feed water alone. I will give one interesting case out of a number I have met in my own experience.

In the Stormont mill of the Canadian Colored Cotton Mills Company, at Cornwall, Ontario, when the average temperature of the boiler feed was 165 degrees, five 80-horsepower boilers were used to do the drying and dyeing. Under the rearrangement now in force about 80 per cent. of the feed water goes back direct to the boilers through return traps at 280 degrees, the make-up water at 140 degrees, making the average temperature of the boiler feed 252 degrees Fahrenheit. This shows a gain of 87 degrees with a theoretical saving of about 8 to 9 per cent. in fuel. In reality, two of the boilers have been shut down, the other three are more efficiently operated, and the actual saving in fuel is over 30 per cent., allowing for the fact that a portion of the dyeing plant has been removed to another mill. This was originally a receiver and pump layout with the make-up water running into the receiver tank to cool the returns, and nothing was done by me except to handle the returns independently from the make-up. The cost of making the change actually pays for itself in this plant every six months.

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*Abstract of paper read before the Canadian Society of Stationary Engineers, at Berlin, Can., July 27, 1910.

In many sections of Canada, and particularly throughout central and western Ontario, another consideration is that of the quality of the feed water. Every engineer realizes the value of using the condensation over again, and how much it improves his general conditions to minimize the amount of fresh water that is fed into the boilers, and most of which in many places has to be paid for. And yet, for some unknown reason, I find many plants where the engineer appears to have failed to convince his owners. Only last week I was in a factory in western Ontario where the feed water is so bad that the 2-inch feed pipes to the boilers are practically choked up completely every four weeks by the heavy deposit of solids from the water. In this same plant, the exhaust from two engines and four pumps is utilized for heating water for manufacturing purposes, but every drop of this valuable condensation is allowed to run into the sewers. Enough distilled water is wasted to feed one 125-horsepower boiler practically continuously.

If it were a matter of enormous expense or engineering difficulty to save and use this water, one could understand the objection. But it is such a simple problem, and as this water is always at or near the boiling point, the saving in fuel would also pay for the cost in a reasonable time. All that is necessary is an oil-separation system, and a trap or pump that will lift this water back to the feed system.

An important matter is that of the proper trapping of steam-using machinery of every description. It is a point that has been very much neglected by the engineering profession. The steam trap in general has been condemned by many engineers and is a cause of dissatisfaction in many factories and plants. An examination of the conditions will often show the reason. The chief reason, in my experience, is that steam traps are seldom purchased intelligently. This may appear to be a strong statement to make, but it is an absolutely true stricture, and one which I am able to substantiate.

The first great cause for this is that extremely few engineers take the trouble to find out what the actual quantity is in pounds or gallons of water that is condensed in any given process or machine. The ratio of water condensed from steam varies enormously according to the service performed by that steam. It is ordinarily calculated that one square foot of radiation used for heating an ordinary building, will condense 0.625 of a pound of water per hour, when the building has been heated to 68 degrees Fahrenheit. But this will vary according to the construction of the building, the amount of glass surface and the humidity of the air, as well as its changes in a given time due to wind. Condensation in a dry room will vary according to the moisture of the goods being dried. A mangle in a laundry will condense more steam, or less, accordingly as the quantity of the goods passing over it in a given time is greater or smaller. Indirect coils where air is forced over the pipes will show enormous differences when the air is drawn from out of doors and when the air is recirculated within the building. A vacuum pan will condense steam much more rapidly than a pan in which the same liquid is boiled at atmospheric pressure. Every process shows different results, and a proper test of the quantities almost invariably shows that the ordinary methods of guesswork run from 25 to 50 per cent. too low. It is quite usual to find a steam trap with an interior valve opening of ¼ inch, and actually able to discharge about 10 pounds of water per minute with five pounds pressure, placed where it has to take care of double that quantity of water, a thing that it is not possible to do. The trap floods, it refuses to do work that is against all natural laws, and in consequence the trap is blamed and not the man who attempted to force it to do impossibilities. A certain amount of blame also attaches to the engineer, because the chances are that he probably did not by actual personal investigation find out for himself the size of the opening through which the water had to pass. On the other hand, a certain amount of blame attaches to the makers of these traps, who fail to inform the engineering public of this most important detail in trap construction.

Ordinary so called catalog ratings are useless. They are usually based upon a fixed pressure of, say 50 pounds, and their drainage capacity in feet of 1-inch pipe is misleading. No allowance is made for varying conditions, and a trap that will handle a given quantity of water at 50 pounds will do nothing like that amount at 5 or 10 pounds pressure.

Every engineer knows that in order to obtain from steam the heat units which he wishes to utilize, he must first condense that steam. Every engineer knows that in order to obtain the best results, the apparatus must be drained quickly, and most engineers know that condensation does not usually leave the apparatus in a steady stream, but that the water nearly always comes in gulps and sudden gushes. It is this maximum condition that must be taken care of by the trap, and the most economical and useful trap is that which is a little larger than the actual average condition calls for.

With steam traps, as with every other kind of machinery, it is mistaken economy in every way to buy too small-sized units. The overloaded boiler and the overloaded engine are expected to give poor results in the long run, and yet I think I am not in the least guilty of exaggeration when I make the statement that fully 50 per cent. of all the steam traps in use in this country are overloaded and expected to do work of which they are not capable. This means another direct loss in the efficiency of the apparatus being drained, and in many cases reduces the output as much as one-half.

I wish to emphasize this important feature, and it covers the two great points to be aimed at in the economical handling of condensation, namely, that, first, all steam-using machinery should be amply and quickly drained, and, secondly, the water of condensation kept under pressure and returned direct to the boilers without loss of its heat units.

The method of doing this is simple. A proper system of traps will do the work automatically. The upkeep expense is practically none, and depreciation allowance less than that of any other part of an ordinary power plant. Return lines should be covered to retain the heat in the water; this is a practice which I always suggest and which the results justify.

It is usual in most steam-using plants to cover the supply pipe and leave the return pipes uncovered. You will tell me that the reason for covering the supply lines is to prevent excessive condensation in the line through radiation of heat through the pipe. My argument goes farther. Prevent that same radiation loss in the return. It represents just as much coal on the return side as on the supply side. If you can force the water into the boilers under pressure as hot as you get it, why allow any unnecessary loss at either end? Pipe covering only represents a small first cost; loss of heat by radiation represents a perpetual expense in coal.

By a proper trap I mean one of ample discharge capacity, and not a trap of the kind that requires a cooling of water to open its valve, such as the usual expansion type. Remember that the object is to utilize as fully as possible every heat unit contained in the water as it leaves the place of condensation. The rest is merely an application of ordinary common sense and reasoning power. Get the condensation back above the boiling point at atmospheric pressure, and you are making money.

It is difficult to imagine what argument can be set up against the economy of this method, and yet I find that many owners and engineers of plants where the old methods exist have the greatest prejudice against venturing on a change that is such a simple and reasonable one. I have made these changes in a great many plants, and in every case the owners have expressed their regret that the change had not been made sooner.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Storage Batteries in Isolated Plants

By Norman G. Meade

A system of battery-voltage regulation for use in installations of comparatively large size introduces the use of a motor-driven shunt-wound booster connected so that its voltage is added to that of the generator, "boosting" the voltage to the amount required for charging the battery. The charging current is controlled by the booster field rheostat and the voltage on discharge is controlled by means of end cells.

The booster is simply an ordinary generator having its field winding connected across the station busbars. The armature winding is connected in series with the battery and its function is to add to the line voltage sufficient "boosting" voltage to force current through the battery. If, for example, a 110-volt generator supplies a 110-volt circuit to which the battery discharges, 60 battery cells will be required, as explained. To bring these cells up to 2.65 volts per cell on charge requires a potential of 159 volts. The booster must furnish the difference between the maximum voltage and that of the generator, for this case 159 — 110 = 49 volts. The armature capacity must be sufficient to carry the maximum charging current for two hours without overheating. These factors fix the size of the machine. Usually the booster is motor driven. The kilowatt capacity of the motor can be less than that of the booster, because the booster never delivers its maximum current except at a low voltage and the overcharge current is usually half the normal eight-hour rate. If the kilowatt capacity of the motor be from 65 to 75 per cent. of the booster rating it will generally be amply powerful for its duty.

Fig. 7 shows an elementary diagram of a battery, with a charging booster, arranged to discharge through end cells. With the connections as shown the booster adds its voltage to that of the generator and the combined electromotive force is sufficient to send current through the battery. When the battery is to be discharged, the single-pole double-throw switch is thrown over so that the blade is in the upper jaw, connecting the battery directly across the line.

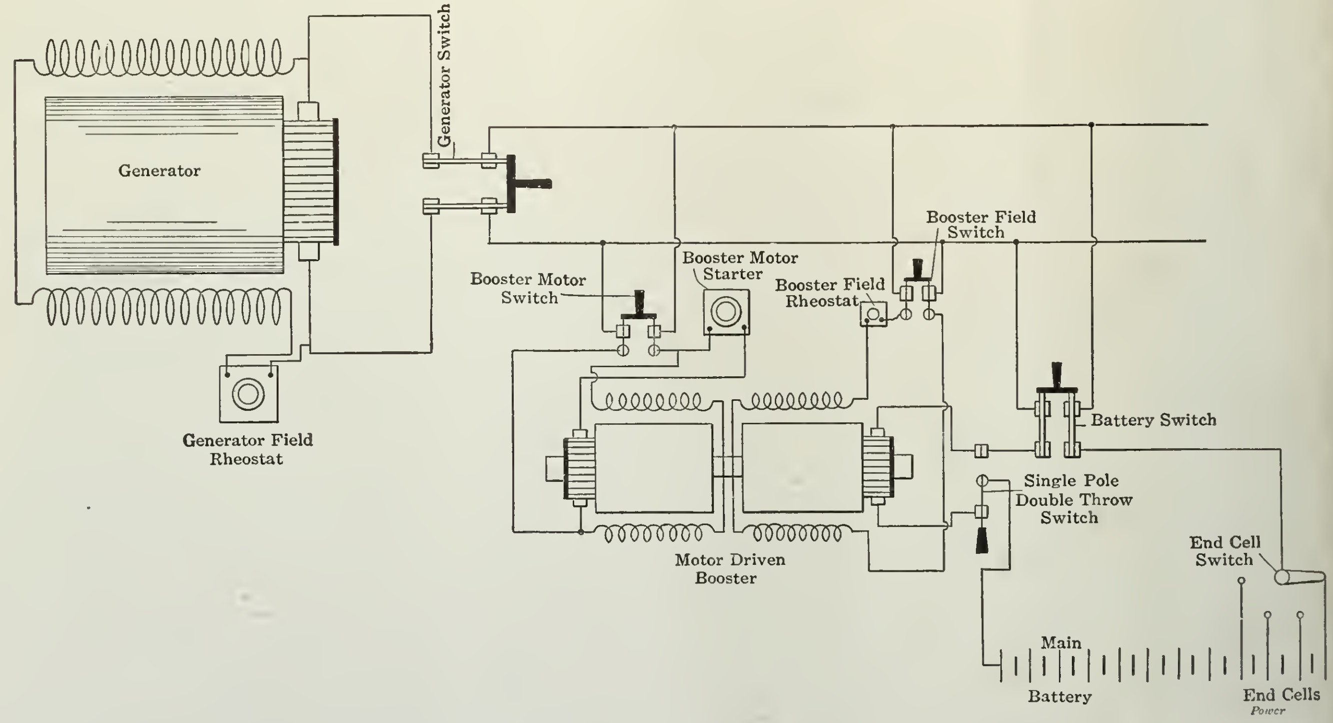

Fig. 7. Elementary Connections for Combined End-cell and Booster Regulation

The rate of charge is controlled by the booster-generator field rheostat which controls the booster voltage. This rheostat should have a sufficient number of steps and a high enough resistance to vary the booster voltage from a minimum of two volts to the maximum, by steps not exceeding three volts each. The field winding is always connected to an outside source of energy, never to the booster brushes. The connections of a switchboard adapted to this system is shown in Fig. 8. Three circuit-breakers are shown in the diagram, two overload breakers, one for the battery and one for the booster motor and one underload breaker to prevent reversal of current through the generator.

In selecting a system to be used attention should be directed to the following points: The equipment for charging in parallel and discharging in series, with rheostatic control, is the least expensive to install and is the one generally used where low first cost, simplicity of construction and ease of operation are of more importance than economy. The system using end cells and switches is adapted to plants of the same size as the first system, but is usually more expensive to install, except where the size of the generator is fixed by the charging rate of the battery, as is often the case. The battery being charged in series requires a generator of 155 volts, but the

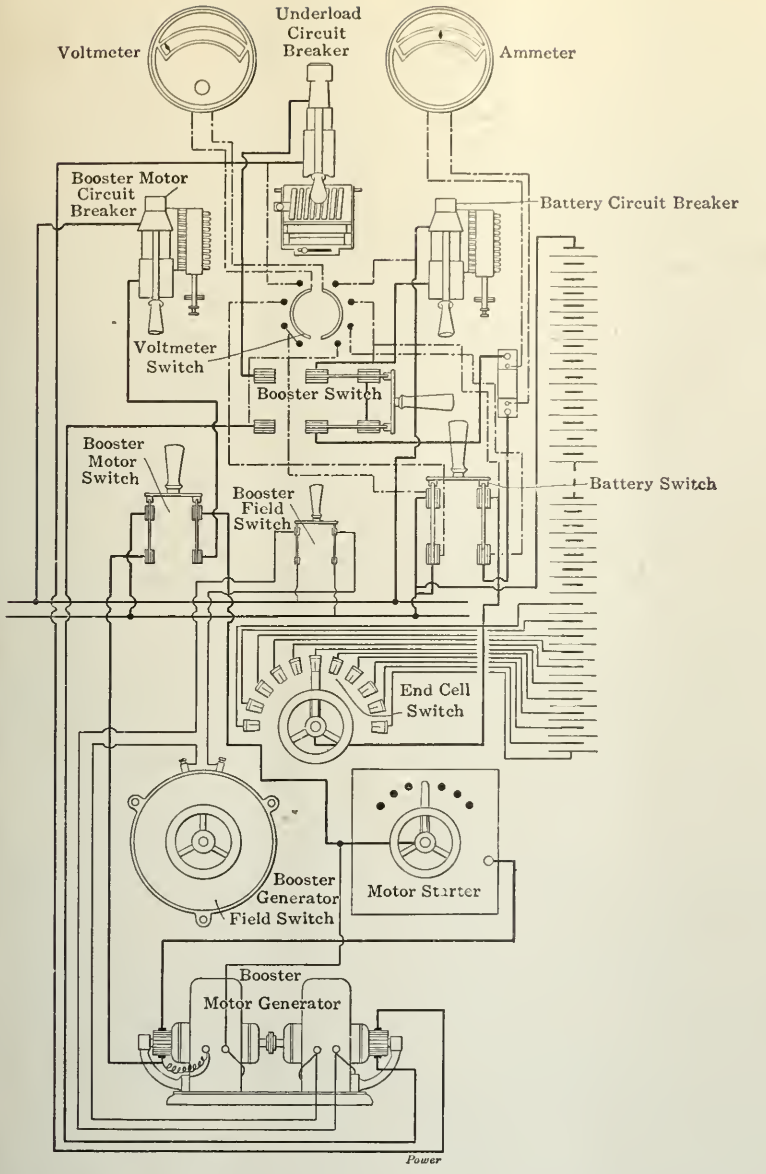

FIG. 8. SWITCHBOARD FOR COMBINED END-CELL AND BOOSTER REGULATION

generator ampere capacity required is one-half that required for the first system. The additional cost due to the end-cell switches must be taken into consideration when comparing the two systems.

The motor-driven booster system is the most satisfactory for plants of comparatively large size, where the generator has a current capacity considerably larger than the normal charging rate of the battery.

In selecting the size of battery to give a certain discharge, care must also be taken that the generator is large enough to charge the battery at a rate not lower than the normal eight-hour rate. In the first system described, where the two halves of the battery are charged in parallel, each half taking the normal rate, the generator must have a current capacity twice that of each half of the battery.

It is well to have the generator capacity sufficient to charge the battery occasionally at a rate somewhat in excess of the normal, as this not only keeps the cells in better condition but makes it possible to shorten the length of the charging period when conditions warrant this. With the first system, a high rate of charge can be obtained without excess generator capacity by charging each half of the battery separately, but the charging time under these conditions would be nearly twice as long as usual.

In isolated plants supplying a widely fluctuating power load, such as elevators, a nonreversible automatic booster can be used to advantage in connection with storage batteries. It causes the battery to charge and discharge as the external load increases or decreases. Fig. 9 shows the manner in which the booster is connected in circuit. The lighting load is connected across the generator mains as shown and the motor load to the mains beyond the booster and battery. The booster is excited by a compound field winding, in which the series and shunt

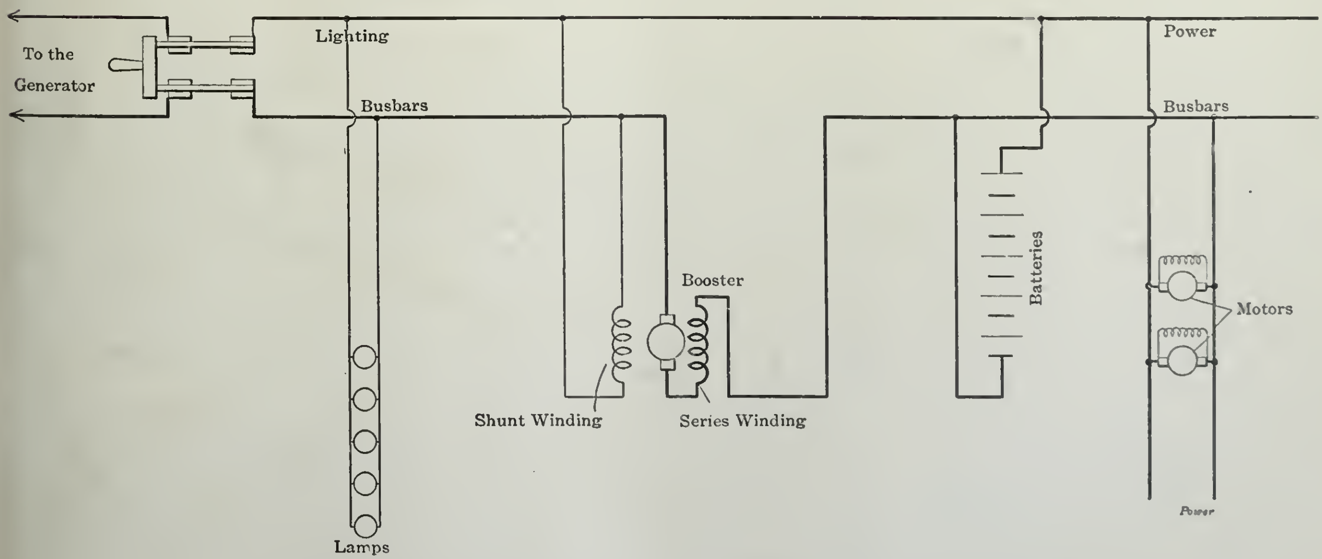

FIG. 9. ELEMENTARY CONNECTIONS FOR A "FLOATING" BATTERY AND BOOSTER

windings are connected in opposition, the series winding tending to produce a booster voltage opposing that of the generator while the shunt winding tends to produce an electromotive force assisting that of the generator. The shunt winding is stronger than the series winding; therefore the booster voltage is in the same direction as that of the generator. The voltage at the power busbars is therefore greater than that at the lighting busbars, the difference being about 15 volts, which is the usual normal electromotive force of the booster.

The battery electromotive force, when there is neither charge nor discharge current passing, is equal to the normal electromotive force of the mains across which it is connected. When the current used by the motor load is equal to the normal booster current, all current through the booster armature and series field winding goes to the load and the battery neither receives charge nor discharges. If the motor load decreases, the current through the booster armature and series winding will decrease, thereby reducing the magnetization of the latter, which strengthens the resultant booster field. This increases the voltage between the motor load busbars, thereby sending current to the battery. By properly proportioning the field windings, the current through the booster is reduced very slightly, the excess of current over that required for the load charging the battery. Conversely if the load should increase, a small increase in current through the booster increases the excitation of its series winding, which reduces the booster electromotive force. The voltage between the power busbars is thereby reduced and the battery therefore discharges, adding sufficient current to the booster circuit to make up for the amount required for the increase in load.

Speed Classification of Motors

The wide range of motor applications and the high degree of adaptability to different service conditions which characterize the electric motor have led to the development of a number of classes of motor, on the basis of speed characteristics. Besides the "old reliable" shunt-wound machine which runs at almost constant speed with any load, there are motors which change their speeds as much as 20 to 40 per cent. between no load and full load; others which can be run at any one of a few widely different speeds, maintaining practically constant speed at all loads, and still others that can be run at any one of a large number of speeds, each differing slightly from the one next in rate.

In view of this development, the Westinghouse Electric and Manufacturing Company has wisely adopted a standard classification of motors according to their speed characteristics, as follows:

Constant-speed motors, which do not vary greatly between no load and full load. These include synchronous motors; ordinary induction motors; shunt-wound direct-current motors and compound-wound machines which do not run more than 20 per cent. faster at no load than at full load.

Multispeed motors, meaning those that can be operated at any of several distinct speeds, each practically constant. These include induction motors in which the primary windings can be grouped so as to form different numbers of magnetic poles; and direct-current shunt-wound motors with two armature windings. The horsepower varies with the speed.

Adjustable-speed motors. This class is restricted, so far, to shunt-wound and moderately compounded direct-current machines of which the speed can be adjusted gradually over a considerable range by varying the field strength. Each speed is practically constant regardless of load, and the maximum horsepower is practically the same at all speeds.

Varying-speed motors, in which the speed varies almost in inverse proportion to the load, such as series-wound machines and those with compound windings heavy enough to cause more than 20 per cent. increase in speed from full load to no load.

Motors provided with heavy series field windings which are used only at starting and are cut out at full speed do not come within the last class. They are classed as shunt-wound machines because at normal speed they are of that type.

LETTERS

An Erratic Power Factor

An underground cable about three miles long transmitted 60-cycle alternating current to incandescent lamps which were supplied from transformers originally built for 125 cycles; these received current at 2200 volts and delivered it to the lamps at 110 volts. At very light loads the power factor of the primary circuit (the three-mile cable) was 85 per cent., at a little more than half load it went up to 100 per cent. and at full load it dropped to 90 per cent. This variation in the power factor was, of course, unusual and seemingly erratic. My explanation of it is that during light loads the inductance of the transformers was rather high, so that the current lagged considerably behind the electromotive force, tending to give a low power factor. This was partly neutralized by the capacity of the cable, so that the power factor was not as low as would be expected from the lagging current. As the load came on the transformers, the capacity of the cable remaining constant, or practically so, the inductive effect of the transformers was a very much smaller proportion than under light load, so that at a little above half load the power factor was unity, the inductance just balancing the capacity. With a further increase in load, the inductive effect still diminishing in its proportion, the capacity effect overbalanced the inductive effect and the power factor decreased until at full load it was approximately 90, the current then leading the electromotive force. That this condition of affairs was the cause of the trouble was satisfactorily proved to my mind by the fact that a number of the other circuits which were air lines, but practically the same otherwise, showed power factors under light loads considerably less than the cable circuit, and never approached unity very closely at full load. These circuits, however, went through exactly the same cycle of changes as the first cable circuit when they were changed to cable.

I should be pleased to have other readers give their opinions of this peculiar case.

HENRY D. JACKSON.

Boston, Mass.

[Mr. Jackson's explanation is evidently the only one that can be considered, unless there were modifying conditions which escaped his notice. The power factor of a transformer is very low at no load and small loads, and increases as the load increases, approaching 99 per cent. at full load. This is due to the increasing magnetic reaction between the primary and secondary windings, as the secondary current increases. At no load there is no secondary current and the transformer core and primary winding constitute a highly inductive reactance coil.—EDITOR.]

A Simple Polarity Indicator

I have read in the columns of POWER descriptions of several ways of finding the polarity of a direct-current dynamo, all of which are good. But there is a very simple method that I have used to advantage which seems to be unknown to other writers on the subject. Take a common water tumbler nearly full of water, connect a small copper wire to each of the dynamo terminals or to the main switch blades, and dip the other ends in the water, moving them slowly toward each other until small air bubbles form on the end of one of the wires; this will be the negative wire. This method applied to 110- or 125-volt machines has never failed with me.

R. T. WHITE.

Waterbury, Vt.

The lowest atmospheric temperature ever observed, —69 degrees Centigrade (—90.4 degrees Fahrenheit), was recorded on January 15, 1885, at Werchojansk in Eastern Siberia.

Gas Power Department

Tar Extractor for Bituminous Coal Gas

The accompanying illustrations show an apparatus devised by Prof. F. W. Burstall, of Birmingham (England) University, for extracting or separating suspended matter from gases, particularly tar from combustible gases. Fig. 1 is a longitudinal section, and Fig. 2 a transverse section of the apparatus suitable for extracting tar from combustible gas. It comprises a box *A* provided with a removable lid *B*, gas inlet and outlet *C C*, and a water-sealed bottom *D*, it being otherwise entirely a gas-tight structure, maintained also in this respect by the lid, and the inlet and outlet branches. In connection with the lid *B* are provided a number of water passages *E*, through which numerous small openings *F* communicate with the interior of the box in such manner that the water is jetted or sprayed into the interior of the apparatus.

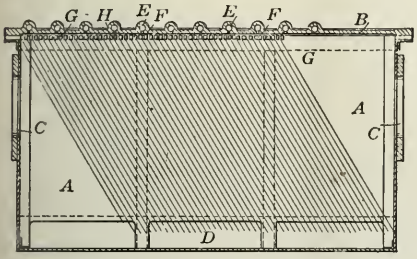

FIG. 1. TAR EXTRACTOR FOR GASES

A number of wires extend at small distances apart diagonally across the interior of the box, with their one ends *G* fixed to carrying bars *H* and their other ends free, and as all these wire lengths are between the inlet and the outlet, and extend almost the full depth of the interior of the box, the gas passing through is broken up and made to pass in quite a circuitous manner between the wires. Each of the wires is able to vibrate independently, and it is not intended that they should touch at any time. The water inlets are arranged at an angle to the wires.

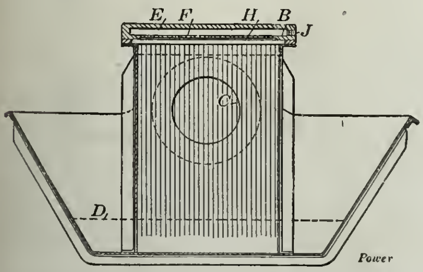

FIG. 2. TAR EXTRACTOR FOR GASES

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

In action it is believed that the wires vibrate within a dense water vapor or atmosphere provided by the liquid, and that the tar in suspension clings to the wires and gradually trickles downwardly off their free ends into the bottom of the box, whence it can be removed, the vibrating or quivering of the wires being effected by the combined actions of the gas passing through the box and the water spraying on to the wires. The water seal at the bottom of the box is sufficient to prevent any gas finding its way therethrough, and it is intended that the same water may be used over and over again by pumping the surplus of the seal into a pressure tank or pump supply connected up to the passages *E*, which passages have their inlets *J* alternately placed on opposite sides.

In addition to the extraction of tar, the gases treated may be cooled by the action of the apparatus.—*The Mechanical Engineer* (London).

Elementary Lectures on the Gas Producer*

By CECIL P. POOLE

Before going any farther in the study of what goes on in a gas generator, it is advisable for the student to get a general knowledge of the different fuels used in producers and a little more information about gases and their behavior under the influence of heat.

SOLID FUELS

As the reader probably knows, there are many grades of solid fuel, not counting wood. The first stage in the formation of coal is the decomposition of vegetable matter in marshes. This produces what is known as "peat." The next stage of evolution produces "lignite," which is a soft coal with a moderate proportion of carbon and a large proportion of moisture. Next comes bituminous coal, with still more carbon and less moisture; finally, there is anthracite, which is mostly carbon. Between "straight" bituminous and "straight" anthracite, however, there are several grades, and the difference is so great that these have been classified into semi-bituminous and semi-anthracite—highly inappropriate names, though fairly descriptive when one allows sufficient latitude in defining "semi."* Semi-anthracite is "near-anthracite" and semi-bituminous is "near-bituminous" coal.

All coals contain solid carbon and several gases, most of the latter being combustible. In addition to the carbon and gases, coal contains sulphur, moisture and a number of noncombustible solid substances, such as silica, aluminum and iron, which are lumped under the head of "ash." The gases contained in coal are hydrogen, oxygen, nitrogen and compound gases called "hydrocarbons" because they are composed of hydrogen and carbon.

Carbon is the most valuable constituent in any producer fuel, because when it is completely burned the product of combustion (carbon dioxide) does not carry away any heat energy with it, except the small amount used to raise its temperature. When hydrogen is burned, on the other hand, the hydrogen and oxygen combine to form water, vapor or steam, and this carries away with it the latent heat of evaporation in addition to the heat used in raising the temperature of the steam above 212 degrees. These questions will be discussed more fully in a future lecture on heat and combustion.

There are two kinds of analysis to which coals are commonly subjected; one is called "proximate" analysis and the other is called "ultimate" analysis. The proximate analysis gives the proportions of fixed carbon, volatile matter, moisture and ash. "Fixed" carbon is carbon not chemically combined with anything else, and "volatile matter" includes all gases except those parts of the hydrogen and oxygen which constitute moisture.

The ultimate analysis gives the proportions of carbon (including both the fixed carbon and that combined with other elements), hydrogen, nitrogen, sulphur and ash. There also remains some oxygen, a small portion of which is "free," the most of it forming eight-ninths of the moisture. It is impossible to determine directly the proportion of oxygen in coal, however, so the custom is to determine the proportions of the five constituents just mentioned and assume that what is left is oxygen. The proportions are determined by weight; that is, if a coal contains 75 per cent. carbon, each pound of coal contains three-fourths pound of carbon, and so on down the list of constituents.

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*This is the fourth lecture of the series.

*"Semi" means one-half, but is frequently used to mean "partly." Even with this definition, "semi-anthracite" does not really mean what it is intended to mean.

Table 1 gives some representative analyses of four kinds of coal and one of lignite. Referring to the proximate analyses, it will be found that the percentage of fixed carbon divided by the percentage of volatile matter is larger and larger, reading from left to right in the table. That is, F. C. ÷ V. M. for the Montana lignite is about 1.19, while the ratio for the bituminous coal is 2.15, that for the semi-bituminous is 3.58, that for the semi-anthracite is 10.2 and the ratio for anthracite is 14.9. This ratio of fixed carbon to volatile matter is generally considered the factor which determines the class in which a coal belongs. There is no sharp dividing line between the adjoining classes that is recognized by legal or other authority. It is customary, however, to call all coals bituminous in which the proportion of fixed carbon is less than three times that of volatile matter. Coal in which the fixed carbon is 12 times the volatile matter, or more, is classed as anthracite. The dividing line between semi-bituminous and semi-anthracite has never been agreed upon.

TABLE 1. DIFFERENT ANALYSES OF COAL.

Kind of Analysis.	Constituents.	Montana Lignite.	West Virginia.		Pennsylvania.	
			Bituminous.	Semi-Bituminous.	Semi-Anthracite.	Anthracite.
Proximate..	Fixed carbon..........	42.08	61.55	71.25	71.79	84.95
	Volatile matter........	35.90	28.58	19.88	7.02	5.70
	Moisture..............	11.05	1.48	3.94	5.41	3.45
	Ash..................	10.97	8.39	4.93	15.78	5.90
Ultimate...	Total carbon...........	59.08	77.82	79.78	72.65	86.45
	Total hydrogen........	5.37	4.89	4.60	3.10	2.38
	Nitrogen	1.33	1.48	1.01	0.77	0.75
	Sulphur..............	1.73	0.90	1.16	0.74	0.29
	Ash..................	10.97	8.39	4.93	15.78	5.90
	Oxygen, etc...........	21.52	6.52	8.52	6.96	4.22
Derived....	Total carbon..........	59.08	77.82	79.78	72.65	86.46
	Available hydrogen.....	4.14	4.73	4.16	2.50	1.62
	Sulphur..............	1.73	0.90	1.16	0.74	0.29
	Nitrogen.............	1.33	1.48	1.01	0.77	0.75
	Moisture.............	11.05	1.48	3.94	5.41	3.45
	Ash..................	10.97	8.39	4.93	15.78	5.90
	Unknown.............	11.70	5.20	5.02	2.15	1.53
Heat value; B.t.u. per pound = ...		10,539	13,718	14,382	12.047	13,470

Referring again to Table 1, it will be found that in addition to the proximate and ultimate analyses there is given a "derived" analysis; this is an analysis derived from the other two analyses, not made directly from the coal. This analysis differs from the ultimate analysis in three points: It gives the proportion of useful hydrogen instead of total hydrogen, the proportion of moisture and the proportion of actually unknown constituents instead of lumping them with oxygen.

The useful hydrogen is that which is not contained in the moisture; the latter is of no use as fuel, because just as much heat is used up in decomposing the moisture into hydrogen and oxygen as is liberated or made available by burning the hydrogen. Moreover, when the hydrogen is burned, the resulting steam is superheated far above the temperature of evaporation (212 degrees at atmospheric pressure), and this absorbs still more heat. The proportions of carbon and useful hydrogen, therefore, are much better indications of the true fuel value of the coal than either the proportions of carbon and total hydrogen or those of fixed carbon and volatile matter.

Take, for example, the analyses of the Montana lignite. The proximate analysis shows 11.05 per cent. of moisture, or water, and the ultimate analysis shows 5.37 per cent. of hydrogen. Of the moisture, one-ninth is necessarily hydrogen, and this one-ninth is

$$11.05 \div 9 = 1.23$$

per cent. of the weight of the coal. Subtracting this from the percentage of total hydrogen gives

$$5.37 - 1.23 = 4.14$$

per cent. of hydrogen that is available as fuel. The total proportion of actual fuel material, therefore, is 59.08 carbon + 4.14 hydrogen + 1.73 sulphur = 64.95 per cent. The proximate analysis, however, shows 77.98 per cent. and the ultimate analysis 65.78 per cent. of combustible material, the former being very much, and the latter slightly, greater than the actual combustible material, if the analyses were accurately made.

The derived analysis is clearer than the others also in showing the true proportion of unknown substances in the coal; in the proximate analysis these are included in the volatile matter and in the ultimate analysis they are included with the oxygen.

Upon comparing the analyses it will be evident that the "volatile matter" content commonly given in proximate analyses includes sulphur. This does not seem a rational plan, because the sulphur that is commonly known is not volatile. But the sulphur contained in coal *is* mostly in volatile form, supposedly as sulphide of iron. However, since the exact character of the sulphur in coal is not positively known, it is more nearly accurate to state the sulphur content separately, although it is so small that it is not of much importance, and it is practically impossible to analyze coal with exactness.

Coke and Charcoal

Besides the coals just discussed, there are two solid fuels known as "charcoal" and "coke." These are not strictly additional fuels, because they are made from wood and coal by distilling out the volatile constituents of those fuels and leaving the solid carbon and the ash (mineral impurities). Charcoal is made from wood and coke is made from bituminous coal. The process consists, broadly, of heating the wood or coal moderately—not hot enough to set it afire—and maintaining the heat until practically all of the gaseous contents are driven out of it. The temperature at which the wood is charred is somewhere between 400 and 600 degrees Fahrenheit. The temperature to which coal is heated to drive off its volatiles is around 1600 degrees Fahrenheit. The higher the temperature of charring or coking, the less charcoal or coke will be obtained from the wood or coal, within certain limits. Slow distillation (moderate temperature) produces the largest percentage of charcoal or coke.

The weight of charcoal obtained from a pound of wood varies from one-fifth to one-third pound ordinarily, and the weight of coke obtained from coal ranges from about one-third to seven-eighths of the original weight of coal.

Since the ash is all left in the charcoal and coke, and the weight of material is very much reduced, the percentage of ash in charcoal and coke is very high, compared with that in the original fuel. Charcoal and coke, when first made, contain no moisture, but both of them absorb moisture greedily. Consequently, when they reach the market they are several per cent. heavier than when first taken from the charcoal pile or the coke oven, because of the moisture absorbed in the meantime.

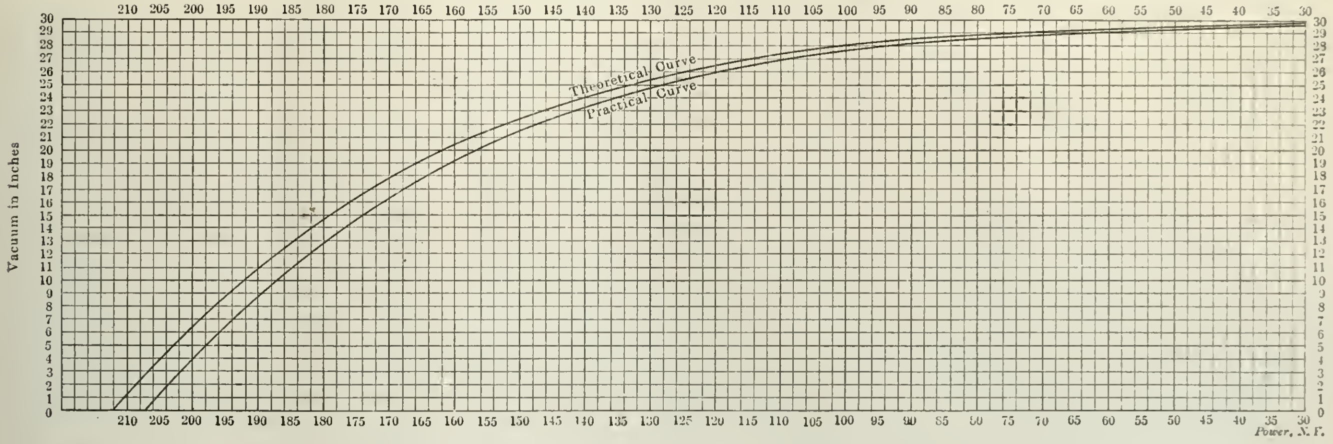
Neither charcoal nor coke is used as a fuel for gas producers, for the reason that they are much more expensive than the fuels from which they are made, and these fuels can be used in producers. For this reason it is not worth while to discuss the composition and fuel characteristics of charcoal and coke in this series of lectures. The general discussion of solid fuels, however, would be incomplete without any reference to charcoal and coke.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Condenser Economy

In the economical operation of condensers it is very important for the engineer to know the proper temperatures of the injection and discharge waters which will give him the best vacuum and the most economical results. The accompanying chart will prove a valuable aid in the economical running of a condenser. The temperature of the injection water is usually beyond the control of the engineer and varies from summer to winter temperatures, making it necessary for him to supply more or less water to the condenser to maintain the same vacuum. The temperature of the discharge water must therefore be the guide of the engineer in determining whether he is getting all the vacuum possible, or whether he is pumping too much or too little injection water.



Temperature, Fahrenheit, of Discharge Water from Condenser

By referring to the chart it will be seen that with 120-degree discharge water, 26½ inches is the greatest vacuum attainable. With 100-degree discharge water 27¾ inches of vacuum is attainable. The relation between the discharge temperature and the attainable vacuum is due to the boiling point of water changing under the various vacua.

If your condenser shows a discharge water temperature of 120 degrees or less and you are not obtaining close to the vacuum attainable as shown on the chart, then you have leaks or uneconomical conditions that should be investigated.

Again, if 25 inches of vacuum is all that is attainable with your plant, due to construction features, and your discharge water shows a temperature of less than 128 degrees, then it would be possible to use less injection water and still maintain the same vacuum, thus effecting a saving proportional to the decrease in speed of the pumps and in the amount of steam thus saved.

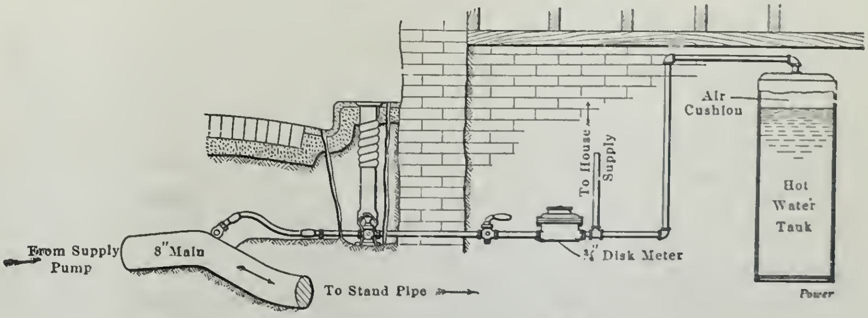
This chart, kept in an accessible place and referred to occasionally, will tend to increase the efficiency of your plant.

N. L. Daney.

Elyria, Ohio.

A Creeping Water Meter

It is not uncommon for an electric meter to "creep" with the customer actually consuming no power, but for a water meter to perform this singular feature is almost unheard of. However, here is the case of a water meter correctly geared and registering 40 gallons per hour without a gallon being used or wasted.



Piping Showing Location of Meter

This meter was of the disk type and had been in service for over two years, never having registered over 2000 gallons per month. The discrepancy was at once noticed by the auditing department and a man was sent to check the reading. This he did, then read it again in two days and found it to be averaging about 1000 gallons per day. He also reported that the meter was observed to be running. The natural conclusion was a leak, so the bill was sent in. The customer immediately made a strenuous complaint, stating that he had not used more water than during any month previous.

The piping was carefully gone over and no leak being discovered the conclusion was reached that the waste had been stopped. On going back to the meter, however, it was found to be still running. On closing the customer's cutoff valve the meter stopped and on opening it immediately started up. The meter was taken out and tested but was found to be correct. The next idea, however, solved the problem—that of inserting a check valve on the pipe line between the cutoff valve and the meter. The explanation of the trouble is as follows:

The meter was connected as shown in the accompanying sketch. The main, passing over a hill, was bent as exaggerated in the drawing and the tap was made at about the highest point of the bend. A four-cylinder pump geared direct to an induction motor pumped water to the standpipe through this main. One of the pistons was put out of commission about a month before and this probably produced a surging in the main from the standpipe to the pump. An air cushion was thus formed at the top of the hot-water tank. This surging compressed and released this air cushion at every part of the revolution of the pump in which this broken piston passed and caused the water to pass back and forth through the meter. The disk-type meter will not run backward although it will admit water through it in the opposite direction. This, of course, caused the meter to register constantly even though no water was consumed.

A. E. Holman.

Anderson, S. C.

Air Leakage through Cast Iron

Some time ago the writer made a few tests on some cast-iron pipe to determine if air leaked through cast iron directly; that is, through the grain of the iron itself. The tests were simple ones, and the results obtained are of a comparative nature.

Two pipes were used in the experiment; the one, designated as the small *pipe* on the curves, was about 9 inches inside diameter by 6 feet long and the other, referred to as the *large pipe,* was about 14 inches inside diameter and 6 feet in length. The pipes were covered at the ends with blank flanges fitted with ⅛-in. sheet packing. A mercury column was connected to one end of the pipe, the other end being connected to a three-throw high-vacuum Packard pump. The tests were carried on under a high vacuum, and only the rate of leakage was noted in the results, as no attempt was made to measure the quantity of air.

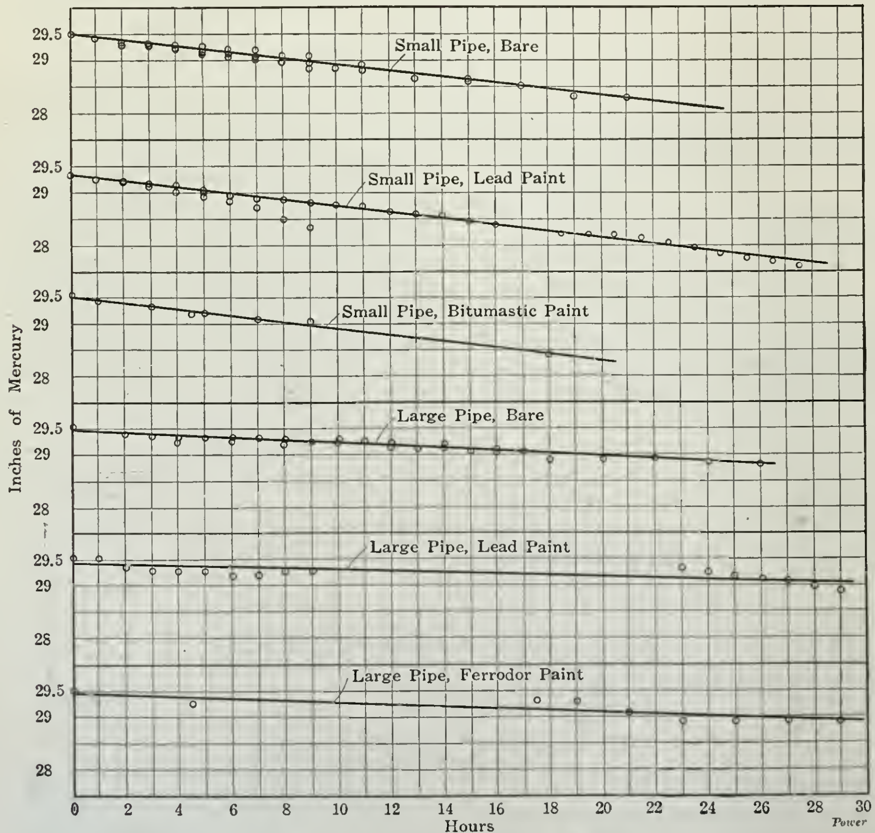
The experiment was performed in the following manner: The vacuum being put on the pipes, all connections to the pump were closed and readings were taken, first at hourly intervals, and then at longer periods, since the drop in vacuum was very small. These readings were corrected by the barometer; that is, the difference between the actual barometer reading and 30 inches was added to the reading on the mercury column at the end of the pipe. These corrected readings were plotted against time, and the slope of the curves represents the rate of leakage. All efforts were made to prevent any leakage at the joints at the ends by keeping these parts well covered with a heavy paint, but still an uncertainty here existed. First, the pipes were tried bare, then they were covered with a lead paint. This paint was removed with lye and hot water, and a special paint known as Bitumastic paint was used on the small pipe, and Ferrodor paint on the large pipe.

It can be seen from the curves that a very small amount of drop was experienced in the tests, some of them running 30 hours. Also for the same pipe the curves are nearly all parallel, which indicates that the rate of leakage through all the different paints and on bare pipe was the same. This tends to show that no air actually leaked through the cast iron, but this statement may be taken as of comparative value only. However, it seems that the greater leakage in pipe lines occurs through the joints and around the prime mover. Compared to the above leakage, the leakage through the pipe itself is probably negligible.

The experiment also shows that joints may be made tight if sufficient care is taken, as each pipe had two joints through which leakage might occur.

G. A. Glick.

Joliet, Ill.



Leakage for Various Periods of Time

Faulty Design and Workmanship

Here in New England, many boilers have been installed with an outside manhole fitted as shown in Fig. 1, having a flange with a raised surface *C,* 1¼ inches wide, extending around the main opening. This is evidently done to save machine work, and the usual method of packing is to use an ordinary molded gasket about 3/16 inch thick. When the bolts are screwed up fairly tight, the flanges outside of the bolt circle are about 5/16 inch apart and further tightening of the bolts is apt to place excessive stress on the flanges.

One of our boilers was fitted in this manner and had been out of service for washing and general fixing up. Before leaving in the evening I left word to cut out one of the other boilers and cut in the clean one after the peak load had passed. Returning to the station next morning I found a slow fire under the boiler and steam just beginning to show. The fireman said he had tightened the bolts on the manhole cover but was afraid that something was wrong with it, so I immediately went on top of the boiler and found the main casting broken as shown at *A.*

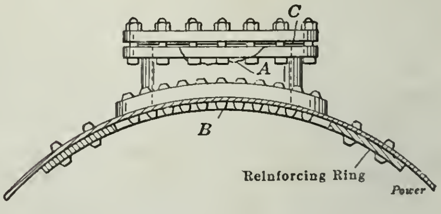


Fig. 1. Broken Manhole Casting

Upon taking off the cover and striking the edge of the flange with a hammer the entire flange was easily knocked off and in a number of places there was evidence that the grain of the iron had been disturbed to a depth of ½ inch.

The manhole was 12x16 inches, or equal to an area of 150 square inches, which at 125 pounds gage pressure equals 18,750 pounds tending to force the manhole cover off. The shape of the casting is such that the tail *B* of the rivets had to be on the outside, they being driven from the inside before the tubes or horizontal stays were put in.

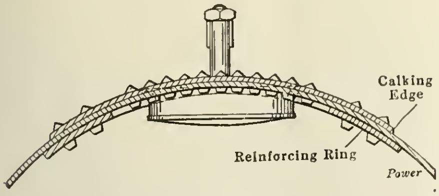


FIG. 2. INSIDE MANHOLE COVER

The photograph of the four rivets marked *X* shows a fair average of the thickness of the rivet heads as compared with that of a standard rivet. Twenty-nine of these held the casting to the boiler shell. The thin rivet heads were inside the boiler and, although much out of proportion to the body of the ⅞-inch rivets, were allowed to pass, but would not have been permissible on the outside of the shell. This proves the truth of the statement in a recent issue of POWER: "The careful inspection of steam boilers when they are in process of construction is quite as important as periodic inspection when they are in operation."

After keeping the boiler out of commission about three months, repairs were made, as shown in Fig. 2. A steel manhole form was fitted to the outside of the boiler and drilled to match the old rivet holes, also new holes were drilled through the new form, boiler shell and reinforcing ring, the latter being left in the boiler because it was too large to pass through the manhole. The calking had to be around the outer edge of the new form instead of the inside edge of the manhole, as formerly. This method of repairing permitted all of the rivets to be driven from the outside and it made a good job.

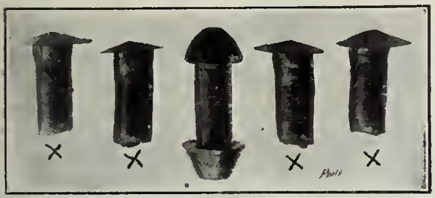


FIG. 3. RIVETS WITH IMPROPERLY FORMED HEADS

It was fortunate that the casting broke at such an opportune time, for if the joint had begun to leak when the boiler was under full pressure and someone had attempted to tighten the bolts, the result would have been disastrous.

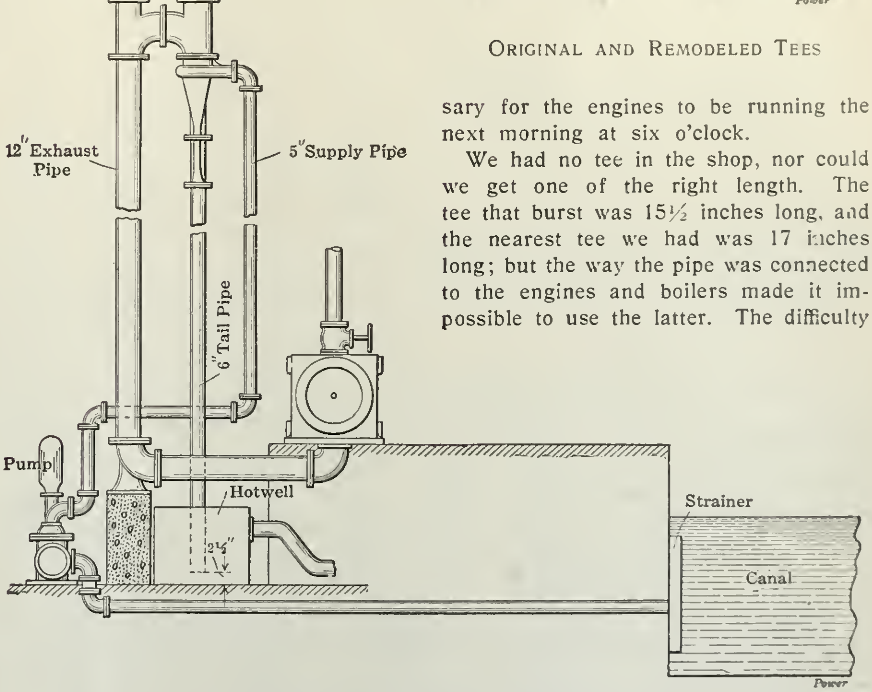
J. W. PARKER.

Clinton, Mass.

Trouble with a Barometric Condenser

We are experiencing considerable trouble with a Bulkley condenser installed as shown in the accompanying sketch. The distance from the overflow of the hotwell to the exhaust nozzle is 34 feet and about 1 foot more to the exhaust pipe. The canal gives a head of about 4 feet to the pump.

Frequently when building up the vacuum, preparatory to starting the engine, such large quantities of water come over from the condenser into the exhaust pipe that it is impossible to start the engine without danger of wrecking it. Sometimes the water appears just as the engine starts to turn over, in which case we open the drip at the bottom of the exhaust pipe to break the vacuum, and large quantities of water are forced through the relief valve. We have no trouble after the engine gets up to speed, nor when shutting down. After the engine has stopped it seems as though the pump could be kept running indefinitely without any water coming over. When the water does come over upon starting up, the level of the water in the hotwell drops very low.



ARRANGEMENT OF CONDENSING APPARATUS

We tested the condenser at 50 pounds for leaks and found only two small ones. These were repaired but it made no difference. Our vacuum holds quite steady at from 24 to 26 inches. An expert from the condenser manufacturers was sent to investigate the trouble but at the time he was here the condenser worked perfectly.

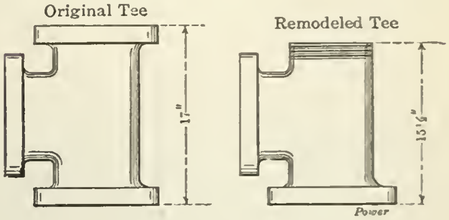
The engine is a simple 26x60-inch Corliss and has been in operation for about 20 years without giving any previous trouble. The pump seems to be in good condition and there is no exhaust-steam heater on the exhaust line. Perhaps some reader of POWER can solve the mystery.

W. P. ELSTON.

Taunton, Mass.

Remodeling a Tee

One afternoon an 8-inch tee on the main steam line feeding our two engines, burst and it was absolutely necessary for the engines to be running the next morning at six o'clock.



ORIGINAL AND REMODELED TEES

We had no tee in the shop, nor could we get one of the right length. The tee that burst was 15½ inches long, and the nearest tee we had was 17 inches long; but the way the pipe was connected to the engines and boilers made it impossible to use the latter. The difficulty was overcome in the following manner:

The 17-inch tee was placed in a lathe and 1½ inches cut from one end, the remaining portion of that end being turned round and smooth. A standard 8-inch pipe thread was then cut on it (see sketch) and an extra-heavy 8-inch pipe flange was fitted. This gave us a 15½-inch tee which served the purpose as well as a solid tee, and inside of six hours we had the engines running again.

E. W. NEAL.

Pittsfield, Mass.

Preventing Boiler Explosions

In a recent editorial entitled "An Epidemic of Boiler Explosions" you say, "steam users will either have to submit to Governmental inspection of their boilers, have them insured with competent inspection, or employ engineers who are qualified to judge of the safety of the apparatus in their charge, make them responsible for it and comply with their advice and requirements." If either proposition were accepted there would be fewer boiler explosions, and if there were a cold-water sprinkler which would automatically admit water into the steam space when the pressure was too high, all would be well.

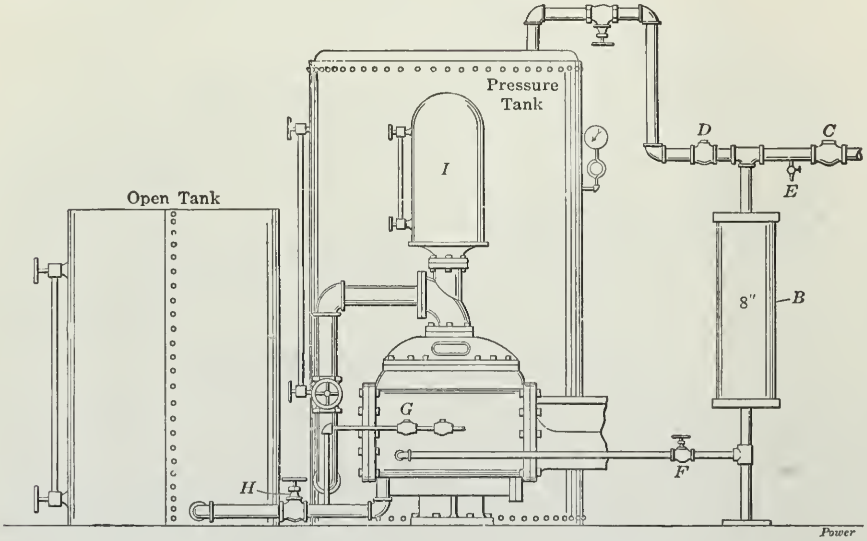
Peter Van Brock

Jefferson, Ia.

Supplying Air to Elevator System

The accompanying sketch shows an ingenious method of forcing air into the pressure tank of an hydraulic-elevator system. The cost of installing this device is small as compared with that of an air compressor, and it gives very satisfactory results.

The method of operation is as follows: First, valve *F* is opened while the pump is in operation and when *B* becomes full petcock *E* will show water. Just before air begins to be sucked in, on the return stroke of the pump, *E* is closed; this draws air in through check valve *C*. On the next stroke of the pump, the pressure closes *C* and forces valve *D* open, and the air passes over into the pressure tank. By this arrangement, no air gets into the body of the pump, the reservoir *B* being filled with water and pipe *F* is not sufficient to drain it at one stroke of the pump.



Piping by Which Pump Is Made to Supply Air to System

The pipe and check valve at *G* is for charging the air chamber *I* which loses air through the gage valves. By closing valve *H*, air is drawn in through *G* and soon appears in the top of the glass on the air chamber.

A. C. Waldron.

Revere, Mass.

Leaky Floats

I would be glad to have some Power readers offer suggestions as to how to repair leaky floats.

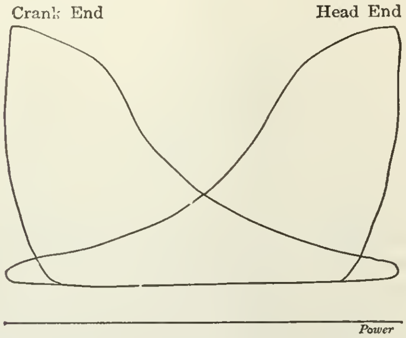
I have seen solder used but it would not be long before the floats, so repaired, were leaking as badly as ever.

E. Granfield.

Boston, Mass.

Diagrams for Criticism

The accompanying diagrams were taken from a 28x36-inch Murray engine running at 120 revolutions per minute. The steam pressure at the throttle was 150 pounds and the scale of the indicator spring was 60 pounds to the inch. Will some readers kindly tell me where the cards can be improved?



Diagrams from a Murray Engine

J. D. Robertson.

Battle Creek, Mich.

Exhaust Steam for Heating Water

It occurs to me that engineers, as a rule, do not realize the saving that can be effected by using the heat contained in the exhaust steam and the discharge from steam traps. It is a common practice to use exhaust steam for heating boiler feed water, but for this purpose only a small amount of the heat contained in the exhaust is used, the remainder being allowed to escape into the atmosphere. In many plants where exhaust steam is used for heating during cold weather, it is wasted into the atmosphere during the summer months.

It is a simple matter to condense the exhaust and return it to the boilers. There are two benefits derived from this; first, the amount of scale formed in the boilers is decreased; second, the water bill is less.

Where exhaust steam is used in heating water for house service, 2½ square feet of cooling surface per horsepower is ample for condensing the steam. A tank 42 inches in diameter by 10 feet long, containing a brass coil having a total surface of 140 square feet with a 2½-inch cold-water inlet and a 2-inch hot-water outlet, should heat water enough for the use of about 700 people; the exhaust from a 50-horsepower engine together with the discharge from steam traps being used to heat the water.

I utilize the exhaust steam from our engines and the discharge from the steam traps to heat all the water for bathing, laundry and cooking purposes in a large institution with which I am connected.

H. B. Brand.

Brooklyn, N. Y.

A Hydrostatic Test

Sometime ago at a marine-engine works in England, a hydrostatic test was applied to a number of small pinnace boilers. The intended working pressure was to be 150 pounds per square inch and the stipulated test pressure was 300 pounds. The hydraulic service at the works is under 1500-pounds pressure and the method of testing the boilers was to connect them to the hydraulic main, and by partially opening the valve, allow the required pressure to accumulate in the boiler. This plan usually worked all right but on this particular occasion the valve stuck and the pressure in the boiler rose to 1500 pounds. The valve was finally closed and after relieving the pressure, we made a careful examination of the boiler without finding any leak or serious deformation. The remarkable part of it was that when the boiler was installed and steam was gotten up, it leaked all over and had to be consigned to the scrap heap.

John L. Sheldon.

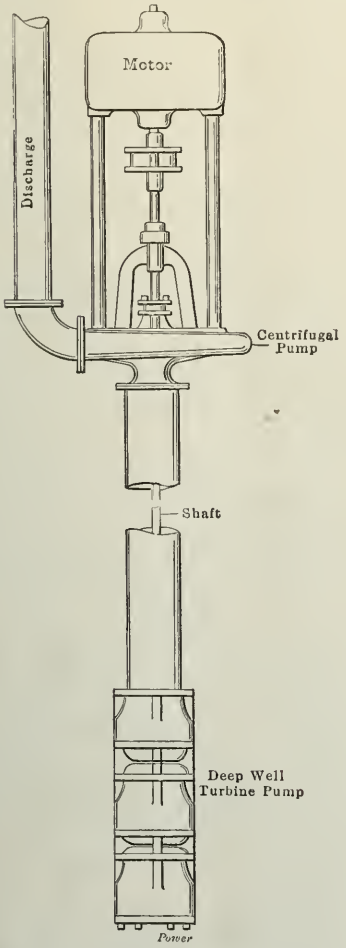
Prince Albert, Can.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Will the Pump Raise the Water?

In the July 5 issue W. F. Callister asks if, when the water level in a well is 70 feet below the surface and an impeller is used to raise the water to within 10 feet of the surface, a pump will lift the water from this point. This has been done. In 1904 there was a pump installed in an orange grove in California, which consisted of a combined auger and horizontal centrifugal pump. This pump was designed to draw 1000 gallons of water per minute from 75 feet below the surface and raise it to 80 feet above the surface. This arrangement is said to be patented by Charles D. Boydston.



Combined Auger and Horizontal Centrifugal Pump

At the present time there is in the course of erection a pump of similar design; the deep-well part of the pump is of the turbine type and the surface part is a motor-driven vertical centrifugal pump. The sketch shows the arrangement of this apparatus. The impellers of the turbine pump and the vertical centrifugal pump are mounted upon a common shaft. Both pumps are driven by means of a vertical motor through a flexible coupling.

In Mr. Callister's case there is no question as to the ability of this scheme to do the work as the theoretical suction distance for a pump is 34 feet at ordinary altitudes, although, due to imperfect vacuum and water vapor, it actually is somewhat less than 30 feet.

F. E. Sperry.

Aurora, Ill.

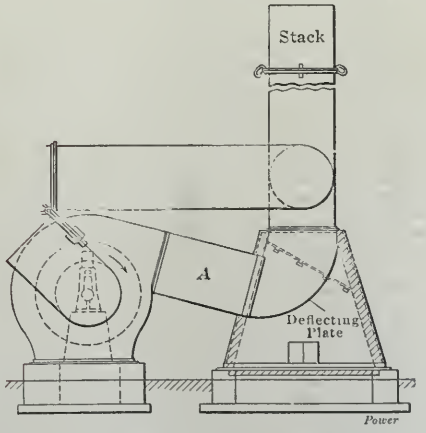
With reference to Mr. Callister's pump problem, as described in the July 5 issue, there is no doubt but that the pump would raise the water the remaining distance. The water is raised to a level of 60 feet above the supply by an impeller and as far as the pump is concerned this is its supply level and the pump should be able to create enough suction to raise the water some 25 feet higher provided the impeller does not prevent the transmission of atmospheric pressure through the water.

John French.

Washington, D. C.

Induced Draft Trouble

I wish to suggest that the trouble with Mr. Faulthier's induced-draft installation, described on page 1263 in the July 12 issue, is caused by the smoke being deflected downward on striking the sloping wall of the chimney base and by the eddies that are set up. If a curved deflecting plate were put in from the lower edge of pipe *A* and curved to as long a radius as possible, most and perhaps all of the trouble would disappear.



Suggested Improvement

H. B. Brand.

Brooklyn, N. Y.

With reference to the article having the above title, in Power of July 12, the trouble with the original outfit was twofold.

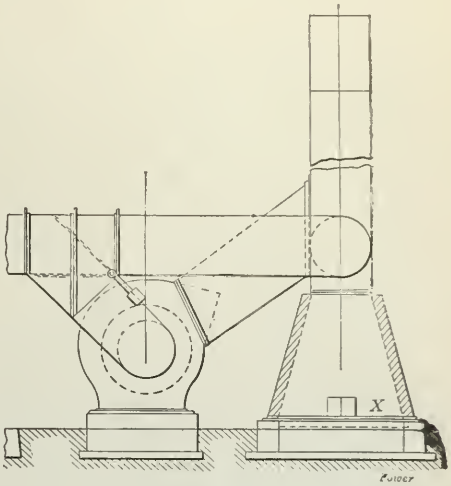


Fig. 1. New Arrangement of Fan Outlet

First, the engine was too small for the duty.

Second, the discharge connection from the fan to the stack could not possibly be of a worse design. The gases are discharged into one corner of the stack base, which forms a pocket. The impact reverses the gases and causes them to eddy.

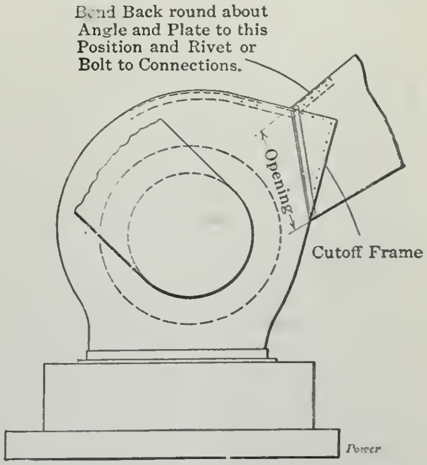

Bend Back round about Angle and Plate to this Position and Rivet or Bolt to Connections.
Opening
Cutoff Frame
Power

Fig. 2. Detail of Fan Alterations

The cheapest remedy is to install the 9x14-inch engine and alter the discharge connection so that it discharges the gases into the stack at a level not lower than the natural-draft connection, as shown in Fig. 1. This can be accomplished with little expense as it only means removing the outlet frame and a short portion of the roundabout plate. Details of this alteration are shown in Fig. 2.

E. W. Smith.

Worcester, Mass.

Breakage of Follower Bolts

Mr. Handley's follower bolts, as related in the June 28 number, broke so soon after setting them up that it looks as though it may have been a case of main strength applied at the end of a 24-inch wrench. With a wrench of that size, and we often see one used, it does not require a very strong man to twist off a 1-inch follower bolt. In this case, although the bolts were not actually twisted off, the beginning of a break may have been made and the jar of a few hours' operation finished the job. A 14-inch wrench is the largest that should ever be used for this kind of work; and it is not a good idea to try to bend the wrench either. For smaller sizes, a 10- or 12-inch wrench is large enough.

After putting in the new piston, Mr. Handley should have disconnected one end of the connecting rod and moved the piston till it struck at each end of the cylinder, marking these striking points on the guide. Then, after connecting up again and finding each center mark on the guides, he would have known what clearance he had, and would not be wondering if the piston were striking. Steam should not have been turned on until this had been done.

William E. Dixon.

Hudson, Mass.

Withholding the Facts

In the July 12 issue appears an editorial headed "You Will Never Know," commenting upon a boiler explosion at the Midvale Steel Works.

It states that management of the steel works refused to give any information to Power's representative as to the cause of the explosion and the resultant death of two employees.

To me this seems to be a stand which should be impossible to maintain by anyone. In some States the officers of the Midvale Steel Company would very soon be taken from their high perch, especially in Massachusetts.

In a case like this we have two departments which can and do make thorough investigations and their reports are public documents open to all. I refer to the Boiler Inspection Department and the Board of Medical Examiners.

The Boiler Inspection Department has the right to "enter any premises upon which a boiler or engine is situated, and any person who hinders or prevents or attempts to prevent any State boiler inspector from so entering shall be liable to the penalty as specified in this section." The penalty referred to is fine or imprisonment or both.

When the State boiler inspector arrives, and he is usually "Johnny on the spot," he gets all of the details fresh from the men who were present, and the report he submits to the chief of the department is such that anyone can understand, and is as near to the truth as it is possible to get.

Should an accident to a boiler involve the loss of human life, the Medical Examiners step in. These examiners are men of high standing in the medical profession; they use their power, which is great, in case of homicides, suicides or accidental deaths, for the benefit of the commonwealth.

In some large mills life is held to be a cheap commodity. It is proper to bring these matters before the public whenever occasion permits, so that public sentiment may be aroused to secure the establishment of proper authority for investigation.

Albert S. Smith.

Boston, Mass.

Inspection of the Governor

While reading an account of a recent flywheel disaster, my thoughts became centered upon the question of flywheel insurance and the conditions governing it. Looking at the question fairly and squarely, the flywheel protection is somewhat of a misnomer. While admitting that the question of rim velocity and the factor of safety for the various stresses should be thoroughly looked into, to my mind it is really of secondary importance. The governor is more important and should be looked after and insured, as it is the controlling feature of the whole machine. We read of many cases wherein the failure of a governor to act promptly has been the cause of disaster.

While engaged in visiting several engine rooms I ran across one when the engineer was in the act of mending his governor belt while the engine was running; the temporary regulation being made by throttling the steam to suit the load. During the operation of fixing the belt the engineer was temporarily called away to attend to some minor detail, and the thought came to me of what would be the result should the greater part of the load be suddenly thrown off by the breaking of a belt, or the throwing out of a circuit-breaker during his absence. Of course, this does not revert to the insurance of the governor, but the fact remains that insurance of the flywheel would not help matters any. Take the case of a Corliss engine the regulation of which is affected by the action of the governor on the cutoff. Suppose that from improper adjustment or wear the governor refused to act when the balls were rotating in a higher plane than usual and the load were suddenly reduced; the result can better be imagined than described.

A case is related in the July 19 issue of Power, in the article by G. Trube on "Runaway Engines," in which he states that "Owing to low steam on a Corliss engine the throttle had to be opened." This should have been done anyway as it is impossible to obtain a good full initial pressure and good economy by a partly closed throttle valve. He also says that "Shortly after the steam was raised the speed of the engine became dangerously high and the engineer ran to close the throttle, but could not do so." The question I would ask is this, "What was the matter with the governor that it did not take care of the increase in speed?" If engines are to be governed by throttling it is useless for engine builders to spend valuable time trying to improve and bring the sensitiveness of the governor down to the finest possible point, as it is by this means that economy in the expansion of the steam is produced. Therefore I reiterate that it is governor insurance that is needed more than flywheel insurance.

Charles H. Taylor.

Bridgeport, Conn.

Increasing the Power of an Engine

The question of "Increasing the Power of an Engine" which was discussed in the July 19 issue of Power, is a practical one, asked by many examiners in Massachusetts. It seems that the examiner wishes to draw out the ideas of the applicant on how he would know when his engine had reached the limit of its power. My answer to this question is as follows:

The limit to the range of cutoff by the governor being two-fifths of the stroke, the valves must trip at this point if they trip at all. To make the engine do more work, shorten the rod connections to the steam valves. This will give more port opening at the beginning of the stroke. To overcome the objection of too much lead, the eccentric should be moved back. The position of the eccentric, if it leads the crank, would be about 90 degrees in advance; after the change in the rods, connecting the wristplate and the steam valves, had been made.

Very little margin for such a change can be made on a single-eccentric Corliss engine without sacrificing compression or late release. But the double-eccentric Corliss engine can be made to cut off as late as half stroke, and even more work can be gained by shortening the steam-rod connections to the wristplate and by moving the eccentric back from 90 degrees in advance of the crank. This may be carried to the point where the throw of the eccentric, having reached its limit, would have to close the valve on its return stroke, allowing the steam to be partially forced out of the cylinder as the piston returned. This would give a Corliss valve the same economy in operation as a slide valve, and the engine would not cut off at all; being overloaded the valve would not trip. The engine would be doing much more work than it did with the eccentric set at 90 degrees plus the lap and lead with the cutoff by the governor ranging from one-fifth to two-fifths stroke. This is the change the engineer is expected to make before adding a condenser or compounding or changing the size of the flywheel.

R. A. Cultra.

Boston, Mass.

Chimney Design

In the issue of July 5 appears an article on "Chimneys and Draft" by John S. Leese, in which the author, under the subhead, "Types of Chimneys," states that "the wind pressure in designing chimneys may be calculated at 56 pounds per square inch."

Whether he wishes us to believe that this is a feat within the power of men, or simply may be the practice among designers, is hard to guess, although I am inclined to the latter view on account of his obvious unfamiliarity with this subject, as shown by this as well as other statements.

In most quarters, and I say this after considerable experience in designing chimneys, a chimney designed to withstand a wind pressure of 50 pounds to the square foot is as securely rooted to the face of the earth as the most extraordinary occasion would require.

H. R. Austin.

Oshkosh, Wis.

Centrifugal Pump Characteristics

In the issue of July 5, on page 1211, in an article on "Centrifugal Pump Characteristics," there appears, in paragraph four, this statement: "The efficiency which at zero head is zero." This is not borne out by the curves and data shown.

At zero head the table shows that 600 gallons per minute were moved, with an expenditure of 400 brake horsepower.

This means that 600 gallons per minute were not lifted to any hight, but were moved at a certain velocity, with the natural result that 400 brake horsepower was expended, or two-thirds of the maximum. If Mr. Sanford will give us the power demanded to produce the 400 brake horsepower, at zero head, the efficiency of the pump at zero head can be obtained. Otherwise, the curves and data are misleading as regards zero head.

Evidently, the pumps cannot deliver 400 brake horsepower, and have no efficiency, which is what the reading matter says, and the curves do not show.

H. G. Manning.

Pittsburg, Penn.

Long Runs of Pumping Machinery

In the July 19 issue, Mr. Dreyfus asks for information regarding "Long Runs of Pumping Machinery." I have charge of three cross-compound Corliss pumping engines, built by the Laidlaw-Dunn-Gordon Company. Each is 16 and 30x11x30-inch and raises water 247 feet for irrigation purposes. No. 2 engine has run continuously for over 56 days and No. 1 for 42 days, each carrying full load two-thirds of the time.

R. A. Davidson.

Colton, Cal.

Increasing the Capacity of Boilers

I was very much interested in T. T. Parker's letter in the July 19 issue on "Increasing the Capacity of Boilers," especially that portion concerning vertical fire-tube boilers. This type of boiler has many features of merit over other types of either fire- or water-tube boilers, with only one objection, that is, the removal of scale. But this, in itself, is not by any means insurmountable, even with bad water.

One feature of this type of boiler that does not exist in any other type is the fact that every tube evaporates the same amount of water, owing to the hot gases being evenly distributed over the tubes. In the horizontal tubular boiler, the three upper rows evaporate far more water per square foot than the lower rows and only the upper half of each tube is available as a positive heating surface. In the water-tube boiler there is a very high rate of evaporation at one end of the lower tubes, and as a result these tubes give trouble by bulging and bagging. This may be charged to a poor distribution of the heat, the high temperatures being applied locally to a very small portion of the total heating surface. It is true that in the vertical fire-tube boiler, the greatest heat is first applied quite a distance from the steam space, but in practice this amounts to little as the temperatures would be equal in the water contained above the grate line and we have superheat to dry the steam.

The theory of having the entering feed water meet the products of combustion just prior to escaping to the chimney is of little practical value, for, in a steam boiler, with a well-defined circulation, the temperatures are practically equal where the water surface is exposed to the products of combustion.

Mr. Parker refers to vertical boilers without water legs, a type I have never met with. I would consider a water leg valuable in permitting the lower tube sheet to be flanged downward, thus facilitating the cleaning of the sheet and the collection of loose scale at the mud ring below the grates. In addition, by using hollow staybolts with the holes 3/16 or ¼ inch in diameter, we would have the admission of air above the fire that is necessary for proper combustion. This would be properly distributed and diffused equally to all parts of the furnace, which is not the case with other types of boilers. The air would enter through very small apertures and absorb heat from each hot staybolt. Consider a vertical boiler with a 72-inch lower tube sheet having a 4-inch furnace, with staybolts on 6-inch centers; 266 bolts would be required and with 3/16-inch holes one would have about 7 square inches of air inlet, granting the inlets were free. One could easily determine the proper admission and then block off the inlets not needed, doing this from the furnace sides of the staybolts. But the value of such air admission in completing combustion before the gases enter the tubes appeals to me and is a special feature of this type of boiler provided with a water leg.

The boiler would probably require more care, as regards the admission of cold air when cleaning the fire. The tube ends would be susceptible to such mistreatment as in a locomotive boiler and tube leakage would be in proportion to the care the boiler received. Men accustomed to leaving a fire door open without taking the trouble to close the damper would, no doubt, spend far more on repairs than others. The personal equation would figure largely, but with proper care, tubes should give no more trouble in this type than in other types.

As the shell is out of the fire and is subjected only to temperatures equal to that of the water and steam, its life should be much longer than that of a horizontal tubular boiler. Properly jacketed outside, the inside could be painted with graphite, or other good paint, thus eliminating internal corrosion. Without internal corrosion and with no wear and tear on the shell, the annual depreciation would be greatly decreased.

With a horizontal tubular boiler of 150 horsepower I would assume the annual cost of maintenance to brickwork as $50, or $1000 for 20 years—the average life of such a boiler; this is equal to 50 per cent. of the first cost. But this brickwork is of no value to the steam user except for jacketing the furnace and boiler. It is in a sense a flywheel in only the best furnaces and is inefficient in proportion as it permits air leakage beyond the bridgewall or over the tubes, where such leaks are extremely wasteful. Bricks are porous to water and much more to air or gases. With the closure of the fire door in a vertical fire-tube boiler, provided with a water leg, there are no such deleterious air leaks, no first cost for brickwork and no repairs of this nature, that in 20 years aggregate one-half the cost of the initial installation. The amount of water evaporated per pound of combustible is practically the same in all types of boilers.

A. J. Foran.

Paterson, N. J.

Boiler Failures

The safety of a boiler demands closer attention at the present time than ever before, from the fact that the majority of our cylindrical boilers are now made of steel plate. In practice this plate shows that it does not deteriorate evenly, and that crystallization is more liable to occur in the steel plate than in the old charcoal-iron plate, from which boilers were formerly made. As a result, a boiler that is apparently sound may have a hardened spot which will snap like glass if subjected to an uneven stress. This condition is often found on the flange of the flue head, or on the rear head of a return-tubular boiler.

We find many boilers in which the ends of the tubes have partly rusted away. Now, a tube may be considered as a stay, and it will hold in exact proportion to the shear on the bead; therefore, when the beads are gone, the stress comes on the diagonal brace supporting the head. Frequently this brace is in a corroded condition, or perhaps the base of the brace is cracked and the slot has become worn until the brace pin becomes loose, allowing it excessive play. A loose brace pin is a dangerous thing for any boiler, and especially if the boiler is otherwise weakened. To illustrate this point, consider an elevator cable which will hold a car in ordinary service, but should the car drop a few stories, the cable would probably snap. This is true of a boiler head if the flues are rusted away and the brace has play; a slip of the head is apt to cause an explosion. With the brace pin tight, this would not happen.

This condition is more pronounced at the top head on upright boilers and especially so on boilers that have no hood over the smokestack to prevent rain from coming down. A wire brush should be used on the tube heads to remove the scale, in order to see the exact condition of the tube sheets. Many of the accidents are from small boilers that have no regular inspection. Merely looking over a boiler does not mean that it is receiving a careful inspection. Brace cracks are found only by close observation and often a brace foot will be almost gone before the crack can be easily seen.

C. R. McGahey.

Sheffield Ala.

Bypass on Boiler Stop Valves

The letter of James W. Little, in the July 26 issue, came as a surprise to me, as I was under the impression that all well designed plants used the bypass on their stop valves.

In 1898 I designed and installed a small high-pressure plant in the South, and in this instance each of the two 300-horsepower Babcock & Wilcox boilers was connected to the main header by an 8-inch Walworth rising-spindle, heavy-pressure, flanged gate valve, on which a bypass valve of similar construction, of about 1½-inch diameter, was placed.

These valves and all other pipe flanges on the system were packed with ring gaskets of asbestos millboard, soaked in water and then drawn down until the flanges were practically together. These valves and connections gave no trouble during my stay at the plant and I have heard of none since.

I thought at the time that I was keeping in line with modern practice by installing those bypass valves, instead of violating established precedent, but now Mr. Little mentions it, I am reminded of the fact that no other plant where I have worked has been thus equipped. Whether this state of affairs is caused entirely by precedent, or largely because of first cost, would be an interesting subject for discussion.

I still believe it to be a good investment if only as a matter of insurance against carelessness or inexperience.

L. L. Arnold.

East Greenwich, R. I.

Vibrations in Boiler Header

Referring to a letter by G. E. Miles, under the caption "Vibration in Boiler Header," in the issue of July 19, it appears to me that the vibrations were started from the boiler which had not been cut out from the line. If he had closed the main steam valve on the boiler from which the fire had been drawn, the vibrations would have ceased immediately.

Mr. Miles did not state how long the fire had been withdrawn from under the boiler when the vibrations began. There is, no doubt, some peculiar action which goes on in a boiler from which a fire has been withdrawn previous to closing the main steam valve.

I believe that the opening and closing of the steam valves in a cylinder cause the steam in the header to surge forward and back with great rapidity. The boilers which have a fire under them generate enough steam to run the plant and at the same time, on account of the continuous generation of steam, they are in a condition to resist the surging of steam, just mentioned. This is not so with the boiler to be cut out; the ebullition of the water has ceased and no more steam is being generated; consequently the boiler is subjected to a succession of rapid shocks which would cause the header to vibrate if other conditions were favorable.

Mr. Miles states that the vibrations slowly but gradually increased. This is just what would be expected, and the engine running under a light load tended to increase the intensity of the vibrations on account of shorter cutoff.

I recall an incident that happened several years ago, which would tend to show how steam acts under such circumstances. It was in a steel mill which had waste-heat boilers located over the puddle furnaces. The puddlers had drawn the heat and shut off the steam blast. The steam pressure in the other boilers, all of which were in operation, was raised to the blowing-off point; and the safety valve began to hammer on the boiler from which the puddlers' heat had been withdrawn. The men became frightened and began to run in every direction, thinking the boiler was going to explode. The assistant master mechanic climbed up on top of the boiler and attempted to screw down the adjusting nut on the stem, in order to stop the hammering. It hammered so much that he could not get the wrench on the nut and had to give up. After repeated efforts by others to stop the noise, all of which were failures, it finally occurred to someone to close the main steam valve. This was done and the trouble ceased instantly.

W. W. Scott.

Youngstown, O.

Packing Condenser Tubes

The method of packing condenser tubes as described in Mr. Gibson's article in the July 19 issue, I have found to be unnecessarily slow and tedious.

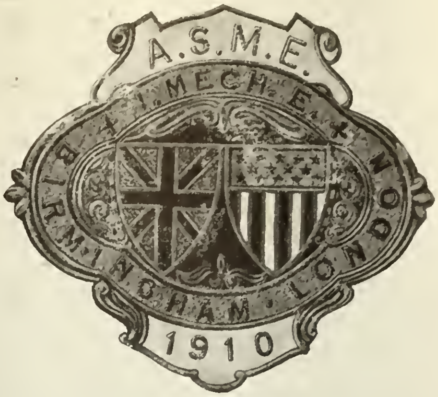
A much better method is to use an old piece of tube, one end of which is cut to fit between the condenser tube and the stuffing box, and the other end to fit a carpenter's brace. With packing cut into lengths of about 18 inches and having been previously soaked in boiled oil, a perfectly tight job can be easily obtained in a very short time.

W. E. Simpson.

New York City.

The Mechanical Engineers Abroad

The "Celtic," bearing a party of 148 members of the American Society of Mechanical Engineers and their families, dropped anchor in the Mersey, early on Sunday evening, July 24, and was shortly boarded by the deputy lord mayor of Liverpool (John Lea), President J. A. F. Aspinwall of the Institution of Mechanical Engineers, Professor Watkinson, of the University of Liverpool, and others. The three gentlemen named extended the welcome and greeting of the city, the Institution and the university to the visitors. Professor Watkinson spoke very kindly of his treatment in "The States" last year and complimented our universities, that of Illinois especially, upon their research work. The addresses were responded to by Dr. W. F. M. Goss, dean of the University of Illinois and vice-president of the American society.



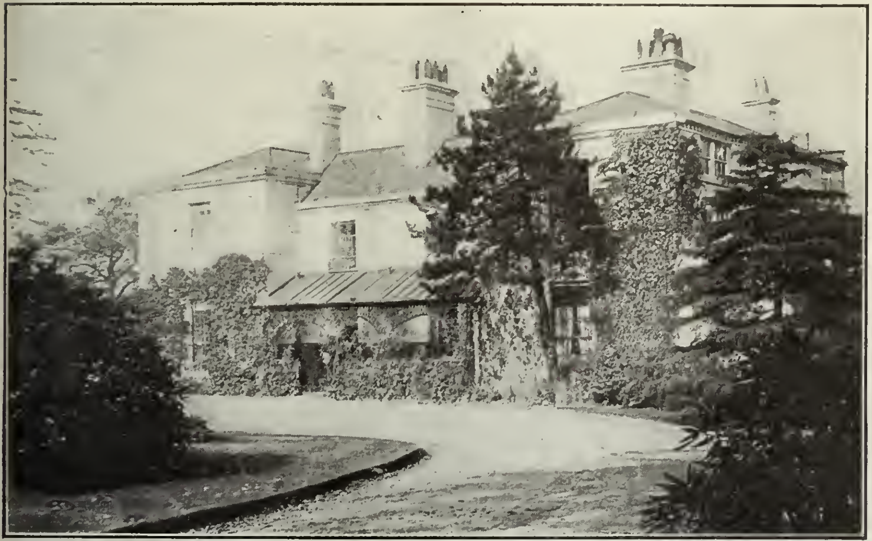


Fig. 1. James Watt's House, Heathfield Hall, at Birmingham

Most of the party remained aboard Sunday night and left Liverpool on a special train composed entirely of dining cars, at 11:15, arriving at Birmingham somewhat after 2 p.m. The afternoon and evening were spent in getting settled, meeting old and making new acquaintances. Handsome international badges enameled in red, white and blue were distributed.

The first session of the meeting was called to order in the lecture room of the Birmingham and Midland Institute by the right honorable the lord mayor of Birmingham, Alderman W. H. Bowater, who said that it was quite fitting that the Institution should bring its guests to Birmingham, for it was here that the Institution was founded at the Queens hotel, over sixty years ago. He extended a hearty welcome in the name of the city of Birmingham and called upon George Tangye, who presented to the American society an autograph letter written by James Watt to his partner Matthew Boulton in 1777, a photograph of which is reproduced herewith. The addresses of welcome were responded to and the gift accepted on behalf of the American Society of Mechanical Engineers by Doctor Goss.

President Aspinwall, upon taking charge of the meeting, suggested that the regrets of the meeting for his inablity to be present be telegraphed to President Westinghouse, of the American society. This was done and after approving the minutes of the last meeting of the Institution and electing 122 new members, the meeting passed to the consideration of the professional papers. These dealt for the most part with railway practice, and such portions of the meeting as will interest Power readers are fragmentary, as F. H. Clark's statement that out of over 1000 locomotives they have about two failures per day sufficiently serious to incapacitate the engine, and that of these failures 40 per cent. are due to leaking tubes and 5 per cent. to burst tubes. Lunch was served at the town hall.

In the afternoon, excursions were made to Dudley Port, Tipton, where are the South Staffordshire Mond Gas Company's works and the testing station of the Pump and Power Company, of which H. S. Humphrey is the managing director, and where the famous Humphrey internal-combustion pumps were on view; to several industrial establishments and to Stratford-upon-Avon, the waterworks, Worcester, Stoneleigh Park and Kenilworth.

Your editor spent a delightful after-

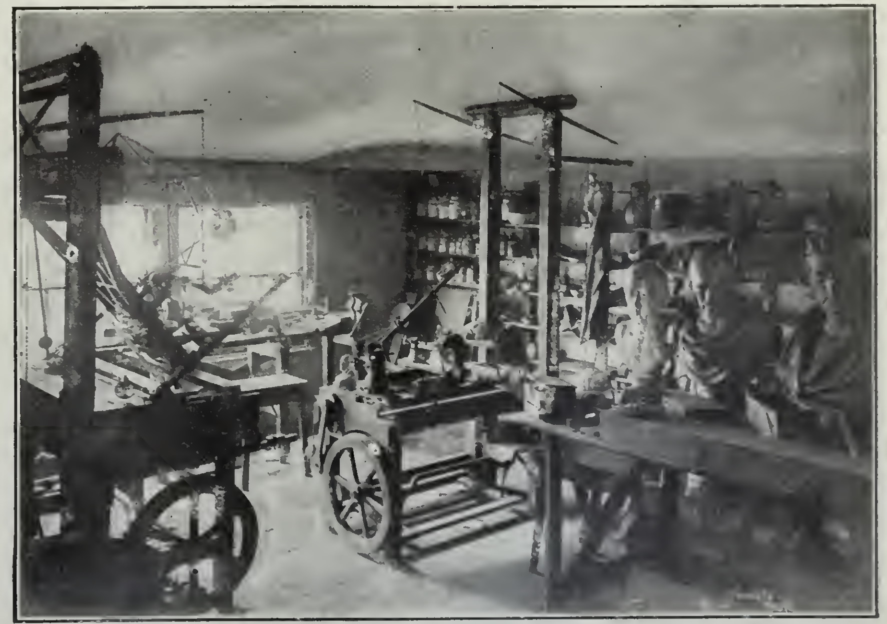
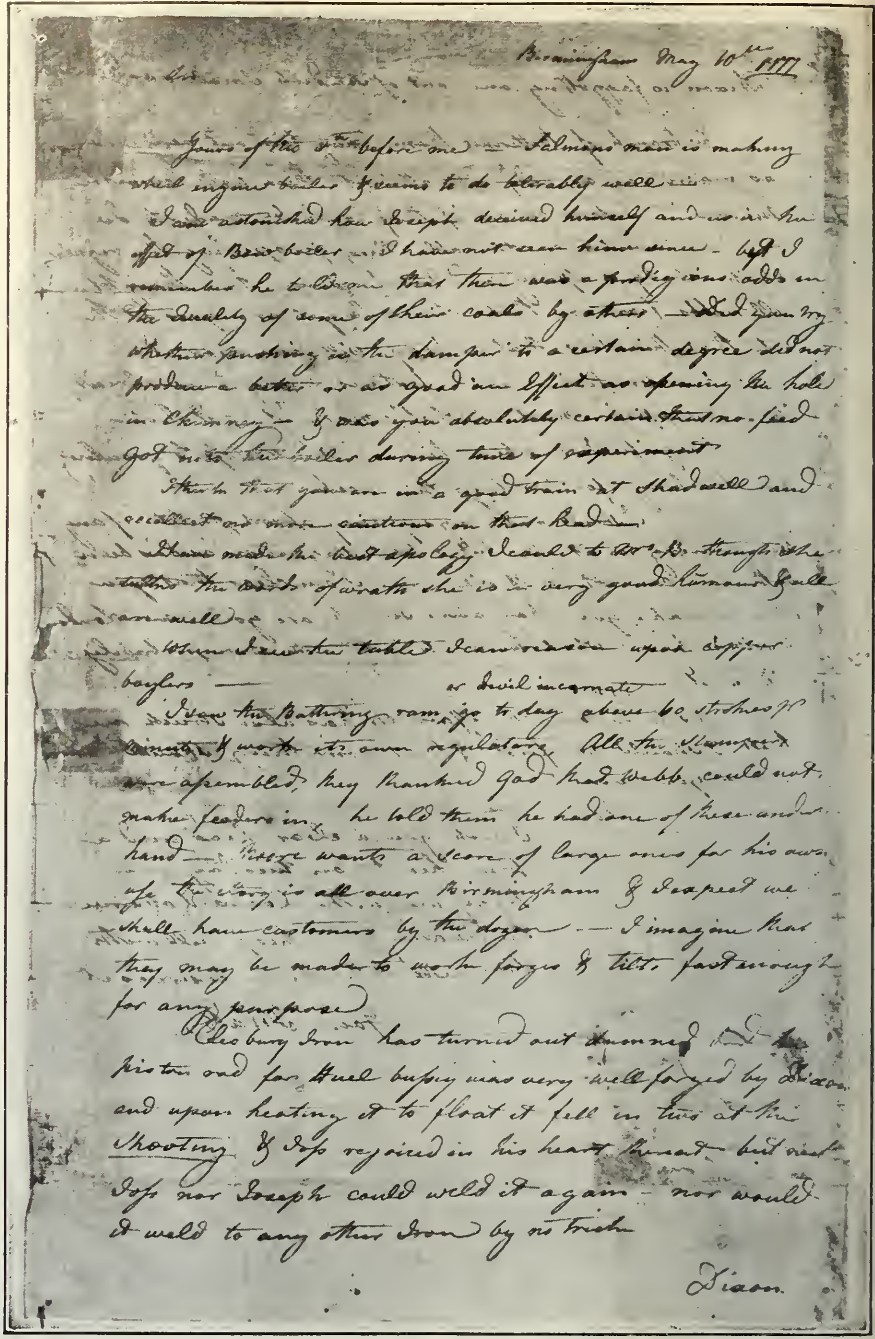


Fig. 2. Watt's Workshop at Heathfield Hall



Birmingham May 10th 1777

Yours of the 8th before me — Falmouth man is making wheel engine boiler & seems to do tolerably well

I am astonished how Joseph deceived himself and is in the affair of Bear boiler — I have not seen him since — but I remember he told me that there was a prodigious odds in the quality of some of their coals by others — did you try whether pushing in the damper to a certain degree did not produce a better or as good an Effect as opening the hole in Chimney — & was you absolutely certain that no feed got into the boiler during time of experiment

I think that you are in a good train at Shadwell and recollect no more cautions on that head

I have made the best apology I could to Mrs B. through the cutting the cords of wrath she is in very good humour & all are well

When I saw the tickled I can reason upon dipper boylers — or devil incarnate

I saw the Battering ram go to day above 60 strokes pr minute & work its own regulators. All the Stampers were assembled, they thanked God that Webb could not make feeders in he told them he had one of these under hand — George wants a score of large ones for his own use the story is all over Birmingham & I expect we shall have customers by the dozen — I imagine that they may be made to work forges & tilts fast enough for any purpose

Cleobury Iron has turned out [illegible] the piston rod for Hull bussy was very well forged by Dixon and upon heating it to float it fell in two at the Shooting & Jos rejoiced in his heart thereat — but neither Jos nor Joseph could weld it again — nor would it weld to any other Iron by no trick

Dixon

Fig. 4a. Watt's Letter to Matthew Boulton

noon at Heathfield, the residence of James Watt in his later days and the present home of George Tangye. The original part of the house is at the right of the accompanying picture, and the double windows showing through the trees in the upper story are those of "the Watt room," the photograph of which is also reproduced, where the master had his little foot lathe and worked out his later inventions. Watt was not only an engineer but a diplomat, as a review of his correspondence shows. Mr. Tangye says that it is also established that he made sun pictures long before Daguerre. The interior of this room, with his belongings and materials as he left them—oil and chemical in bottles and jars, a handkerchief where his head rested on the bar of a reproducing machine, wax tapers and lots of little personal effects—is kept by Mr. Tangye as Watt left it, full of tradition, of memories of the Lunar Society, whose membership, including Josiah Wedgwood, Matthew Boulton, Dr. Erasmus Darwin and Doctor Priestley, met here on moonlight nights, because they could not see to come otherwise, and of leading strings back into the beginnings of the age of power. And the genial and kindly present owner, glad in the possession of it, pleased to be in a way a part of it, for his family is related with the Trevithicks and his own house and name are prominent in the more recent history of engineering, fits charmingly with the picture and finds his greatest pleasure in sharing it with those who appreciate its memories and significance.

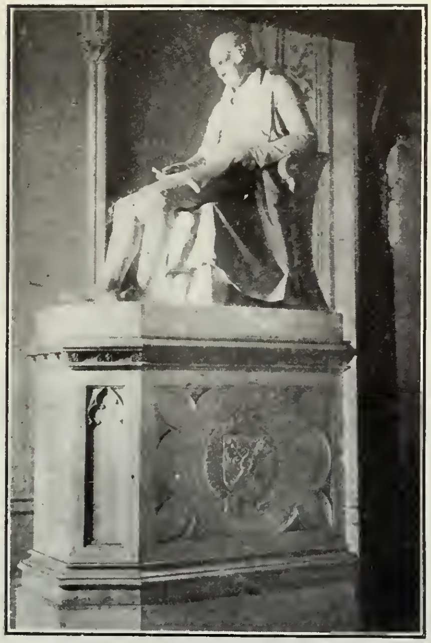


Fig. 3. Monument of Watt over Vault Where He Is Buried

Across the fields is St. Mary's church, of Old Handsworth, where James Watt is interred. The accompanying photograph shows the monument above the family vault. At the right of the altar is a bust of Matthew Boulton, and at the left one of William Murdock, the inventor of illuminating gas. In the Gillott pen factory nearby is a Watt engine which has been running for 98 years and is still at work.

For kindly interest and attention during my visit to Heathfield Hall, grateful acknowledgment is made to R. H. Kirton, curator of the Watt museum.

In the evening there was a garden fete at the Edgbaston Botanical Gardens, at which the lord mayor, his wife, and Neville Chamberlain, Esq., chairman of the executive committee, received and the band of the Portsmouth division of the Royal Marines furnished the music.

Wednesday morning was devoted to the consideration of papers, the afternoon to a visit to the University of Birmingham, or to Mitchells & Butler's brewery for those who preferred, and in the evening a reception was tendered in the Council House by the lord mayor and his lady.

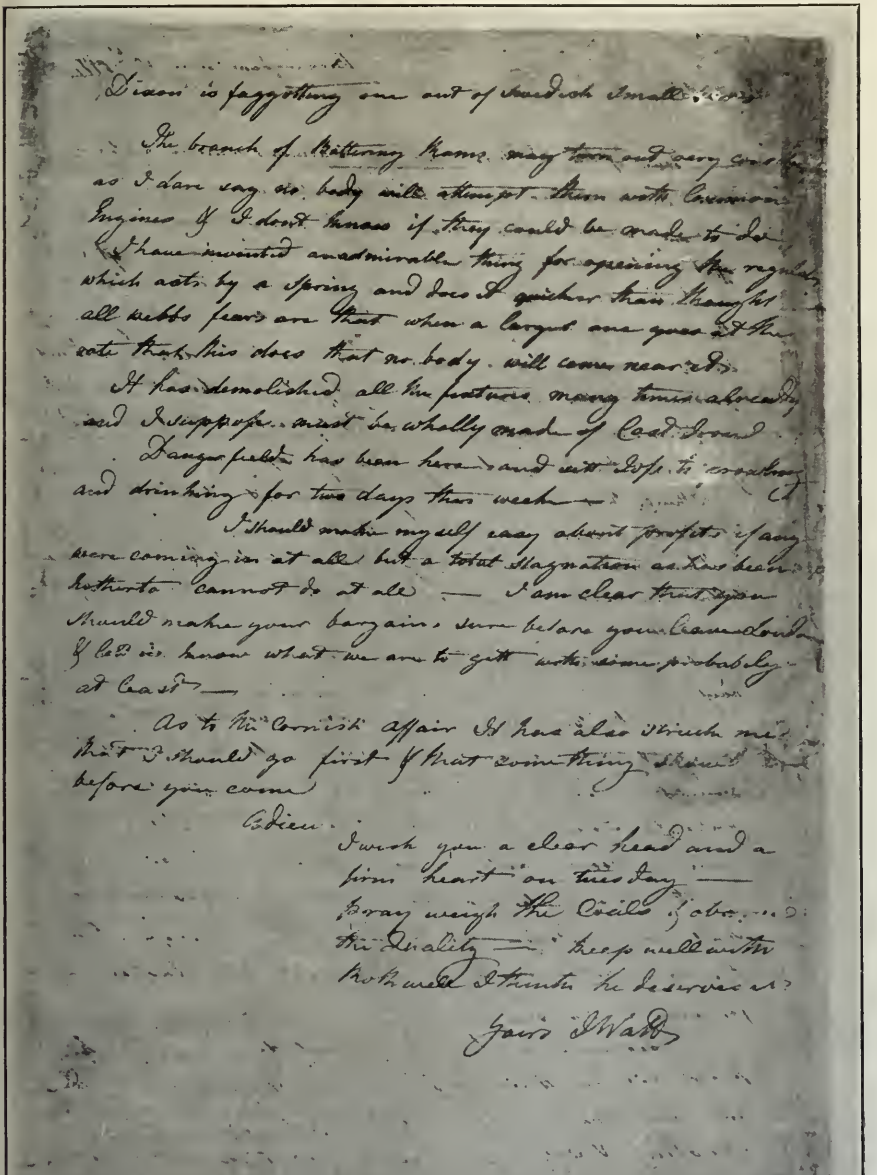
Thursday was devoted entirely to excursions, of which choice was invited from a number of the principal attractions which these historic and industrial Midlands afford. On Thursday evening the party left for London, arriving in time to take part in a conversazione at the house of the Institution of Mechanical Engineers.

Grooving in Steam Boilers

By W. H. Booth

When pressure is raised in a boiler with longitudinal lap seams the outer plate tends to bend over the edge of the inner lap as the plates strive to become truly circular. The bending stress causes grooving of the plate. By proper precautions this grooving may be avoided.

Every vessel containing internal pressure tends to assume the form of a perfect sphere. Vessels such as steam boilers are restrained to shape by means of stays to their flat surfaces. The staying of the ends of a cylindrical boiler enables the barrel or body of the boiler to strive only to maintain a truly circular form in cross-section. The cylinder form is up to a point as scientific as the spherical, but it introduces the difficulty of the stayed flat end. The old egg-ended boiler once so common in Great Britain has hemispherical ends and cylindrical body, and is therefore exempt from any tendency to change its form, for both ends and body are ideal in shape.



Dixon is faggotting one out of Swedish Small [illegible]

The branch of Kettering Rams may turn out very [illegible] as I dare say no body will attempt them with Common Engines & I dont know if they could be made to do

I have invented an admirable thing for opening the regul which acts by a spring and does it quicker than thought all Webbs fears are that when a larger one goes at the rate that this does that no body will come near it.

It has demolished all the fixtures many times already and I suppose must be wholly made of Cast Iron.

Dangerfield has been here and with [illegible] to [illegible] and drinking for two days this week —

I should make myself easy about profits if any were coming in at all but a total stagnation as has been hitherto cannot do at all — I am clear that you should make your bargain sure before you leave London & let us know what we are to gett with some probability at least —

As to Mr Cornish affair It has also struck me that I should go first if that something should [illegible] before you come

Adieu.

I wish you a clear head and a firm heart on tuesday — pray weigh the Coals & obs the Quality — keep well with [illegible] I think he deserves it

Yours J Watt

FIG. 4B. WATT'S LETTER CONTINUED

If a plate be bent into a circle and the ends laid together and riveted, there will not be formed a true circle, but when put under pressure this ring of plate will strive to become truly circular, and in so striving there will be a tendency to bend the outer plate over the edge of the inner lap. It is this bending stress which causes grooving of the plate because the sharp nip over the edge of the plate cripples the bent plate along that line and facilitates the action of the water upon its overstrained particles. Steam does not appear to have such an effect, so that it became usual to place the longitudinal seams of shell boilers always above the water line so as to avoid this action of the water.

If a boiler ring be drawn on paper to a true circle the lap joint can be designed so as not to disturb the true circularity. This is done by bending one plate lap at an outward angle and the other equally inward. So shaped, the rivets through the two plates are not exactly radial to the boiler, and there will be no movement of the joint, though the rivets will now be partly in tension and there will be a steady stress along each lap at the angle of bend. There is also a great tendency to injure the plates when bending them to the necessary angle. As a matter of good practice, therefore, the lap weld has been abolished and rings of plate are now made as truly circular as possible, the ends butting together and the joint being made by a pair of cover strips placed inside and outside the boiler. All rivets are now radial and there is no tendency to depart from the initially correct form.

Did boilers work under a steady pressure, there could be no change of curvature once the shell had taken the form of equilibrium. But since the stiffness of the plates is exerted against the steam pressure the boiler will become more or less truly circular as the pressure rises and falls. This variation of pressure is at the root of all grooving troubles of the order referred to, for the plates are always striving to approach their initial form and the pressure is always striving to push them to the exact circle.

Some years ago several lap-jointed boilers on a British railway exploded as a result of grooving along the longitudinal seams. The effect of observation of other boilers and of the exploded boilers brought about, first, the placing of such seams above the water level and, next, of abolishing this type of seam in favor of the butt seam with double butts. The single-butt seam has a stress in it along the middle line of the butt strip. Hence the use of the double-butt strap, each strip or strap somewhat less in thickness than the plate to be jointed. With this seam double riveted and with a third row of rivets at double pitch beyond, the strength of a joint may be made about 85 per cent. of the strength of the solid plate, and the tension is always through the middle line of the plate thickness, or at the point equally on each side of the middle thickness, so that no tendency to alteration of shape is induced so long as the finished rings are truly circular.

Grooving does not appear to be set up unless the water is slightly corrosive, but very little corrosive influence appears to be required to cause grooving when there is the necessary changing stress in the plate. Flat surfaces are very apt to groove, especially when exposed to a sharp bend. Thus, in the Lancashire type of boiler the front end plate where attached to the furnace tubes is thrust forward ¼ to ⅜ inch by the expansion of the furnace crowns, and the plate often grooves at the point where it is sharply stressed at the edge of the connecting angle iron. This grooving was very severe at the time when the front end plate was stayed by gusset stays and the lowest rivets of the gusset angle irons to the front end plate came as near as 4 inches to the rivets of the furnace angle iron. Today the free space is made 10 inches and the plate is so much less sharply stressed over the edge of the angle iron that grooving has been very largely eliminated. The stress may also be much moderated if, when cold, the front end plate had actually to be drawn inward to meet the furnace angle iron. Thus, if drawn in say ¼ inch and the furnace tube, when expanded by the fire, pushes it out ⅜ inch the net expansion is only the difference of the two amounts, or ⅛ inch, and this represents very slight bending when the free plate is 10 inches.

Grooving will occur in short lengths of only an inch and always where a plate is pulled over the edge of an angle iron or other thickness. This conclusively proves that its initial cause is the overstress of the surface fibers where nipped short over a corner. A study of grooving is a lesson in boiler stresses and is convincing proof of the need for true forms or elasticity. True forms are not always convenient and flat surfaces are very much employed in boiler work with suitable staying. But flat surfaces require very careful design, and the designer will be better fitted for the work if he has observed the defects to which flat surfaces are prone. The whole question of grooving in boilers seems to resolve itself into one of stress in the outer fibers of the plate. If these are too much stressed they appear to be more open to attack by the water. The extreme fiber stress is of course reduced when the bent portion of a plate is long rather than short. Any acidity of the water simply intensifies the action. That these are the causes may be taken as pretty clearly proved by the fact that active grooving has often been stopped by cutting away one or two of the attaching rivets of a gusset stay angle iron, thereby adding to the length of the part of plate under stress and diminishing the angle of bend.

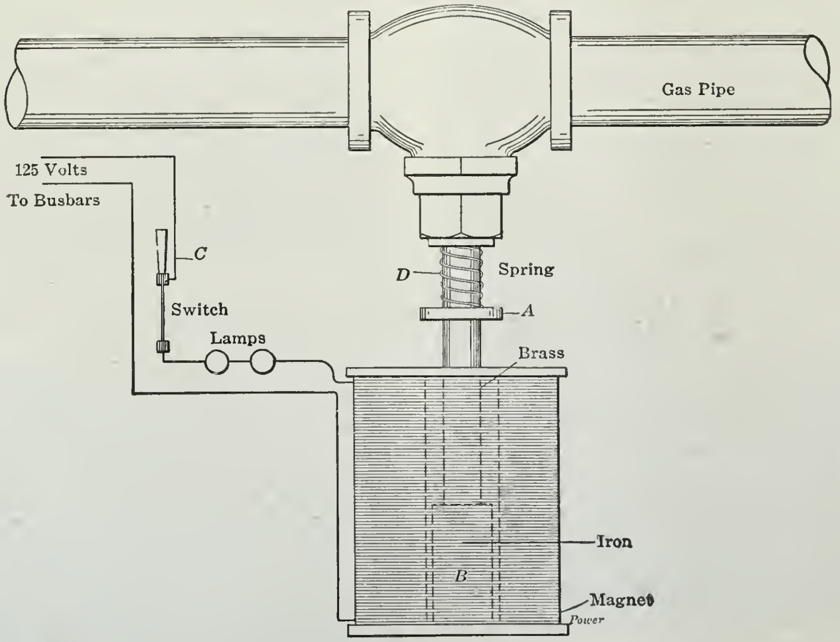
Grooving will take place round a very stiff flange of a manhole mouthpiece on a flat plate and reason would lead one to make all such blocks with flange attachments, as thick as possible, and moreover extended to some considerable extra width by an extension of tapering section finishing quite thin at the edge. This is perhaps an unnecessary refinement though the principle underlying the suggestion might be made useful for some special cases or conditions.

Gas Lighting in Electric Plant

Although no chief engineer cares to admit the probability of affairs ever reaching such a crisis that the entire steam plant will be out of commission, it is nevertheless a wise precaution to provide for some means of illuminating the engine and boiler rooms of his power plant in case the current should fail.

One method is to install a small storage-battery system, to which a number of very small incandescent lights are wired and distributed at suitable points about the engine and boiler rooms, one light always being in circuit and located at the switchboard. This system can be arranged at a small cost, and the battery charged at any convenient time.

Another, and perhaps the better plan where gas is obtainable, is to arrange for a number of gas lights, at a convenient point on the ceiling or elsewhere, fitted with a small gas jet that is always burning, so as to ignite the gas when it is turned on to the larger burners. An automatic method of lighting is shown in the illustration, and is the system installed in a large power station at Birmingham, Ala. The system consists of a magnet, the winding of which is connected to the 125-volt busbars which are always "alive" so long as the machines supplying them are in operation.



Automatic Gas-light Control

At a convenient point a globe valve is connected to the gas pipe with the valve stem in a downward position, as shown. The stem has no threads and moves easily up and down when operated. To the valve stem is screwed a piece of brass rod having a collar *A* on one end. This brass rod also screws into the iron plunger *B*, which is suspended in the coil at a point below the center. As the current is always passing through the coil when the switch *C* is thrown in, the tendency is to draw the plunger *B* toward the top of the coil, as the magnet is strong enough to overcome the resistance of the spring *D*, the weight of the valve and stem and the pressure against the valve disk. The valve will open the moment the current passing through the coil fails. This permits gas to pass to the various gas burners distributed throughout the plant, where it is ignited by the small jets in the burners.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for July, 1910, *was* 138,000.

August 2	34,000
August 9	34,000
August 16	34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

Safety of Boilers

It is a well known fact that there are fads and fashions in matters mechanical as well as in spring millinery or the latest cut of clothes. The most recent fad affecting the steam-generating end of power-plant equipment is to try and see how much steam can be generated per square foot of water-heating surface, and some engineers have apparently gone "daffy" on the subject. It seems that since the results of certain experiments made at St. Louis to prove or disprove Prof. John Perry's theory regarding the transmission of heat in boiler practice have been published, many engineers have been trying to prove to their own satisfaction that Professor Perry was in the bounds of reason when he stated that boilers should be made to do ten and possibly twenty times their usual rate.

High rate of heat transmission in boiler operation is by no means a new thing, although engineers are doubtless learning more every day about the laws and conditions governing transmission of heat.

The old two-flue boiler, which is now practically discarded, could deliver a boiler horsepower on one and one-half to two square feet of water heating surface without the slightest difficulty; in fact, the available capacity of this type of boiler was such an uncertain quantity that the only satisfactory method to pursue in replacing them with other types was to measure the actual evaporation obtained and use this as a guide in selecting the proper capacity for the new boilers. Comparison on a heating surface basis between the two-flue boiler and other types was likely to lead to most embarrassing results for the engineer responsible for such changes. The high evaporative performance of the two-flue boiler was, however, not obtained by uniformly increasing the evaporation over all the heating surface, but by extending the area of direct heating surface so as to include all of the heating surface of the boiler, for the combustion of the fuel in this type could be completed in the flues, and it was sometimes completed in the stack, if forced enough, all of which did not contribute to high efficiency, but did tremendously increase the capacity.

To make modern types of boiler do many times the work that is ordinarily expected of them, and do it with safety, it will be found necessary to have the additional work done by the heating surface not directly exposed to the fire, just as in the case of the two-flue boiler. Anyone conversant with customary boiler operation of today will not believe that the direct-heating surface of a boiler, if made of the usual materials, can possibly transmit ten times the heat that it is usually transmitting at present and still retain its physical characteristics.

Boiler-insurance companies have frowned on the various attachments for furnaces and boilers that have been brought out from time to time with promises to double capacities, and experience has taught that many such devices are detrimental to safety and seriously reduce the life of boilers on which they are used. With the agitation for, and the actual enactment of laws governing, the safe construction and operation of steam boilers, there will come a time when the maximum capacity that is allowed to be taken from a given amount of heating surface will be limited just as the steam pressure allowed is limited today. Overworking a boiler as regards the quantity of steam generated is just as dangerous as overpressure, and possibly more so, because of the insiduous manner in which defects rendering a boiler unsafe are produced by this means. Careful bookkeeping will probably show that with the possible exception of plants having a high peak load lasting for comparatively short periods each day, it is not economy to force boilers much beyond the usual rates that are obtained now. There are other items beside interest on the investment and operating charges to be considered, and reasonable safety of operation is one of these that must be given due weight.

Paying for the Right to Work

When it is demonstrated by proper examination that an engineer is competent to take care of a steam plant, whose interests are directly safeguarded, the engineer's or those of the public? Is it the purpose of a license law to prevent incompent engineers from hurting or killing themselves or to protect the public from the results of their ignorance? There is, of course, only one possible answer. Public safety is the object of all engineers' license laws, in the broad sense; protection of the engineer himself is not one of the considerations on which the laws are based. Why, then, should an engineer be made to pay any fee for examination? Is the foreman of a powder mill required to pay a fee to any municipal authority before he is allowed to earn his living in his hazardous vocation? Not to our knowledge. Nor is any other skilled worker, no matter how much opportunity he may have for inflicting damage to life, excepting prescription clerks in pharmacies. We believe that they have to pay for their examination and pharmaceutical licenses, which is utterly unfair, but we do not know of any other classes of employee that are mulcted in this way.

The so-called argument that merchants, contractors, etc., have to pay for license to "earn a living" is sometimes dragged out as a reason why engineers should do so. Leaving aside the question of whether it is just to charge a business man for a license, the case is so obviously different from that of a man who works for a stipulated rate of pay that the "argument" is not worthy of discussion.

The bare, fundamental facts are that an engineer is examined in order to determine whether or not he will be a menace to the community if he should be allowed to control a steam plant, and a license is issued to him to show that he has given evidence of trustworthiness. The whole procedure is for the protection of the community, and the community should pay for it, just as it pays for police protection and a fire department.

The Biggest Factor

In the agitation that is going on at the present time for the enactment of laws prohibiting the construction of unsafe boilers and prescribing examination for the determination of the fitness of those who are to be intrusted with their care, the boiler operative should not lose sight of the fact that eternal vigilance on his part is one of the most necessary requisites to safety. It is unfortunate that an attendant's carefulness cannot be definitely determined by an examination.

The water column may be of the best design, its connections to the boiler may be made in the most approved manner and of ample size; but unless the attendant is careful to blow out these connections regularly, scale may stop them up and render accident inevitable.

A safety valve may be of the best make and of ample area, but unless it is tested regularly to determine its freedom, there is no certainty that it will respond to requirements when conditions arise that tend to produce an increase in the pressure beyond the safe limit.

The feed water used may be of excellent quality but very few sources of water supply remain absolutely constant, and unless frequent examinations of the interior surfaces of a boiler are made, there is no telling when scale or deposit may become a menace to safety.

The original supports of a boiler might be all that could be required from the standpoint of safety, but settings are subject to wear, and sometimes under forced conditions this wear is extremely rapid; only careful examination at frequent intervals renders a lug-supported boiler safe from accident due to change of position.

A small leak of steam coming from a seam may be a calking defect that can be remedied at leisure, but a careful examination to determine the exact nature of many such defects has shown that they were warnings of a cracked sheet, which, if neglected, would certainly lead to a disastrous explosion.

No matter how high a percentage in examination was given, what class of license was obtained, what kind of previous record a boiler attendant may have back of him or how complete or safe an equipment he may be operating, the fact still remains that his vigilance, or the lack of it, on the job is the biggest factor rendering boiler operation safe or otherwise.

Small Leaks

A steam consumption of sixteen pounds per brake horsepower-hour and an actual evaporation of eight pounds of water per pound of coal are considered a fair performance for the average plant. With coal having a heat value of 14,000 heat units per pound this would represent only about 9 per cent. of the latent energy of the coal available as mechanical energy at the flywheel of the engine—a very small proportion when one considers the numerous devices which have been introduced to improve the economy of the steam plant. The remaining 91 per cent. of the heat is lost in various ways; some passes up the stack with the waste gases; a large part is rejected in the exhaust from the engine, part of which is regained in raising the temperature of the feed water, but most of which is wasted by being given up to the condensing water, and a moderate amount of the heat is lost through radiation. In addition, there is the friction loss in the engine. These losses, however, cannot be eliminated. The best that can be done is to reduce them to a minimum by means of properly designed equipment and intelligent operation.

There are, however, other sources of waste which are entirely within the power of the engineer to avoid. These are the small leaks through blowoff valves, valves in the steam lines not in use, and the returns from neglected traps. These leaks, although apparently unimportant in themselves, nevertheless amount in the aggregate to a considerable sum in dollars and cents during a year; and in view of the very small thermal efficiency which is possible under the best conditions, any saving, however slight, is significant. A well designed plant does not insure economy; it is only a means toward that end, for after all the problem of economical operation is up to the engineer in charge.

Another Perpetual Motion Scheme Hunting for Capital

It does seem that there is no limit to human stupidity and credulity. In view of the many demonstrations which have been made of the futility of trying to get energy for nothing, one would suppose that the educated public would have better sense than to put money into that sort of a scheme. But every perpetual-motion "crank" whose chimera differs in any respect from those which have been exposed before seems to be able to hypnotize otherwise sensible persons into a condition of childlike credulity that would be highly amusing if it were not so serious a calamity.

The result of the latest effort to get something for nothing is labeled the "hydrodynamo." It is an aggregation of hydrostatic cylinders, chains, walking-beams and floats that would do credit to the genius of a mechanically-inclined lunatic but embodies a simple physical fallacy so obvious to a trained observer that we can only gasp with astonishment upon finding the thing indorsed by two men who are graduates of technical schools and have had several years of practical experience in engineering work. The "hydrodynamo," we are informed in a pamphlet issued by the Indiana Power Company, which appears to be the proud possessor of this ridiculous contraption, "runs anywhere without fuel," and the only loss is that of the friction of the parts, which amounts to "33½ per cent. of the gross effort" exerted by the machine, leaving "about 66.1 per cent. net efficient energy in the particular machine now being constructed." The slight arithmetical discrepancy in the promoter's figures is unimportant. Moreover, you could scarcely expect anyone absorbed in problems of "barodynic energy," "cognate forms," "paired antiforces," "transaxial vibrations" and such to descend to the plebeian level of a sum in subtraction; that sum was evidently "done" by the office boy.

The fallacy in everyday physics which the machine embodies will absolutely prevent it from working at all under the influence of any inherent force; it will never turn over, unless driven by some external source of power. We shall describe it in detail and point out the fallacy in an early issue. Meanwhile, we advise everyone who is invited to buy stock in the company to demand a demonstration of an actual machine under the unrestricted supervision of recognized experts before parting with a dollar or agreeing to part with one.

Inquiries of General Interest

Air Necessary to Operate Brake

A tank of 250 cubic feet capacity is charged with compressed air at 250 pounds gage pressure. How many times will this air operate two 8x10-inch brake cylinders taking the air at a pressure of 40 pounds?

L. J. K.

Assuming that a temperature of 60 degrees Fahrenheit is maintained and that there are no losses from clearance in brake cylinders and piping at 250 pounds pressure, the tank will contain

$$250 \times \frac{265}{14.7}$$

cubic feet of free air. At 40 pounds pressure the tank will have

$$250 \times \frac{55}{14.7}$$

cubic feet of free air. The quantity of air available for operating the brake will be the difference between

$$250 \times \frac{265}{14.7} \text{ and } 250 \times \frac{55}{14.7}$$

or 3570 cubic feet of free air. The volume of the two 8x10-inch cylinders is 0.582 cubic foot and at each application of the brake would use at 40 pounds pressure

$$0.582 \times \frac{55}{14.7} = 2.18$$

cubic feet of free air, and the brake can be operated as many times as 2.18 is contained in 3570, or 1640 times.

Vacuum Required to Lift Water

I have a triplex power pump 5x8 inches running 45 revolutions per minute supposed to deliver 93 gallons per minute, but it does not throw half of this amount. I put a new vacuum gage on the suction line and it registered 19 inches. Is this enough vacuum to lift the water 27 feet?

T. S.

Your figures and vacuum gage are wrong. If the pump filled the cylinders running at the speed given 93 gallons of water would not be delivered.

Water would rise to a hight of only 21½ feet in a 19-inch vacuum and it would take 23.8 inches to raise water 27 feet. It is probable that the pump cylinders fill to less than half their capacity.

Turning the Eccentric

If I should increase the angular advance of the eccentric what effect would it have on the cutoff, and would it have any effect on the piston?

V. K. S.

Increasing the angular advance of the eccentric simply means turning it farther ahead of the crank. If this is done all of the events in the valve cycle will occur earlier in the stroke than they did before.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Changing Speed of Engine

An engine is making 75 revolutions per minute. The pulley on the engine shaft is 12 inches diameter and the governor pulley is 8 inches diameter.

What changes are necessary to increase the speed to 85 revolutions per minute?

T. E. M.

To increase the number of revolutions per minute from 75 to 85 it will be necessary to decrease the size of the pulley on the engine shaft or increase the size of the one on the governor proportionally to the desired change in speed. The proper size of pulley for the engine shaft may be found by the proportion,

85:75::12:10.58

If the governor pulley is changed the proportion will be

75:85::8:9.06.

Exhaust Steam Jacket

As the exhaust from the cylinder is warmer than the air, would it not be economical to jacket a steam-engine cylinder with it?

E. R. T.

It would be the worst practice possible as the temperature of the exhaust is not so much below that of the air in the immediate vicinity of the cylinder and as its circulation will be so rapid it will carry away a great deal more heat than the air could. A dead air space protected by nonconducting covering makes the most economical cylinder jacket.

Work Done in One Stroke of Piston

The area of an indicator diagram is 3.47 square inches. Ratio of length of diagram to length of stroke is ⅙ and the scale of spring is 40; the diameter of cylinder is 16 inches. What is the work of the engine for one stroke?

W. J. K.

The area of an indicator diagram divided by its length gives the average hight, which multiplied by the scale of the spring will give the mean effective pressure per square inch of piston area. Assuming that the length of the diagram is 4 inches the mean effective pressure acting on a 16-inch piston will be

$$\frac{3.47 \times 40 \times 201}{4} = 6974.7$$

pounds. If the stroke of the piston is six times the length of the diagram it will be 2 feet and the work done in one stroke of the engine will be

$$2 \times 6974.7 = 13{,}949.4$$

foot-pounds.

Cooling Boilers

How would you cool off boilers, and why?

H. J. M.

The damper, fire- and ashpit doors should be closed and the boiler and its contents allowed to cool slowly without the admission of cold air to any part of the setting. If the water is let out of a hot boiler and cold air comes in contact with the shell and tubes while the brickwork is hot, there will be contraction stresses which may cause leakage and perhaps serious strains.

Washing Boilers

How often should boilers be washed out?

D. A. W.

Only experience will determine for each case. In the case of a plant which is new to the operator the boiler should be opened at the first opportunity, cleaned, and its condition noted. At what in his judgment is a reasonable time thereafter the boiler should again be opened, its condition compared with the former one and used as a basis for determining the frequency of future washings.

Effect of Keying

I have keyed the rod on my engine until the wedge has reached the top of the slot and can go no farther. Should I put shims behind the stationary or the moving half of the box, and will keying alter the length of this type of rod?

S. L. G.

Shims should in this case be next to the stationary half of the brass. If the wear is equal at both ends the rod length will not change as it will lengthen at one end as much as it shortens at the other.

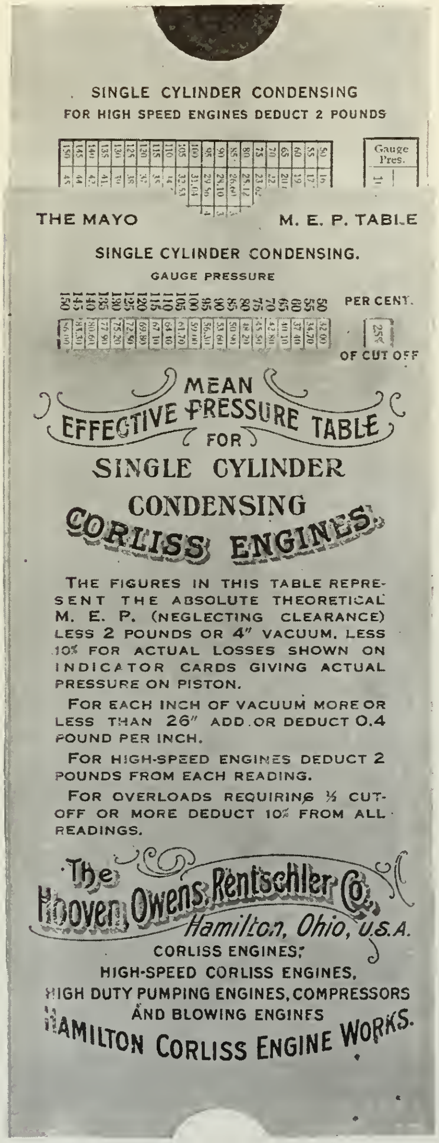
As the box at one end wears and the lost motion is taken up by moving the wedge, the pin is carried toward the end of the rod and, by increasing the distance between the pins, lengthening the rod. As the box at the other end wears, the pin is carried toward the middle of the rod by keying and the rod is shortened.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Mayo's M. E. P. Tables

It frequently happens that the engineer and others interested in the generation and transmission of power by steam desire to know quickly and without resort to cumbersome arithmetical methods what power may be expected from an engine of given dimensions operating under certain conditions of steam pressure, point of cutoff, back pressure, vacuum, etc., or the conditions under which the engine must operate and the load to be carried may be known and the selection of an engine of proper dimensions becomes the problem to be solved.



SINGLE CYLINDER CONDENSING

FOR HIGH SPEED ENGINES DEDUCT 2 POUNDS

Gauge Pres.

THE MAYO M. E. P. TABLE

SINGLE CYLINDER CONDENSING.

GAUGE PRESSURE

150	145	140	135	130	125	120	115	110	105	100	95	90	85	80	75	70	65	60	55	50	PER CENT.
86.00	83.30	80.60	77.90	75.20	72.50	69.80	67.10	64.40	61.70	59.00	56.30	53.60	50.90	48.20	45.50	42.80	40.10	37.40	34.70	32.00	25% OF CUT OFF

MEAN EFFECTIVE PRESSURE TABLE FOR SINGLE CYLINDER CONDENSING CORLISS ENGINES.

THE FIGURES IN THIS TABLE REPRESENT THE ABSOLUTE THEORETICAL M. E. P. (NEGLECTING CLEARANCE) LESS 2 POUNDS OR 4" VACUUM. LESS 10% FOR ACTUAL LOSSES SHOWN ON INDICATOR CARDS GIVING ACTUAL PRESSURE ON PISTON.

FOR EACH INCH OF VACUUM MORE OR LESS THAN 26" ADD OR DEDUCT 0.4 POUND PER INCH.

FOR HIGH-SPEED ENGINES DEDUCT 2 POUNDS FROM EACH READING.

FOR OVERLOADS REQUIRING ½ CUTOFF OR MORE DEDUCT 10% FROM ALL READINGS.

The Hooven, Owens, Rentschler Co.,
Hamilton, Ohio, U.S.A.
CORLISS ENGINES,
HIGH-SPEED CORLISS ENGINES,
HIGH DUTY PUMPING ENGINES, COMPRESSORS
AND BLOWING ENGINES
HAMILTON CORLISS ENGINE WORKS.

There are countless times that a short, exact and ready-at-hand method of making power calculations would be used and appreciated if available. Making mathematical computations is a tedious and purely mechanical process, from which most men naturally shrink, and when horsepower, capacity and related data are wanted, the data log of some engine builder is hunted through for the desired information.

Search through the pages of catalogs and engineers' reference books and stub-pencil arithmetic on the back of an envelop are very neatly "side stepped" by the use of the Mayo mean effective pressure table, lately brought out by Hooven, Owens, Rentschler Company, Hamilton, O.

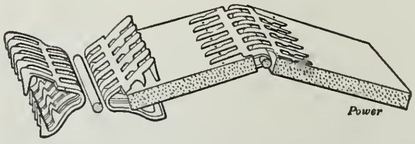
It consists of an oblong celluloid sheath with a transverse slot near the end, through which the mean effective pressure for any point of cutoff may be read from the sliding card inside the sheath.

Both sides of the sheath and the card are adapted for use, one for condensing and the other for noncondensing engines.

In the illustration the slide is set for a cutoff at 25 per cent. of the stroke and the mean effective pressure for any gage pressure from 50 to 150 pounds is instantly read through the slot.

Alligator Steel Belt Lacing

This belt fastener is adapted for use on leather, rubber, cotton or canvas stitched belting and consists of a separable lacing hinging on a rawhide or sectional steel rocker pin. The flexible joint insures quick contact with the pulley whether the belt is run with either or both sides in service, and as the lacing engages both sides of the belt alike the strain is thrown in direct line with the pull, making a joint which is easy on the texture of the belt.



Alligator Steel Belt Lacing

With rubber or textile belting this lace overcomes the tendency to fray at the ends as the members of the lace clamp the stubs so as to protect them. No punched holes are required, the teeth on each side of the lacing entering the belt on different lines of perforation; each prong has two teeth, the longer clinching through the belt and the shorter point on the opposite side giving additional strength. Both prongs enter the belt so that the pull is against the edge of the steel, thus avoiding weakening the belt. In rubber and textile belts the prongs do not cut the long threads in which lie the strength of that class of belting.

This belt lacing is manufactured by the Flexible Steel Lacing Company, 542 West Jackson boulevard, Chicago, Ill.

The Reiter Boiler Cleaner

This device, illustrated in Figs. 1 and 2, is intended to remove impurities from marine, tubular, or water-tube boilers, discharge them into a precipitator or settling chamber and return the purified water to the boiler. An upflow pipe is tapped into the boiler near the water level, where the ebullition is most rapid, and is connected with the top of a precipitating tank which provides a place for the accumulation of sediment, the purified water returning from the middle of the tank into the lower and cooler part of the boiler.

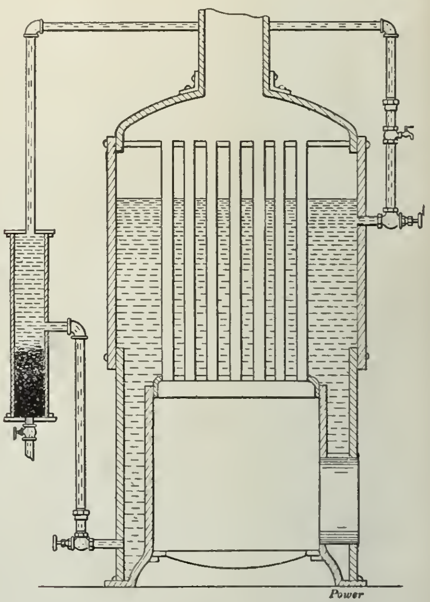


Fig. 1. Cleaner Applied to a Vertical Boiler

There are no moving parts to the system and all valves are either gate or angle valves, which interpose the least resistance to the circulation of water. An air cock suitably located is placed in the system at a convenient point above the water line to relieve the pipes of air when starting.

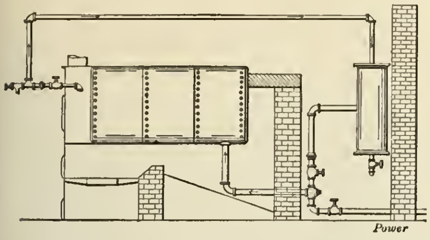


Fig. 2. Cleaner Applied to a Horizontal Boiler

It is claimed that this device will efficiently remove floating impurities, prevent the formation of scale and materially increase the general efficiency of a boiler installation. It is manufactured by the Reiter Boiler Cleaner Company, Elgin, Ill.

The Wattles Stoking Register

This device automatically produces a complete record, on a sensitized-paper chart, of the density of the smoke passing through a stack for every moment of each 24 hours' run.

The device serves three purposes: It records the time of stoking the furnaces, the amount of smoke passing to the stack and is responsible for a reduction in the amount of fuel consumed, because, knowing that a record is automatically kept of the smoke passing up the stack, the firemen are compelled to fire in a consistent, scientific manner.

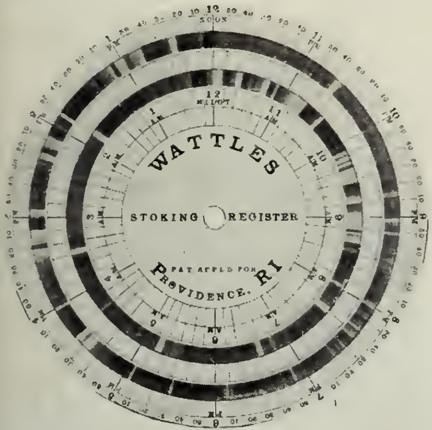


Fig. 1. Chart Showing Faulty Firing

The recorder consists of a box in which is placed an eight-day clock and a disk that fits on the shaft of the clock. On this disk a white sensitized-paper dial is placed, the dial being graduated into hours and minutes on two circles, as shown in Figs. 1 and 2.

The outer circle registers from 12 noon to 12 midnight; the inner circle from 12 midnight to 12 noon. The paper dial is put on the disk in a dark room and then placed in a tight case which admits no light. This case is fitted with a shutter which closes over a slot in the dial case, the slot being made the width of the recording circle. When the dial has recorded on the outer circle for 12 hours, the shutter is tripped automatically and the remaining 12 hours of the 24 are recorded on the inner circle through a second slot in the case. The device is placed and operated as follows:

On one side of the uptake, or chimney, as the case may be, is an incandescent lamp, so placed that its light will shine through a hole provided for that purpose. On the opposite side of the stack, or uptake, is another hole exactly in line with the first. The dial mechanism is placed so that the opening in the chart case will come in front of the hole. The light rays of the lamp will enter the slot in the disk case, and the sensitized paper, being exposed to the light, will turn dark when developed in the proper solution.

If nothing prevents the light rays from reaching the paper, a dark ring will show on the chart when developed, but if smoke passes between the lamp and the chart, the paper is not exposed fully to the light rays, and, consequently, it will not turn dark upon development.

The lamp is placed in a sheet-iron cone at the end away from the flue, so that it does not come in côntact with the smoke, soot and hot gases. The lamp is kept clean by means of air holes placed around it, which admit enough air to fulfil this purpose.

The recording machine is placed with the lens in a second cast-iron cone, and is kept clean by air entering the cone, as in the case of the lamp.

Referring to Fig. 1, which indicates very faulty firing, it may be seen that the stoking was not only irregular during the entire 24 hours' run, but that an excessive amount of smoke passed through the uptake, not only at frequent intervals, but for long periods. The dark sections of the inner and outer rings on the chart indicate a clean stack, while the light sections indicate excessive smoke.

Fig. 2 shows a contrast, and represents careful and systematic firing. But two smoky periods show during the 24 hours, one at 10.40 a.m. and one at 11.45 a.m.

The very dark mark on the outer circle of the chart is due to the back lash of the clock gearing in taking up after changing the dial, as the paper is exposed for a longer period than when the dial has begun to turn.

This device is also a check on the voltage of the lamp, for if the voltage is allowed to rise above normal, the exposed portion of the chart will be darker than the other section. If the voltage is low, the exposed section of the chart will be lighter during that period.

In order to obtain a percentage of the smoke periods, the sum of the white spaces, in minutes, is obtained and divided by the total time the chart has recorded. If a chart is satisfactory as to smoke percentage, and shows that the firemen are doing good work, it is filed; but if faulty, explanations are in order. With this chart recording the work of the firemen, a considerable saving in coal can be made, besides eliminating the smoke nuisance, as is shown by the report on coal consumption and electric energy for 1908-1909, in the Lauderdale building, Providence, R. I.

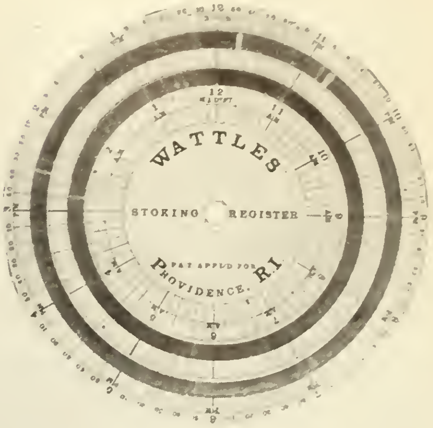


Fig. 2. Chart Showing Careful Firing

This plant has an equipment of three Babcock & Wilcox boilers of 125 horsepower capacity each, one 125-horsepower engine driving one 100-kilowatt generator, two 85-horsepower engines driving two 40-kilowatt generators. It is operated continuously and also furnishes steam for the operation of seven hydraulic elevators, of which five are for passengers and two for freight service. The average cost of coal for the years 1908 and 1909 was $3.32 per ton.

		Amount Saved.
Coal consumption, 1908.....	2,871.50 tons	
Coal consumption, 1909.....	2,638.18 tons	
	233.32 tons at $3.32 per ton	$ 774.62
Removal of ashes, 1908......	6,507 cans	
Removal of ashes, 1909......	5,509 cans	
	998 cans at $0.05 per can	49.90
Electric energy, 1908......	281,503,882 watts	
Electric energy, 1909......	289,715,584 watts	
	8,211,762 watts increase for 1909	
Estimated cost of producing increased electric energy for 1909:		
55 tons of coal at $3.32 per ton..............		182.60
Total saving for 1909......................		$1,007.12

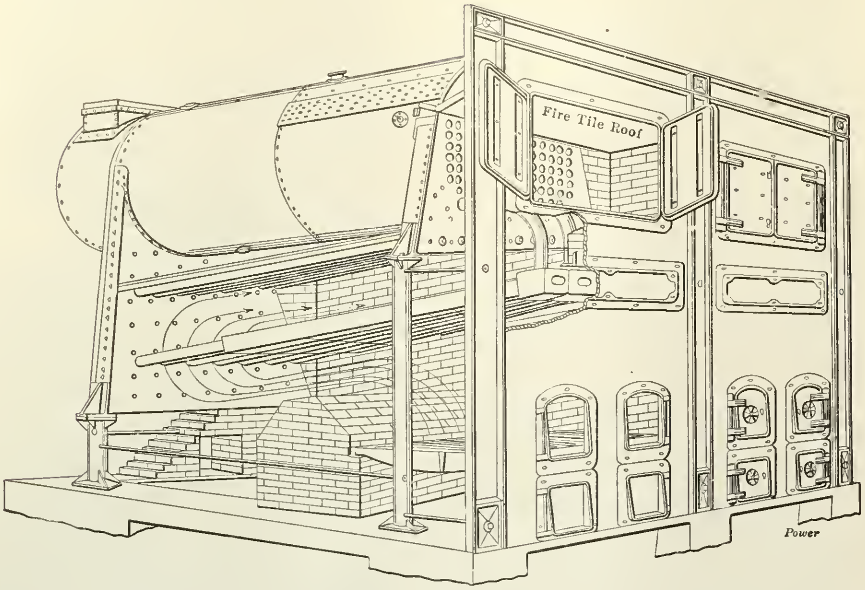
The Wattles stoker register has been in operation at this plant during 1909, saving 10½ per cent. For the first six months of 1910, it is stated that the coal consumption has remained practically unchanged, as compared with 1909. The tons of coal consumed were 1281.76, as against 1208.63 in 1909, and during this period the plants have furnished 11,000,000 additional watts.

The recorder is made by the Wattles Stoking Register Company, Butler Exchange, Providence, R. I.

The Improved Hawkes Boiler

The accompanying sketch shows in perspective the latest features of the Hawkes boiler, the essential difference from former designs being in the front head, which extends flush with the end of the boiler and forms the water leg into which the upper bank of tubes is expanded. This difference in design increases the circulating area and facilitates the steaming action. The lower bank of tubes is covered, as before, with a substantial interlocking fire-clay roofing, forming a reverberatory arch over which the hot gases must pass. A distinctive feature of the design is the independent connection of each lower tube with the upper shell, which allows expansion and contraction to take place in each tube independently.

One of the chief advantages attained by the new construction of the front water leg lies in the substantial support of the front end of the boiler which rests on cast-iron columns entirely independent of the brickwork. The shell itself does not come in contact with the fire and no joint seams or handholes are exposed to the hot furnace gases. All openings are accessible from the outside where repairs can easily be made. The furnace arrangement makes the boiler practically smokeless under any reasonable conditions of firing. The tubes may be blown in the direction of the draft. This boiler is now built by the Hawkes Boiler Company, Marquette building, Chicago, Ill.



Showing Improvements in the Hawkes Water-tube Boiler

An Unloading Car Chute

A device for loading wagons from coal cars is illustrated herewith. It is made by the Quick Unloading Car Chute Company, Birmingham, Ala.

The device is simple and its construction will be at once comprehended from the illustrations. Fig. 1 shows a wagon loaded, ready to be hauled away. The advantages that come from this chute are that it saves the team from waiting while the shovelers in the car do their work and it enables them to work continuously instead of waiting for the arrival of a team. There being no waiting for the wagon, or wagons waiting for loads, one-third more work can be done with a given number of teams.

Fig. 2 gives an idea of the construction of the chute. It is made of sheet-steel angle iron and is hung to the top of the car by two large hooks, which may be attached to the bottom frame of the chute at three different points, according to the size of the wagon to be loaded. The chutes are adjusted by means of a chain, by the hooking up of which the front of the chute is raised, while by letting it down the chute is given a greater inclination.

The chute can be used on box cars as well as on gondolas, and can easily be handled by two men. By using two chutes in a car and dividing the distance, the shoveler can unload the car without moving the chutes.

By using this device a wagon will be detained only one minute and can then start on its return trip. If extra large wagons are used, holding more than one and a half cubic yards, the contents of two chutes can be placed in each wagon, as was done in the case illustrated in Fig. 1. The driver of the team adjusts the bolt which locks the door and discharges the load from the chute into the wagon, then leaves the door relocked and ready for the chute to be reloaded. When cars are regularly unloaded from private tracks, the chutes can be permanently mounted on supports at the side and the car placed in a position for unloading. In plants where the coal must be shoveled from the car, the chute should prove to be a great time saver.

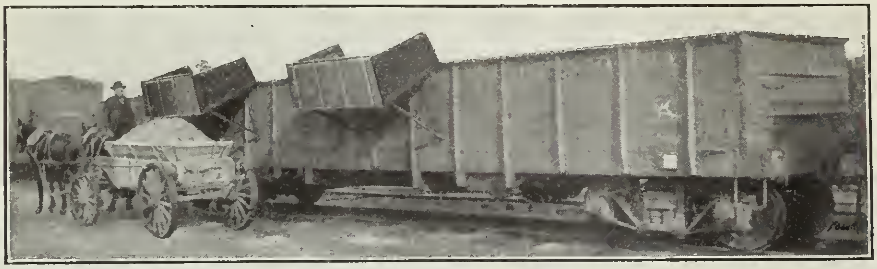


Fig. 1. Wagon Loaded in One Minute

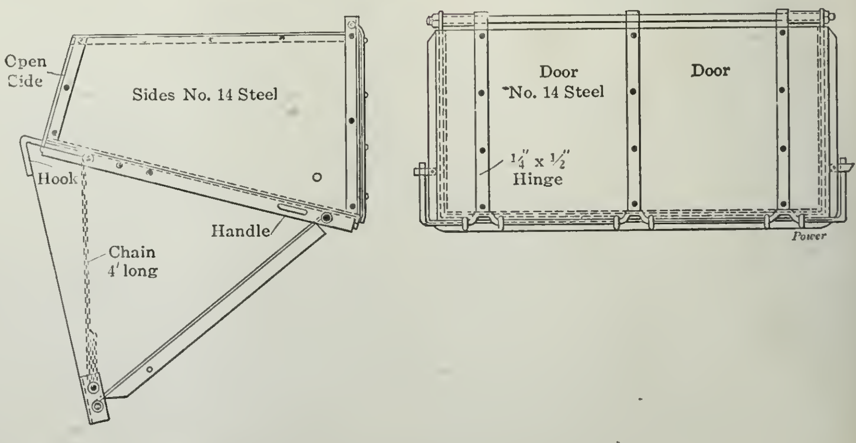


Fig. 2. Construction of Coal Chute

PERSONAL*

C. H. Hughes & Co., mechanical and electrical engineers, formerly at 115 Broadway, New York City, have moved their quarters to larger offices at 64 Wall street.

R. W. Wells, who has been assistant for the last two years to A. E. Cluett, industrial engineer, for the Cluett, Peabody Company's factories at Troy, has been appointed master mechanic of these factories. Mr. Wells was formerly chief civilian engineer at the United States Navy Yard at Mare Island, California, under Naval Constructor Holden A. Evans.

SOCIETY NOTES

On June 18 the first steps were taken to form a permanent organization to be known as the American Society of Engineer Draftsmen, embracing every branch of the profession, including mechanical, electrical, civil, architectural, marine, sanitary, automobile and aëronautical draftsmen. The first meeting of the society was held on July 27. The formation of this organization, which was conceived by E. Farrington Chandler, designer and inventor, marks the first effort to form a national society among draftsmen.

The officers chosen are E. Farrington Chandler, president; William B. Harsel, vice-president, and Henry L. Sloan, secretary and treasurer, with headquarters at 116 Nassau street, New York City.

NEW PUBLICATIONS

How to Sample Coal and Coke is the title of Bulletin No. 4, by E. G. Bailey, Fuel Testing Company, Boston, Mass. It is issued for the purpose of assisting those interested in securing representative samples of coal and coke. The information published therein relative to this subject represents many years of experience gained in the taking of and analyzing thousands of samples. It is necessary to read the first chapter of this bulletin, which deals with the fundamental principles of sampling, before taking up the other chapters. Then, whatever succeeding chapter bears on the conditions under which samples are to be taken should be read and thoroughly mastered. These chapters deal with sampling at the mines, railroad cars, vessel cargoes, boiler plants and store piles, besides sampling coke, ashes, etc. Chapter 12 deals with the shipment of samples and tells the reader how to ship the samples to the chemist. The bulletin is sent free upon application.

Universal Craftsmen Convention

The eighth annual convention of the Universal Craftsmen Council of Engineers was held at Buffalo, N. Y., August 1 to 5, and was decidedly the largest and most successful in the history of the organization. The business meetings were held at the Iroquois hotel, the headquarters of the gathering, and a spacious parlor in this building was suitably arranged for a splendid exhibit of appliances used by engineers.

The preliminary exercises of the convention were largely attended. The meeting was called to order at eleven o'clock Tuesday morning by Joseph N. Nichols, chairman of the local committee. After an invocation by Rev. Frederick H. Coman, Mr. Nichols delivered a brief address, and then introduced John Sayles, representing Mayor Fuhrmann, who welcomed the delegates and guests to Buffalo, Grand Worthy Chief Walter S. Caldwell responding. Addresses were also made by Congressman D. S. Alexander, Robert Ingersoll and Herbert E. Stone.

At the conclusion of the addresses a memorial service was held for members who had passed away during the year. A. M. Wickens read the service, and a quartet sang "The Lord's Prayer" and "Lead, Kindly Light." The committees were then appointed and the meeting adjourned.

At the several succeeding sessions of the delegates considerable important business was transacted, and at the closing meeting the following grand officers were elected and installed: John Cope, worthy chief; Thomas H. Jones, assistant worthy chief; J. U. Bunce, secretary; John MacMorran, treasurer; Henry C. Senn, warden; P. E. Crumrine, guard; Charles F. Seigrist, [illegible]bert Simpson, William H. Armstrong, board of trustees; John Cope, Walter S. Caldwell, John MacMorran, W. J. Ranton, Thomas H. Jones, board of governors.

There were many features of entertainment connected with the convention, including automobile rides to places of interest about the city, a visit to the Larkin Manufacturing Company, a steamboat ride to Fort Erie, a visit to Niagara Falls, a steamboat excursion on Lake Erie to Crystal beach, where a fish dinner was served, and a theater party to Shea's vaudeville house for the ladies.

The closing social event was a banquet at Statter's restaurant on Friday night. After the dinner was served, Joseph N. Nichols, the toastmaster, introduced the following gentlemen who made short addresses: Colonel Francis Ward, John Cope, Robert Ingersoll, John Moore, W. M. Williams, Joseph N. Gregory and Herbert E. Stone. Interspersed between the speeches the following entertainers appeared: Anna Batten Edwards, soprano; Frank Corbett, tenor, Consolidated Valve Company; C. F. Huie, vocal selections, Buffalo Refining Company; "Herb" Wunder, pianist, the Lunkenheimer Company; "Billy" Murray, songs, Jenkins Brothers, and "Jack" Armour, humorist, POWER.

The attendance of ladies was unusually large, and their interests were taken care of by the ladies local committee, headed by Mrs. Joseph N. Nichols.

The following firms had representatives present: The Strong-Carlisle-Hammond Company, the Purimachos Works, Oldman Boiler Works, POWER AND THE ENGINEER, Lagonda Manufacturing Company, Lake Erie Boiler Compound Company, Crane Company, Leroy Chemical Company, the Lunkenheimer Company, C. E. Squires, Buffalo Mill Supply Company, the McLeod & Henry Company, Skinner Engine Company, General Electric Company, Jones Under-Feed Stoker Company of America, *Southern Engineer*, Frontier Engine Company, John Knepper & Son, the Whittet-Barrett Company, Colza Chemical Company, Manzel Brothers, Greene, Tweed & Co., the Powell Valve Company, George W. Knowlton Rubber Company, the Buffalo Refining Company, Ashton Valve Company, Dearborn Drug and Chemical Works, Keystone Lubricating Company, Anderson Trap Company, International Steam Pump Company, Home Rubber Company, Joseph Dixon Crucible Company, Jenkins Brothers, Farrar & Trefts, McCarthy Brothers and Ford, Garlock Packing Company, Stebbins Oil Company, H. W. Johns-Manville Company, Liberty Manufacturing Company, Robertson Cataract Company, Ziermore Regulator Company, Buffalo Belting Company, Tashenburg Brothers, William A. Case & Son, Buffalo Brewers' Supply Company, Timmins & Butler, Hills-McCanna Company, Ruckel & Son, Standard Oil Company, Kempner Boiler Compound Company, Carl W. Schantz, Warren & Dellwardt Company, Howard Brothers Boiler Works, R. & P. Coal and Iron Company, Pomeroy Stelrecht, Weger Machine Company, Howard Iron Works, Hoffman Oil Company, Commercial Oil Company, Penn Oil and Supply Company, Cling Surface Company, Murphy Iron Works.

It was decided to hold the next annual meeting in Philadelphia, Penn., during week commencing Monday, August 7.

A Marvelous Accident

According to *The New York Times*, the Hudson river steamboat "Mary Powell" met with a very unusual—not to say astounding—accident the other day. The newspaper reports that "the crosshead which fastens the piston head to the slides in which it moves broke." Undoubtedly this was caused by the triple-expansion crank pin becoming fouled with the boiler eccentric and thereby straining the water-leg of the off cylinder at the base of the air pump.

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*Items for this column are solicited.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

REVERSING GAS ENGINE. Daniel Schlegel, Dryville, Penn. 964,564.

ROTARY ENGINE. Emil F. Chader, Pearl River, and Vincent Anderson, Arlington, N. Y. 964,754.

POWER - GENERATING APPARATUS. Frank W. Lierman, Milwaukee, Wis.; Augusta Lierman, executrix of said Frank W. Lierman, deceased. 964,923.

ENGINE. Joseph D. Ferry, East McKeesport, Penn. 965,094.

INTERNAL COMBUSTION ENGINE. Hugo C. Gibson, New York, N. Y. 965,135.

BOILERS, FURNACES AND GAS PRODUCERS

FURNACE. Charles Sinclair Drummond, London, England. 964,502.

FURNACE. Hermann Kurtenacker, Berlin, Germany. 964,656.

BOILER SETTING. George B. McLeod, Windsor, Ontario, Canada. 964,931.

VAPOR BURNER. Emery M. Byers, Belvidere, Ill. 965,074.

GAS PRODUCER. Roscoe P. Conkling, San Luis Potosi, Mexico. 965,086.

POWER PLANT AUXILIARIES AND APPLIANCES

COOLING SYSTEM FOR INTERNAL COMBUSTION ENGINES. Thormod Odee, Beloit, Wis., assignor to Thormod Odee and A. S. Koto, copartners, doing business as Odee & Koto Manufacturing Company, Beloit, Wis. 964,542.

MEANS FOR INCREASING THE ENERGY OF STORED COMPRESSED AIR. William Horace Sodeau, Newcastle-upon-Tyne, England, assignor to Sir W. G. Armstrong Whitworth & Co., Ltd., Newcastle-upon-Tyne, England. 964,574.

HOSE COUPLING. John H. Stephens, Vernon, Tex. 964,578.

SIGHT-FEED LUBRICATOR. Floyd L. Swanberg, Cincinnati, Ohio, assignor to the D. T. Williams Valve Company, Cincinnati, Ohio, a Corporation of Ohio. 964,583.

OIL-FEEDING MECHANISM. Guido Fornaco, Turin, Italy, assignor, by mesne assignments, to F. I. A. T., Poughkeepsie, N. Y., a Corporation of New York. 964,637.

FLUE THIMBLE. Ralph Hall, Leadwood, Mo. 964,642.

STARTING MECHANISM FOR GAS ENGINES. Atwood B. Keyes, Henniker, N. H. 964,649.

VAPORIZER FOR HYDROCARBON ENGINES. Elden P. Lamb, Camden, Me., assignor to Camden Anchor-Rockland Machine Company, Rockland, Me. 964,657.

FEED-WATER CONTROLLER. Reuben M. Reynolds and Benjamin F. Page, South Bend, Ind. 964,690.

EXHAUST-VALVE OPERATING MECHANISM FOR EXPLOSIVE ENGINES. Walker H. Spayd and Fred M. Spayd, Van Wert, Ohio. 964,706.

CARBURETER. Robert E. Wynn, Brownsburg, Ind., assignor to himself, and John W. Hayden, Indianapolis, Ind., a Copartnership. 964,831.

CONDENSATION METER. Harry A. Austin, Lockport, N. Y. 964,837.

DAMPER - REGULATING MECHANISM. Benjamin C. Wickes, Auburn, N. Y. 964,892.

BOILER FEEDER. August Junger, Metz, Germany. 964,916.

HIGH-PRESSURE TURBINE PUMP. Walter L. Forward, West Berkeley, Cal., assignor to Byron Jackson Iron Works, West Berkeley, Cal., a Corporation of California. 964,963.

COMPRESSOR. John T. Love, Charlotte, N. C., assignor to William H. Flint and James W. Conway, Charlotte, N. C. 964,987.

INDICATOR FOR PRESSURE GAGES. Robert E. Fields, Fort Worth, Tex. 964,634.

COMPRESSOR AND VACUUM - PUMP VALVE. Charles Wainwright, Erie, Penn. 965,052.

PRESSURE-CONTROLLING DEVICE FOR BOILERS. John Daniel Walsh, Lockport, N. Y. 965,053.

FLUID-PRESSURE REGULATOR. William E. Beech, Ferris, Tex. 965,064.

PIPE COUPLING. James Clark, Bradford, Penn., assignor to S. R. Dresser Manufacturing Company, Bradford, Penn. 964,609.

ELECTRICAL INVENTIONS AND APPLICATIONS

SYSTEM OF MOTOR CONTROL. Ernest L. Gale, Sr., Yonkers, N. Y., assignor to Otis Elevator Company, Jersey City, N. J., a Corporation of New Jersey. 965,175.

MOTOR CONTROLLER. Herbert W. Cheney, Milwaukee, Wis., assignor to Allis-Chalmers Company, a Corporation of New Jersey. 965,280.

ELECTRIC WATER HEATER. Frank A. Robinson, Albany, N. Y., assignor to William G. Van Loon, Albany, N. Y. 965,333.

AUTOMATIC POTENTIAL REGULATOR FOR DYNAMOS. John J. Mueller, Jr., Peru, Ill. 965,396.

ELECTRIC MOTOR. Frank L. Sessions, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 965,543.

ELECTRIC PUMP GOVERNOR. Charles W. Townsend, St. Louis, Mo., assignor to the Westinghouse Air Brake Company, Pittsburg, Penn., a Corporation of Pennsylvania. 965,609.

CONTROLLING MEANS FOR ELECTRIC MOTORS. George H. Whittingham, Baltimore, Md., assignor to Monitor Manufacturing Company, of Baltimore, Md., a Corporation of Maryland. 965,674.

MOTOR STARTER. George H. Whittingham, Baltimore, Md., assignor to Monitor Manufacturing Company, of Baltimore, Md., a Corporation of Maryland. 965,675.

ELECTRIC-LIGHT FIXTURE. Edwin E Hayward, Waterbury, Conn., assignor to the Plume & Atwood Mfg. Co., Waterbury, Conn., a Corporation of Connecticut. 965,710.

ELECTRIC CABLE. William Geipel, Southwark, England. 965,810.

ARC LAMP. George M. Little, Pittsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 964,664.

BRACKET FOR COMBINATION ELECTRIC AND GAS FIXTURES. James D. Coder, Elizabeth, N. J. 964,493.

STARTING DEVICE FOR ELECTRIC LAMPS. George M. Little, Pittsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 964,665.

ELECTROMAGNET PROPELLING SYSTEM. Rexford Dean, Findlay, Ohio. 964,498.

HOT - WIRE ELECTRICAL MEASURING INSTRUMENT. John T. Irwin, London, England. 964,521.

ELECTRIC SWITCH. William S. McLewee, Yardley, Penn. 964,536.

INSULATOR. John Russell Tufts, Jr., Weymouth, Mass., assignor of one-half to F. W. Clark, Boston, Mass. 964,586.

FUSE. Frederick H. Weston, Jr., Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 964,592.

ALTERNATING-CURRENT MOTOR. Per Utne, New York, N. Y., assignor to the Union Switch and Signal Company, Swissvale, Penn., a Corporation of Pennsylvania. 964,714.

STORAGE-BATTERY INDICATOR. Paul Max Marko, New York, N. Y. 964,994.

STORAGE BATTERY. Paul Max Marko, New York, N. Y. 964,994.

PROCESS OF PURIFYING WATER. John E. Angell, St. Louis, Mo. 965,148.

CHECK VALVE. Emmet P. Gray, Detroit, Mich. 965,178.

BELTING. Varney K. Sturges, Detroit, Mich., assignor to Joseph H. Schulte and Harry Kaiser, Detroit, Mich. 965,250.

METALLIC ROD PACKING. Charles L. Trapp, South Omaha, and Otto C. Gehrman, Omaha, Neb., assignors of one-fourth to Frank J. Meyers, Omaha, Neb. 965,255.

POWER PLANT TOOLS

LIFTING JACK. Robert M. Fraley, Champaign, Ill. 964,905.

BALL-BEARING JACK SCREW. Oscar E. Tidd, Georgetown, Mass. 964,824.

RATCHET DRILL. Domenic Cortese, Spooner, Wis. 964,616.

INSULATION-CUTTING PLIERS. Harry A. Adams, East Hartford, Conn. 964,600.

COMBINATION TOOL. Norman McAslan, Yorkshire, Iowa. 964,532.

UNIVERSAL SQUARE. De Witt C. Meeker, Carthage, Mo., Mary Ann Meeker, administratrix of said De Witt C. Meeker, deceased. 965,312.

DIE STOCK. Herman W. Oster, Cleveland, Ohio, assignor to the Oster Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 965,320.

DIE STOCK. Herman W. Oster, Cleveland, Ohio, assignor to the Oster Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 965,403.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., V. Bunce, Hotel Statler, Buffalo, N. Y Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

Vol. 32 NEW YORK, AUGUST 23, 1910 No. 34

MOISTURE in steam is usually caused by particles of water being carried over with the steam bubbles during evaporation, or by condensation in the steam pipes and cylinders.

This moisture is inert, doing absolutely no work and in some cases doing harm.

In small quantities its effect is not noticeable upon the reciprocating engine, except that it increases the steam consumption; but with a steam turbine it causes friction and a pitting action upon the blades.

Separators and superheat are employed to overcome the moisture in the steam, and are a material aid toward that end, but to intelligently cope with this difficulty it is necessary to know the quality of the steam entering the engine or turbine.

Here is where the calorimeter is invaluable.

Like the steam-engine indicator, the calorimeter was formerly regarded by the operating engineer as a laboratory instrument or, at the best, a device to be used only by experts in conducting tests upon a plant. But the calorimeter is now generally regarded as a necessary adjunct to the engine room.

Not only is it indispensable in testing engines and boilers, where its omission would entail erroneous results, but it furnishes a convenient means for determining at any time whether the boiler is supplying sufficiently dry steam.

The accuracy of a calorimeter depends upon the manner in which it is attached to the steam pipe. Many an instrument has been unjustly condemned when the fault lay with the man who connected it to the steam pipe.

The quality varies in different portions of a pipe; this variation being greater in a horizontal than in a vertical pipe. Steam drawn from a point near the inside surface is apt to contain more moisture than that taken from the center of the pipe.

One of the best methods of obtaining a fair sample of steam is to cut a long thread on a ½-inch nipple, into which a series of ¼-inch holes have been drilled. Let this nipple project into the steam pipe about three-fourths of the diameter, and as near as possible to the point at which the quality is desired. Care should be taken to place the calorimeter close to the steam pipe to avoid radiation losses, and an ample flow of steam should be allowed through the instrument.

More accurate methods have been devised in the form of sampling tubes, which are more or less complicated, but for all practical purposes the above method will suffice.

There are several types of calorimeters which are divided into classes depending upon their mode of operation. These are the condensing, separating, superheating and electrical calorimeters. Each has its field of application, but the best makes of the three types last mentioned agree very closely as to results, and apparently are much more reliable than one of our contributors will admit.

It may be worth while to "dig up" on this subject and become as familiar with the calorimeter as most engineers are with the indicator.

The Manufacture of Coal Briquets

By A. R. Maujer

Coal dust is dried, ground to uniform size, mixed with a suitable binder, pressed into briquets and baked to remove objectionable odor. The entire operation is continuous and automatic.

The difficulty of burning coal in the form of dust, especially in the cities, where smoke is objectionable, is so great as practically to prohibit its use. Until a comparatively recent date the price of coal in convenient form has been sufficiently low to make it unprofitable to spend any time or effort in making coal dust available for use. As a consequence young mountains of culm have grouped themselves about the mouths of the coal mines to remain there as monuments of man's prodigality of nature's store. As the price of coal advanced, due to the increase in the rate of diminution of the known supply, the increase in the demand and the rise in the costs of production and transportation, the efforts to make coal dust available for use were stimulated in proportion to the advance.

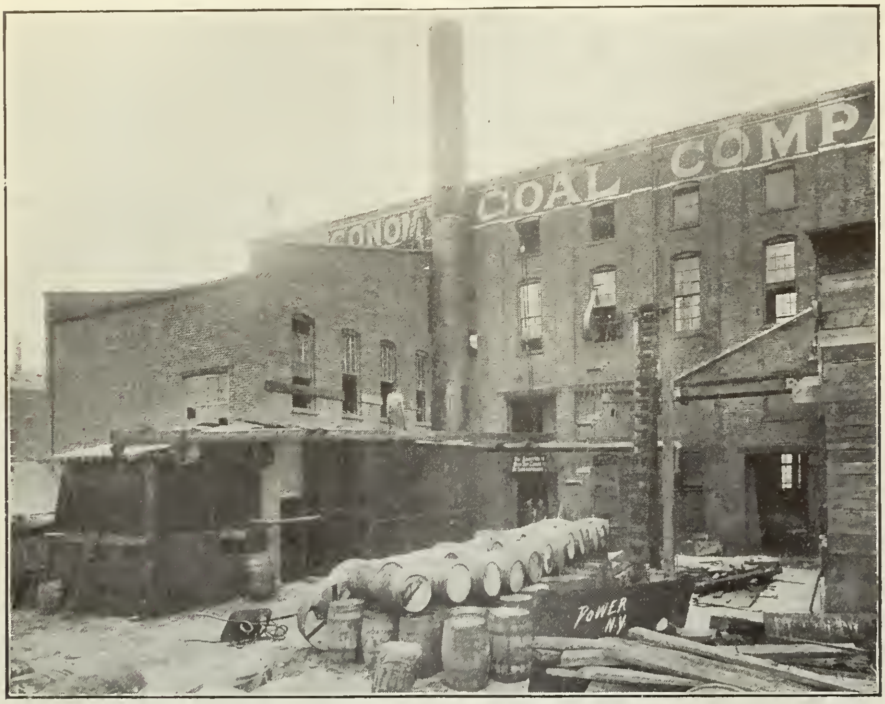


FIG. 1. GENERAL VIEW OF THE PLANT

An obvious way in which to make culm suitable for use, is, of course, to mold it into lumps or briquets of desirable size. This is done by mixing with the culm some suitable material to cement or bind the particles of coal together and pressing the mixture into the desired shape.

The chief difficulty usually encountered was in securing an efficient binder. A suitable binder should be capable of forming briquets which will not break easily in handling, should have no bad effects upon the nature of the ash, should not diminish unreasonably the heat value of the finished product, should not cause smoke or difficulty in combustion and, preferably, should not cause the briquets to have any undesirable odor.

The Economy Coal Company, of New York City, of which brief mention was made in the June 21 issue of POWER, holds the patents issued to Miss Pauline Grayson, the inventor of a briquetting process and a binder of special composition which is said to possess the qualities just enumerated.

The No. 1 plant of the company is located at the foot of West Forty-eighth street on the river front. A view from the river is given in Fig. 1. The wooden bin, a corner of which may be seen to the right, contains the coal dust. To the left of the dust bin a new storage bin for the finished briquets is to be erected. This is to be of concrete and steel construction.

The small addition at the left contains the boiler and engine rooms. The main engine is a 175-horsepower Corliss. The two steel tanks to be seen at the sides of this addition are the receiving tanks for the binder.

At the left of Fig. 2 is shown the grinder, and in the background the drying kiln, extending across the room and along the left wall to the inclosed bucket elevator which may be seen at the far side of the grinder. A screw conveyer in the floor of the dust bin pushes the coal dust onto a short bucket elevator which empties into a hopper located on top of the kiln just beyond the limit of the view at the right. The hopper discharges onto a belt conveyer which carries the culm to the aforementioned elevator which discharges it to the grinder through the 45-degree chute, as shown. The temperature in the kiln is maintained at about 400

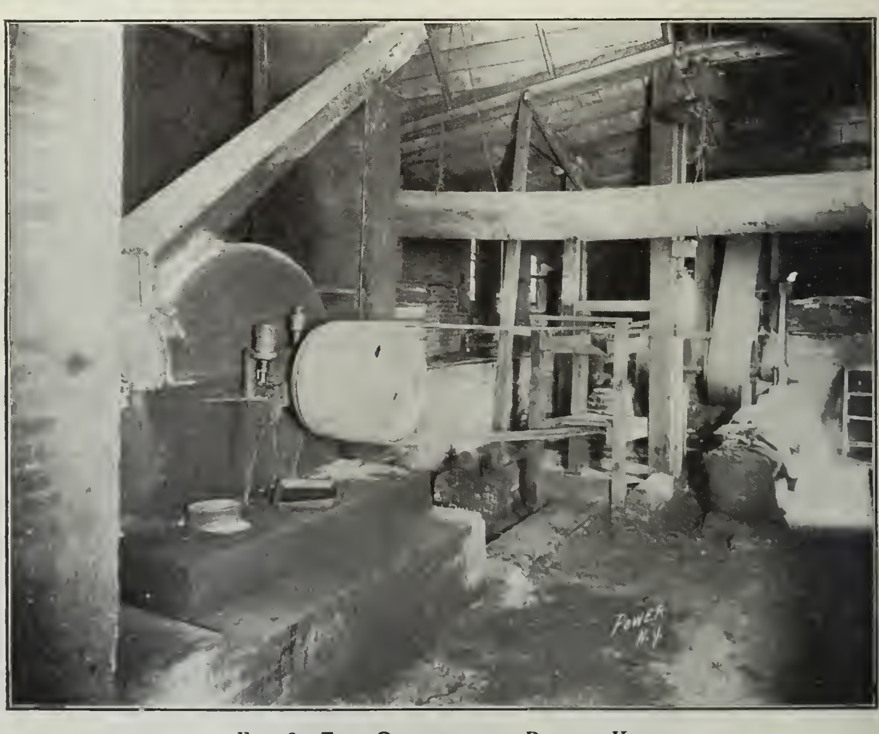


FIG. 2. THE GRINDER AND DRYING KILN

degrees Fahrenheit, so that the excess of moisture in the dust is all driven off by the time it reaches the grinder. In the grinder all of the lumps are reduced so that they will pass through a screen having an 1/8-inch mesh.

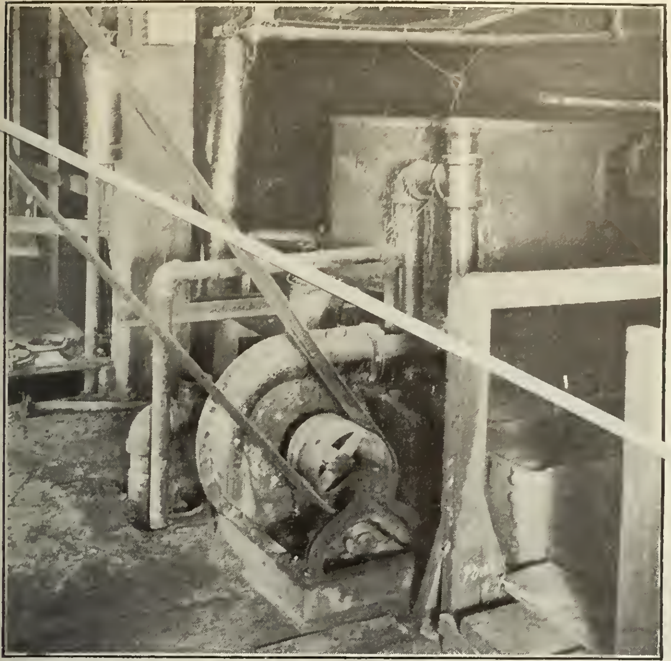


Fig. 3. Distributing Tanks and Pump

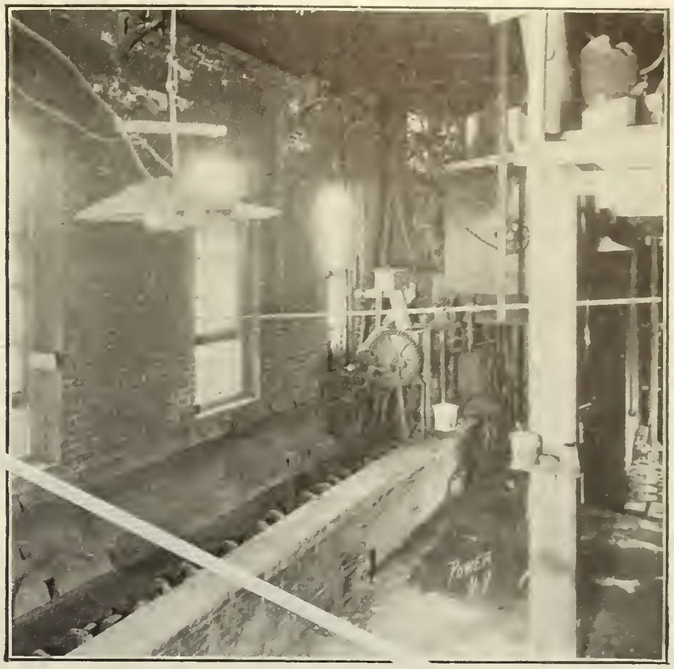


Fig. 4. The Mixing Trough

A second bucket elevator, which is hidden behind the brick pier at the extreme left of the view, raises the culm to the first of the two mixing troughs in which the binder is added.

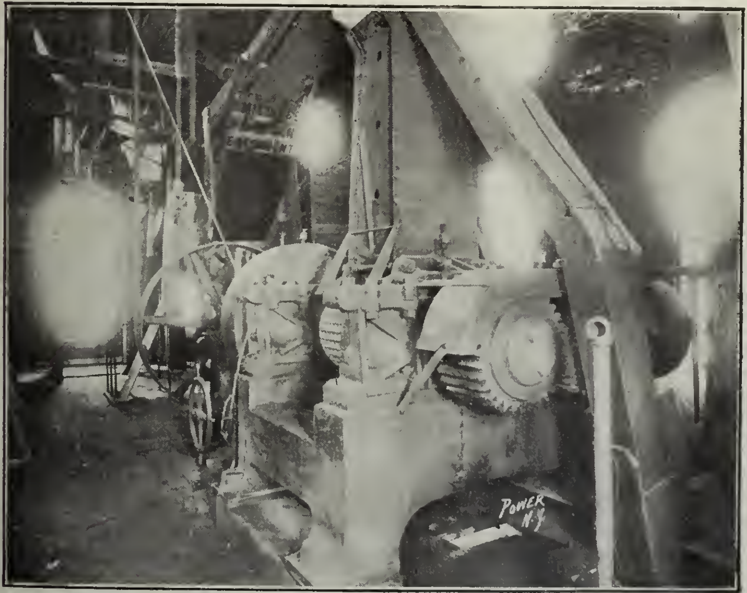


Fig. 5. One of the Presses

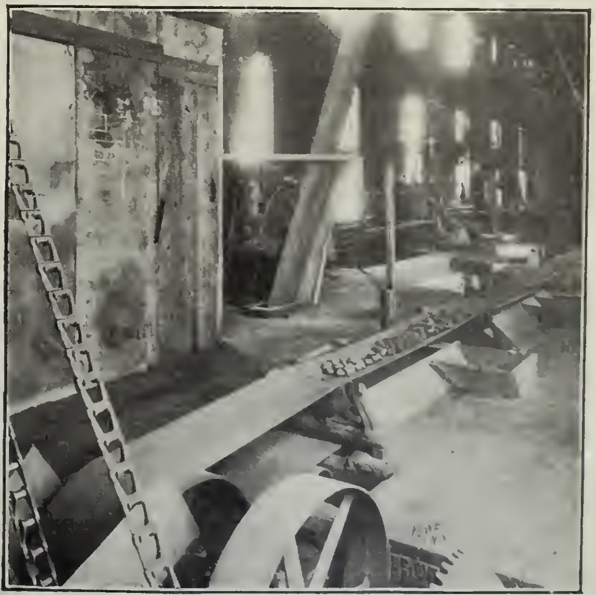


Fig. 6. Finished Briquets on the Conveyer

Fig. 3 shows the mezzanine floor which is located just above the upper trough. A portion of this floor may also be seen at the upper right-hand corner of Fig. 4. The centrifugal pump seen in Fig. 3 pumps the binder from the receiving tanks to the distributing tanks shown in the same view and in Fig. 4.

The mixing troughs are shown in Fig. 4. The upper one is 10 feet long. The lower one, in which the bulk of the binder is added, is 40 feet long. The cover of this trough has been removed to afford a view of the mixing blades.

This trough empties the mixture of dust and binder into a pit from which it is raised by means of a bucket elevator to the floor above, upon which the presses are located.

One of the presses is shown in Fig. 5. The mixture passes along on a belt conveyer and drops down through the sheet-iron chute above the press to the rollers which press it into briquets. Part of one of the rollers may be seen in the middle of the press just beneath the chute.

The briquets fall onto the belt conveyer which passes under the press and which may be seen in Fig. 6. [illegible] this conveyer the briquets are raised to another one on the same floor but at a higher level and traveling in the opposite direction. This conveyer empties the briquets into a hopper which passes them into the oven in which they are baked for about 15 to 20 minutes. The object of the baking is to deodorize the briquets and drive off the smoke producing volatile matter.

As the briquets leave the hopper they fall onto a wide, slowly moving conveyer made up of lengths of ¼-inch wrought-iron pipe spaced 1¼ inches apart. The oven is located directly over the boilers in the power house. The waste gases from the boiler furnaces are passed through the oven and furnish the heat to do the baking. Falling off the conveyer at the end, the briquets pass through the outlet hopper onto another conveyer which distributes them in the storage bin.

The entire operation is continuous and automatic. Except for getting the culm out of the barges, no hand labor is required in the process. Adjusting devices are located at various points so that the speed of each step can be synchronized with all of the others and the entire system can by this means be kept at its most efficient working speed. The present capacity of this plant is about 12 tons per hour.

Briquets can be made from any kind of coal dust. At present, however, the principal product used is Pennsylvania anthracite dust, of which large quantities are available at a reasonable charge for both coal and freight.

We are indebted to Dr. L. H. Thomas, secretary of the company, for his courtesy in extending to us the priviledge of publishing this description. We acknowledge the assistance given and courtesy shown by Charles A. Hercht, general superintendent of the company.

Water in Steam May Cause Trouble

By W. H. Booth

Arrange the boiler layout with the stop valve near the crown, slope the piping toward the engine and avoid the use of all drains but the one from the separator near the engine.

If a steam pipe has a slope back to the boiler, it is likely to be at a small gradient, only, and the effect of gravity which tends to send back to the boiler any water formed in the pipe is more or less counteracted by pipe roughness. The water flows backward with its upper surface moving faster than its lower part, and the movement is against the direction of flow of the steam. Unless the flow of steam is very slow it is practically impossible for the water to flow down the small descent to the boiler. Some of it is sure to be whisked up by the steam and carried toward the engine, and ultimately there will come a time when a plug of water will be projected forward to the engine, of a size large enough, perhaps, to do mischief. Both experience and reason point to the system of sloping pipes toward the engine, thereby helping the flow of steam to hurry forward all wetness into the separator. So much is this the case that the boiler stop valve should be placed at the highest part of the steam range. This means that the valve must be higher than the engine and that the valve must stand at the head of a pipe branching out vertically from the boiler. Some condensation must occur in this length, but it has the full aid of gravity to return it to the boiler, and if any of it is carried off by the steam it passes through the valve and thence descends to the engine.

When the valve is closed it is free from water. If placed close down to the boiler, with a vertical branch between the valve and the steam main, then when the valve is shut the pipe becomes full of water above the valve. When the valve is opened, the water may be projected violently toward the engine along the empty and perhaps vacuous main. Or, the main may already be in communication with other boilers. In that case the water in the vertical length is probably cool. When the boiler valve is opened, if the boiler is at a lower pressure than the main, the water will be pushed back into the newly opened boiler and all may be well. But it may be that the valve is one which cannot be opened, but must open itself when the pressure below it becomes sufficient to do so. Then when steam does pass the valve it probably lifts the water in the vertical pipe. That portion of the water next the valve is probably hotter than that higher up the pipe. Lifting proceeds without disturbance for a time. Then the column of water breaks; the water mixes with the steam; there is a sudden condensation below the water and a vacuum is formed. The water is driven violently downward by the pressure above and probably the valve is broken. If the valve is opened suddenly and widely this may not occur, for steam flows from the boiler below as quickly as from the main beyond. But if the valve is just "cracked," then a vacuum can form and trouble will ensue.

Kinetic Theory of Gases

As regards the flow of steam and its velocity under such conditions, the kinetic theory of gases is a help in understanding what may happen. According to the kinetic theory, a gas consists of an infinity of particles endowed by heat with the power to move continuously at a high velocity of several hundred feet per second. Obeying the law of motion, they move in straight lines but are in a continuous state of collision with each other and the bounding surfaces around them. But, if by condensation, all particles along a certain section are made suddenly to disappear, a vacuum has been formed and this is simply an absence of live particles. The particles beyond the limit of destruction, having now no opposition, rush into the empty space and drive forward any interfering plugs of water and so produce a destructive water hammer. If the run is long enough, such a plug of water will get up a velocity equal to that of the steam, or several hundred feet per second. No plug of inert water can travel faster than the molecular velocity of the steam, for the steam has a velocity limited by its own physical conditions. But this velocity is so great that immense pressures are created by projected plugs of water.

Where valves are thus set at the base of vertical lengths of pipe, it is proper to apply a drain pipe and a trap. And these drain pipes may fail; they may be frozen, or choked by scale or dirt, and then comes trouble. Clearly, then, the proper thing to do is to arrange everything so as to obviate all drains but the one near the engine, and this should be of fair diameter unlikely to choke. The boiler layout will then properly be such as to give a flow for water toward the engine, while keeping the boiler stop valve near the boiler crown, but not too near. If the steam main is near the stop valve there will be lack of elasticity and good practice will place the valve a couple of feet or so above the boiler crown, with a few feet of pipe connection to the main. Further elasticity may be given by an expansion bend, or U, in the course of this branch, the bend standing out horizontally from the branch pipe and not vertically. Water is then free to flow with no obstruction.

Nowadays, when it is becoming more and more the practice to pay attention to smoke or its nonformation, the raising of the boiler so high above the firing floor is not regarded so seriously. I am speaking now of water-tube boilers which cannot be worked without smoke, unless provided with room sufficient to build in arches and return arches between the tubes and grate. But, placed thus high, the steam-gage and the water-gage glasses are out of proper observation from the floor, and a satisfactory design should insure a platform at a suitable hight from which to attend to such matters.

With shell boilers of small diameter, this question of an upper platform will not arise. The question of water in the steam pipe only exists in the absence of superheat. But even with superheated steam the layout should be as though the steam might be wet.

Few engines will not bear a temperature of 500 degrees Fahrenheit. Even if they will not bear this they will bear some superheat. And if there is any superheat at all at the engine, then it is known that the steam is dry. A calorimeter to test steam returns probably does record with approximate accuracy the wetness or the dryness coefficient of the steam that passes through it; but the man is still missing who can devise an instrument or a method by which the calorimeter can be supplied with steam of the same degree of wetness as that which is traveling along the steam main. All calorimeter tests are therefore a delusion, unless they are made on superheated steam, and with superheated steam a thermometer tells all that a calorimeter can tell and tells it with more accuracy. All boiler tests which are calculated out on the basis of a steam production so many per cent. wet, as shown by calorimeter tests, are worthless.

The only method of determining the amount of water carried out of a boiler is to catch it as it drains out of the separator when the boiler is being worked at test rates and pressure, and the steam is getting away at such a rate that the pressure is kept steady. Probably a calorimeter test would show the quality of this steam if taken just past the separator, and the wetness thus found would then be increased by whatever the separator drains showed to be separated from each thousand pounds of steam. The result would not be accurate, but it would be far more approximately true than an ordinary calorimeter test, especially of the kind encountered not long ago, where the testing engineer removed the calorimeter from the branch provided for it and put it in a better place because at the first place the steam came so very wet.

But how did he know that this wet steam was not what the boiler was giving all along? Probably it was so, for the test results were beautifully abnormal and good.

An Example of Lubrication Economy

In a plant equipped with a well-designed oiling system an expert was able to effect a reduction of 42 per cent. in the annual lubrication cost. This was accomplished by doing away with small wastes and a careful selection of the proper oils to meet the various requirements.

With a plant in which the engines, shafting, etc., are well equipped with oiling systems, and where those in charge operate with an economical use of oil, it would seem that the lubrication account must be down to the point of lowest practicable cost. Yet the engineering chemist is able to step in and reduce the lubrication cost by 35 to 50 per cent., without any sacrifice of efficiency.

Two things are probably true in the majority of plants whose owners have attempted to enforce practicable economy: first, real economy in use does not make up for lack of economy in buying lubricants; second, the lubrication expert, because of his more varied experience, can usually discover some substantial economy in usage that has not occurred to the engineers and superintendents, whose intention to economize has not compensated for lack of the technical knowledge that enables the expert to economize.

A striking example of gain in lubrication economy, due to expert technical knowledge of the subject is provided in a report by Arthur D. Little, chemist and engineer, of Boston. This case is the more striking because the plant was very well equipped, and was well managed up to the limit of the technical knowledge of those in charge. During the year preceding this investigation and report, the plant had used 18,500 gallons of various oils, ranging in price from 14 to 49 cents per gallon; and 4675 pounds of grease and gear coating, at prices ranging from 5 to 9½ cents a pound; the total lubrication cost being $4425.

An indefinite amount of waste was due to the absence of a properly equipped central storage place for the oil. Barrels of oil were distributed in different places, and a number of these barrels were found to be leaking. The first remedial step was the construction of an oil house containing metal tanks provided with key faucets, into which the oil could be emptied as soon as received. Provision was made for gaging the barrels as received, and measuring the deficiency of oil in each barrel. This often leads to a substantial saving, but it is rare to find a plant in which it is done. The next step was the introduction of a better system of recording the amounts distributed and the place of use. Previously, upon requisition, oil had been issued by the barrel, but there had been no system of records from which could be made a quick computation of consumption and cost. The new system made it possible to hold each engineer and fireman responsible for his own lubricating costs, and this, of course, tended toward economy. The four large engines in the plant lost about 150 gallons of engine oil per month. The report laid the greater part of this loss to wiping up engine frames and floors with waste, which was then thrown away or burned. To check this loss an oil- and waste-saving machine was recommended, which makes waste fit for repeated use and saves a large proportion of the oil contained in the waste, often as much as 90 per cent.

Economy in purchasing lubricants is reached by following two rules: First, it should be determined just what oils and greases are needed, and the exact chemical and physical properties of each. Second, these necessary qualities should be expressed in specifications, and all lubricants purchased by these specifications, on annual contracts awarded after competitive bidding.

Analysis of the oils and greases used at this plant showed that all were of good quality. Their chemical and physical properties were the same as those which some other plants had succeeded in getting, by buying on specification, at a much lower cost. A cylinder oil, equal in every way to that for which 48 cents had been paid, was obtainable for 25 cents, or even less. The machine oil, costing 33 cents a gallon, and used on the engines, proved to be substantially the same in chemical and physical qualities as the 14½-cent shafting oil. As a substitute for these two oils the report gave specifications for an oil that was obtainable on annual contracts at about the price which had been paid for the shafting oil. It was shown that greases of the desired quality could be bought at from 3 to 5 cents a pound to displace those for which 9½ cents had been paid. Further economy was attained by substituting for the engine and machinery oil used on the slow-speed machinery, a good black oil at about 9 cents a gallon. Similarly, by using a more viscid gear coating during the summer, it was possible to make a reduction of 25 per cent. in the annual consumption.

By reducing the cost of the different oils and greases, by using lower grade oils where these grades were good enough, and by removing certain sources of actual waste, the experts were able to point out a reduction in the annual lubrication cost from $4425 to $2570, or a saving of nearly 42 per cent.

The most important lesson in the facts of this case is that maximum economy in both the use and the purchase of lubricants requires expert chemical knowledge of oils and greases plus a knowledge of the operation and requirements of machinery, and finally, a knowledge of how to buy the required lubricants at the lowest price. The practical value of the lubrication expert depends upon all these things, but in the last analysis his chemical knowledge is the one essential thing that the busy practical engineer or superintendent does not have, and has no time to acquire. Most operating men, having found an oil that serves them well, go on buying that oil because they have nothing but the brand name, or the reputation of the dealer, by which to identify it. When chemistry steps in, the plant ceases to pay for reputation, the lubrication outlay is cut down, and the best oil for the requirements is secured.

Calculating Heat Value of Coal

By Julian C. Smallwood

Calculation of the amount of air required to burn a pound of coal based on either proximate or ultimate analysis of the coal. How to determine the heat value of the coal; the loss of heat through various causes and the "heat balance."

In a previous article a method was given of making calculations from flue-gas analyses based upon the heat value of the carbon burned in the furnace. As stated there, a pound of pure carbon, burned completely to carbon dioxide, yields at the greatest, 14,650 B.t.u. Instead of all this heat being transferred to the water in the boiler, part of it escapes up the chimney as heat added to the flue gas, part of it is lost through incomplete combustion so that not all of the heat that can be is generated, and part of the carbon is not burned at all on account of slipping through the grate and into the ash pit. A list of the various directions in which the available heat of the coal is dispersed is as follows:

Useful heat added to the water, from 50 to 70 per cent. Radiation from boiler and setting, from 5 to 10 per cent. Ashes and unburned carbon in ashes, 5 to 10 per cent. Flue gases and unburned gas in flue, 20 to 40 per cent.

It is with the last item that the flue-gas analysis enables us to deal and, by our calculations, to reduce to the least possible amount. This loss may be further subdivided as follows:

Heat carried up the chimney by the flue gases analyzed and by the moisture in the coal. Loss from incomplete combustion of carbon monoxide and other gases.

The previous calculations have taken no account of the moisture in the flue gas or of combustible material other than carbon which may be in the coal. If something is known about the make-up of the fuel we may include these items in our considerations. Further, if we know something about the actual heat which one pound of the coal will deliver when completely burned (that is, its heat value), we can base our calculations upon this instead of the heat value of carbon. Finally, if we know a little more about the flue gas constituents we shall be able to determine, and therefore regulate, some material losses that may occur. With this in view the calculations to be presented in this article will include such as may be obtained from a usual gas analysis as before described in conjunction with a coal analysis and further calculations from a more complete gas analysis. The methods of obtaining the additional data will be briefly described.

To understand as clearly as possible the process of coal combustion we must first consider in a more detailed way the constituents of the coal. Bituminous, semi-bituminous, and anthracite coals may be distinguished by the amount of so called volatile matter which they contain. This is burnable material which may be driven from the coal by a low heat without causing it to ignite just as sap is driven from some woods in the form of smoke when they are heated to a point just below the burning point. The amount of volatile matter contained in the different kinds of coal when dry is roughly given in Table 1, which also shows the amount of carbon and ash left when this matter is driven off.

TABLE 1. CONSTITUENTS OF DRY COAL.

Per Cent. by Weight.

	Bituminous.	Semi-Bituminous.	Anthracite.
	From	From	From
Volatile matter....	20 to 35	10 to 20	3 to 10
Average........	30	17	8
Fixed carbon.....	60	74	85
Ash..............	10	9	7

As the volatile matter forms an important part of the heat value of the fuel (particularly of bituminous coals) we should be much interested in burning all of it.

A more complete analysis of coal shows it to consist of the elements shown in Table 2.

TABLE 2. ELEMENTS IN COAL.

Per Cent. by Weight.

Element.	Bituminous.	Semi-Bituminous.	Anthracite.
Carbon (C)...........	76	82	89
Hydrogen (H_2).......	5	4	2
Nitrogen (N).........	1.5	1	0.8
Oxygen (O_2).........	7.5	4	1.2
Ash.................	10	9	7

The volatile matter contained in coal is made up of carbon and hydrogen in some form and if it is completely burned the products of combustion are carbon plus oxygen or CO_2 and hydrogen plus oxygen or H_2O. For the last named product two atoms of hydrogen combine with one of oxygen and H_2O or water is formed.

We do not know just how these elements are held by the coal; whether the oxygen and hydrogen are there in the form of oxygen and hydrogen, or whether the hydrogen exists as a chemical compound with part of the carbon. What we do know is that when using bituminous coal, a large amount of hydrogen which, when *freed* from the coal is combined with carbon, is driven off and if this hydrocarbon, as it is called, is not burned under the boiler a considerable loss of heat results. It is therefore an important matter to learn whether there is any hydrogen in our flue gases, particularly when using soft coals.

In Table 1 is given an illustration of the results obtained from what is called the proximate analysis of coal. This can be made quite readily by one not experienced in chemical measurements. As a rule, the determination of moisture is made as well, for all coals contain a certain amount of water, though they may appear to be dry. It is aside from the purpose of this article to tell in detail how coal analyses are made, but the following brief outline of the proximate analysis may be of help. The coal is first sampled carefully by "quartering," a method of getting a small representative sample from a large amount, and then ground, sifted and weighed. The sample is then heated in a platinum pan at a low heat to drive off the moisture. The difference between the original weight and the weight after this heating gives the amount of moisture. The sample is then again heated at a somewhat higher temperature until the hydrocarbons are all driven off, and their amount is determined by the difference in weight. Next, the sample is heated to the burning point and the carbon consumed. The fixed carbon is thus ascertained; the residue is ash, which contains shale, dirt, etc.

In Table 2 are given results of what is called an ultimate analysis. Such an analysis is quite impracticable for any but an experienced chemist to make successfully. It is useful, however, occasionally to have such an analysis made of samples of the coal with a view to checking closely the boiler's performance. The results of such a test generally do not include moisture, the sample being completely dried before testing.

To determine the heat value of the fuel one of two methods may be used. A special test may be made of a sample or the result may be calculated from the proximate or ultimate analyses. The former is the more accurate way and is not difficult, but the apparatus is expensive. The measurement is made by actually burning a small sample of the coal in an air-tight chamber immersed in water. The weight of the water and its rise in temperature due to the combustion being known, the heat generated may be calculated. There are various types of these heat-measuring devices (known as coal calorimeters) on the market and every steam-power plant of any size should have one, if for no other reason than to ascertain definitely the value received for the money put into coal.

To calculate the heat value of a fuel from its chemical analysis, various formulas have been proposed. These are based partly on the fact that the heat values of the elements composing the coal are known. For example, the heat value of one pound of hydrogen is 62,100 B.t.u. and if we had a coal containing three-quarters carbon and one-quarter hydrogen, we should expect the mixture to yield

$$14{,}650 \times \tfrac{3}{4} + 62{,}100 \times \tfrac{1}{4} = 26{,}500 \text{ B.t.u.}$$

and this would be the case if the carbon and hydrogen existed as such in the coal. This, however, is not so; they are combined as hydrocarbons and otherwise. Now when combustion takes place some heat must be given up to these gases to separate the carbon from the hydrogen so that they may recombine with oxygen. That is, when a chemical compound is broken up into its elements, heat must be added to it; when these original elements combine again with themselves or some other element heat is generated. So there is an interchange of heat in burning coal which makes its heat value dependent upon the previous combination of its elements. We therefore cannot calculate the heat value precisely but certain experimental formulas enable us to make very acceptable estimates of it. One* of them is:

Heat value equals 14,650 times the weight of carbon in one pound of fuel plus 62,100 times the weight of hydrogen in one pound of fuel minus 5400 times the sum of the weights of oxygen and nitrogen.

Taking as an example the data of the complete analysis given later in this article, there results

$$14{,}650 \times 0.837 + 62{,}100 \times 0.047 - 5400 \times 0.058 = 14{,}870 \text{ B.t.u.}$$

Another formula,† for use with the proximate analysis, is:

Heat value equals 146.5 times the per cent. of fixed carbon in one pound of fuel plus *K* times the per cent. of volatile matter in one pound of fuel.

The quantity *K* has different values depending upon the amount of volatile matter in the coal and should be selected from the following table:

Volatile Matter. Per Cent.	K.
2 to 15	234
15 to 30	180
30 to 35	171
35 to 40	162

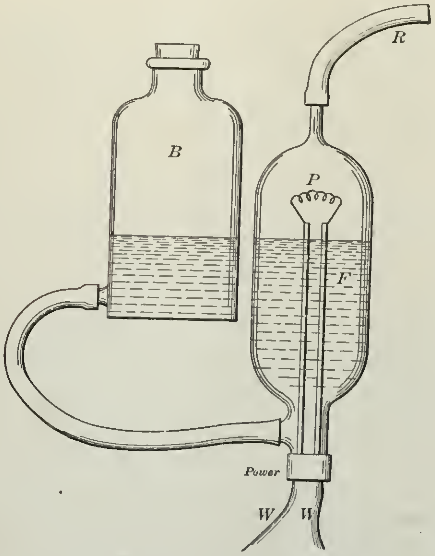
Taking for an example the data from the same sample as was considered in the calculation just made, the heat value is

$$146.5 \times 72 + 180 \times 22 = 14{,}500 \text{ B.t.u.}$$

To return to the flue gases, the moisture they contain can be calculated from the hydrogen and moisture in the coal. The moisture may also be measured directly by the use of special apparatus but this is hardly necessary. The unburned hydrogen in the flue gases may readily be ascertained with a small piece of apparatus to be used with the Orsat in a way which will be described. The only constituents we do not consider, then, are a gas formed by sulphur in the coal whose effect is negligible and, perhaps, some soot or solid carbon due to the too sudden cooling of the glowing particles against a comparatively cool part of the boiler shell.

Briefly, the method of determining the hydrogen is, first to convey the sample, to which some air has been added, to a flask in which all of the hydrocarbons present are completely burned. From the consequent reduction in volume the hydrogen can be calculated. In more detail the process is as follows:

The gas sample is first analyzed for CO_2, CO and O, in the usual manner with the Orsat apparatus. Part of the residue, consisting of nitrogen and hydrocarbons, is then forced out of the measuring tube into the air, 50 cubic centimeters being left. Then, 50 cubic centimeters of air is drawn into the measuring flask and mixed with the 50 cubic centimeters of the sample. The combustion flask *F*, shown in the figure herewith, is then connected by means of the rubber tube *R* to the entrance nozzle of the Orsat apparatus. The combustion flask and tubing have previously been filled with water by elevating the bottle *B*. Having made the connection, the cock between the Orsat apparatus and the combustion flask is opened and by manipulating the two water bottles the sample is slowly forced into the combustion flask. This contains a coil of platinum wire *P*, the ends *W* of which are connected to dry batteries or to a properly arranged lighting circuit so as to heat the coil to a bright red heat. As the gas enters the chamber it passes this hot coil and the hydrocarbons are burned to H_2O and CO_2. The residue is then returned to the measuring flask and the diminution in volume is noted, care being taken to allow time for the gases to cool.



Combustion Flask for Hydrogen Determination

Each atom of carbon present before combustion has been taken up by one molecule of oxygen to form CO_2 and there is no reduction in volume due to the burning of the carbon as one molecule of oxygen occupies the same volume as one of CO_2. The hydrogen, however, has formed water which condensed at once so that the total volume before combustion is diminished by the oxygen turned into water. In this combination there is a total of three parts of gas; two of hydrogen and one of oxygen. The hydrogen originally present is, therefore, two-thirds of the reduction in volume. To obtain the amount of hydrogen in the 50 cubic centimeters of the residue, the diminution of volume after treatment in the combustion flask is multiplied by two-thirds. We can then allow for the hydrogen contained in that part of the discarded residue as in the following example:

Fifty cubic centimeters of air added to 50 cubic centimeters of residue. Diminution noted, 0.9 cubic centimeter. Hydrogen in 50 cubic centimeters,

$$\tfrac{2}{3} \times 0.9 = 0.6$$

cubic centimeter. Hydrogen in 81.5 cubic centimeters,

$$\frac{81.5}{50} \times 0.6 = 1$$

cubic centimeter, or 1 per cent.

In testing for hydrogen in this way nothing is learned about the forms in which it appears as hydrocarbons. There are methods by which the free hydrogen and the hydrocarbons can be measured separately, but as the hydrogen alone gives a very good idea of losses due to unburned hydrocarbons, these other more complicated methods are not recommended.

In the process described if there is any ethylene gas (one of the hydrocarbons) present it will be absorbed in the cuprous chloride solution and appear as CO. On this account the determination may not be absolutely exact, but it is close enough for all practical purposes.

Coming now to the calculations to be obtained from the tests, let us assume the following results of coal analyses:

PROXIMATE ANALYSIS.

	Per Cent.
Moisture	1
Volatile matter	22
Fixed carbon	72
Ash	5

ULTIMATE ANALYSIS.

	Per Cent.
Carbon	83.7
Hydrogen	4.7
Nitrogen	1.6
Oxygen	4.2
Ash and sulphur	5.8

and of a flue-gas analysis,

CO_2 = 9 per cent., O_2 = 8 per cent., CO = 1.5 per cent., and N = 81.5 per cent. Heat value as determined by the calorimeter, 14,800 B.t.u.

Air Theoretically Required

Theoretically, only enough air is needed to burn all of the carbon to CO_2

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*Mahler's.

†Goutal's.

and all of the hydrogen to H_2O. In one molecule of H_2O there are two atoms of hydrogen weighing one atomic weight each and one atom of oxygen weighing sixteen. Each atom of hydrogen then combines with eight times its weight of oxygen; one pound of hydrogen would require eight pounds of oxygen for complete combustion and so on. To burn the 0.047 of a pound shown by our analysis we must furnish,

$$0.047 \times 8 = 0.376$$

pound of oxygen. To burn the 0.837 of a pound of carbon we need

$$0.837 \times 2\tfrac{2}{3} = 2.232$$

pounds of oxygen since carbon requires $2\frac{2}{3}$ times its weight of oxygen. This makes a total of 2.608 pounds. As the coal contains 0.042 pound of oxygen, only

$$2.608 - 0.042 = 2.566$$

pounds need to be furnished by the air supplied to the boiler. As air weighs four and one-third times the oxygen it contains, the air actually needed is,

$$2.566 \times 4\tfrac{1}{3} = 11.1$$

pounds of air per pound of coal. This calculation is useful to determine how much more air than necessary we are using.

Air Actually Supplied

Calculation to determine the weight of air actually supplied is made, as described in the second article of this series, by first finding the weight supplied per pound of carbon. From the flue-gas analysis, the total atomic weight of oxygen is

$$9 \times 32 + 8 \times 32 + 1.5 \times 16 = 568$$

and the total atomic weight of carbon is

$$9 \times 12 + 1.5 \times 12 = 126$$

and

$$\frac{568}{126} = 4.5$$

pounds of oxygen per pound of carbon, or

$$4\tfrac{1}{3} \times 4.5 = 19.4$$

pounds of air per pound of carbon.

Since there is only 0.837 pound of carbon in a pound of coal,

$$0.837 \times 19.4 = 16.2$$

pounds of air per pound of coal.

So far this calculation does not account for that part of the air used which has been turned into water nor does it account for that part of the carbon which may exist in the flue gases as hydrocarbons. As far as the former is concerned we may assume that all of the hydrogen is changed to water before reaching the stack. In the calculation for the necessary air we find that 0.376 pound of oxygen is needed for the hydrogen. The coal furnished 0.042 pound of this, leaving

$$0.376 - 0.042 = 0.334$$

pound to be furnished by the air. The total air is, therefore,

$$16.2 + 0.334 \times 4\tfrac{1}{3} = 17.7$$

pounds of air per pound of coal.

Loss from Excess Air

The theoretical amount of air necessary was found to be 11.1 pounds. The excess is

$$17.7 - 11.1 = 6.6$$

pounds. If the air is heated from 60 to 560 degrees, the heat added, since the specific heat of air is 0.238, equals

$$0.238 \times 500 \times 6.6 = 785 \text{ B.t.u.}$$

Dividing this by the heat value of the coal gives

$$\frac{785}{14{,}800} = 5.5$$

per cent. loss.

Total Heat up the Stack

The total weight of flue gas that comes from one pound of coal is equal to the sum of the weights of the air fed to the coal and that part of the coal which is burned. That is, the air and the combustible part of the coal which appear as flue gas must retain the weight they had before combining. The analysis of the coal shows it to contain 5 per cent. of ash. If all but the ash is burned, 0.95 pound of each pound of coal passes up the chimney. This added to the weight of air per pound of coal equals the total weight of flue gas to each pound, or

$$0.95 + 17.7 = 18.65$$

pounds of flue gas per pound of coal. This includes the moisture from chemical union. As 0.047 pound of hydrogen combines with 0.376 pound of oxygen, the moisture resulting is

$$0.047 + 0.376 = 0.42$$

pound. The weight of the flue gas without the moisture is, therefore,

$$18.65 - 0.42 = 18.23$$

pounds. Taking the specific heat of flue gas as 0.235, which is very closely accurate, and the same temperature range as before, the heat added to the gas is

$$0.235 \times 500 \times 18.23 = 2130 \text{ B.t.u.},$$

or

$$\frac{2130}{14{,}800} = 0.142 \text{ or } 14.2$$

per cent. loss.

This calculation is approximate. In order to make an exact determination of this loss it would be necessary to use the specific heat of each of the flue gases and multiply the weight of each constituent by the corresponding specific heat and the temperature range; the sum of these would be the total loss. This, however, is hardly worth while in view of the uncertainty of some of our quantities.

Loss Due to Moisture

The specific heat of moisture, or superheated steam as it becomes, is comparatively high, about 0.48, so that its influence is to increase the heat lost up the stack. Moisture in the flue gases comes from two sources, from the chemical union of the hydrogen in the coal with oxygen and from moisture in the coal which exists in the form of water, as shown by the proximate analysis. Considering the former we do not know just when the chemical combination takes place or the temperature range through which the resulting steam is heated, so it is impracticable to make an accurate calculation for this loss. An easy way of estimating it is to assume that this moisture has heat added to it while in the form of steam through the whole range of temperature. The heat added then is

$$0.48 \times 500 \times 0.42 = 101 \text{ B.t.u.}$$

Considering now the loss due to the moisture shown by the proximate analysis, the moisture is first heated as water from the temperature of the boiler room to the boiling point, then turned into steam, and finally superheated. The specific heat of water is one, and when it reaches the temperature at which it is about to turn into steam at atmospheric pressure, that is, 212 degrees, each pound of it requires 970 B.t.u. to change its molecules from the solid to the gaseous form. The heat added to the 0.01 pound of water in our pound of coal is, therefore, found as follows:

$$\begin{aligned} 1 \times (212 - 60) \times 0.01 &= 1.5 \text{ B.t.u.} \\ 970 \times 0.01 &= 9.7 \text{ B.t.u.} \\ 0.48 \times (500 - 212) \times 0.01 &= 1.4 \text{ B.t.u.} \\ \text{Total} &= 12.6 \text{ B.t.u.} \end{aligned}$$

This added to the 101 B.t.u. previously calculated gives a total of 113 B.t.u. and the loss is

$$\frac{113}{14{,}800} = 0.0076$$

or nearly 0.8 of 1 per cent.

Loss Due to Incomplete Combustion of Carbon

The loss due to the incomplete combustion of carbon is obtained as described in the article preceding this one, except that the heating value of the coal is used. The total atomic weight of the carbon in the CO shown by our analysis is

$$1.5 \times 12 = 18$$

and the heat lost by each pound of carbon is

$$\frac{18}{126} \times (14{,}650 - 4400) = 1460$$

B.t.u. Since there is only 0.837 pound of carbon in each pound of the fuel, the heat lost per pound of coal is

$$\frac{1{,}460}{14{,}800} \times 0.837 = 0.083 \text{ or } 8.3$$

per cent.

None of the foregoing calculations from the flue-gas analysis must be regarded as absolutely accurate because of the assumptions which it is necessary to make. Chief among these is that all of the hydrocarbons are burned to H_2O and CO_2.

We are basing our calculations upon the carbon appearing in the flue gas, but there may be some unaccounted for as hydrocarbons. There is, however, only a small part of the original pound of coal existing in this form, if any, as most of the volatile matter is actually burned completely. All of our calculations as described are sufficiently accurate even if we have the results of a test for hydrogen. If we have such results we may determine the additional loss, previously unnoted, due to incomplete combustion of the hydrocarbons.

Let us suppose that this more complete test has been made and that the following data have been obtained from the same sample as that which we have been considering:

CO_2, 9 per cent.; O_2, 8 per cent.; CO, 1.5 per cent.; N, 80.5 per cent.; H_2, 1 per cent., and additional CO_2, 0.5 per cent. found after combustion of the hydrocarbons.

We may, if we wish, alter our previous calculations for this increased weight of carbon found by the additional test. As it causes but little error to neglect it, however, this is not necessary.

Loss Due to Incomplete Combustion of Hydrogen

The method is similar to that used in the preceding calculation. The part by weight of the unburned hydrogen is 2/128 since the weight of a molecule of hydrogen is 2 atomic weights. The heat value of a pound of hydrogen is 62,100 B.t.u. The heat lost in our example is

$$\frac{2}{128} \times 62{,}100 = 970$$

B.t.u. per pound of carbon or,

$$\frac{970}{14{,}800} \times 0.837 = 0.055 \text{ or } 5.5$$

per cent. This result is approximate since the heat value of the hydrogen and its weight were used instead of those corresponding to the hydrocarbons. It is, however, closely approximate.

This shows that a small percentage of hydrogen in the flue gases causes a considerable loss of heat. To complete the combustion of this gas in the furnace it may be necessary to reduce the rate of combustion or to use a greater excess of air. If the latter is done a greater loss of heat due to this cause results. But in this as in all of the adjustments to regulate the losses a compromise is effected and conditions maintained such that all the losses sum up to the least possible.

It is useful, after making the calculations, to make a heat balance; that is, an accounting for all of the heat the coal must deliver.

Heat value equals, heat up the stack; plus CO losses; plus H_2 losses; plus heat lost by unburned carbon slipping through the grate; plus radiation from the boiler; plus heat added to the feed water.

Assuming 10 per cent. to be lost through radiation and the inefficiency of the grate, the results of our example in percentage when arranged in this manner are as follows:

One hundred per cent. equals, 15.2 per cent. (up stack); plus 8.3 per cent. (CO); plus 5.5 per cent. (H_2); plus 10 per cent.; plus 61 per cent. The last figure represents the heat utilized and the efficiency of the boiler.

Although these calculations have taken some space to explain, they will not be found difficult to perform if once understood. Enough has been presented to show their principles. It is only necessary to suggest, in closing, that when the operator thoroughly understands these principles, he need not be confined to the exact rules set down here. He may not have all of the data necessary to apply them and, if not, he can easily make estimates of what he lacks and alter the basis of his calculations so as to fit such data as he has.

Uncle Pegleg's Philosophy

The reaction jet problem is straightened out by Uncle Pegleg who clearly shows that it is "change of velocity which takes energy."

It was not until some days afterward that Uncle Pegleg came back at me about the reaction jet. You remember that he had explained how and why a turbine or reaction wheel like Fig. 1 had to run at one-half the speed of the jet to get the motion and energy all out of the water. He had also explained that a movable nozzle had to move as fast as the jet in order that the water might pay out of it like a rope and simply fall dead with no backward motion. The force with which the jet pushed on the nozzle tank in Fig. 2, as he explained it, was:

$$\text{force} = \frac{\text{wt. of water}}{\text{discharge, per sec.}} \times \frac{\text{vel. of jet}}{\text{in feet per sec.}} \div 32$$

$$F = \frac{wv}{g}$$

I had evidently set him off upon a two days' think by saying that if the car with the jet on it moved at one-half the velocity of the jet the *space* through which it would move in a second would be one-half the jet velocity or $\frac{v}{2}$ and that

$$\text{force} \times \text{space} = \text{energy}$$

$$\frac{wv}{g} \times \frac{v}{2} = \frac{wv^2}{2g}$$

which, as he had explained before,* was all the energy that there was in the jet. It seemed then to figure that you could get the energy all out by running the nozzle at half the jet speed, while it was clear enough that the energy was not all out for the jet was still squirting backward over the ground with half the velocity that it came out of the nozzle.

One of the fellows stopped on his way to the engine room where we were talking one morning to discuss the ball game. After he had gone Uncle Pegleg asked me, "Does it make any difference whether a pitcher faces east or west?"

"No. Why?"

"Well, the earth is some 24,000 miles in circumference and it makes a turn in 24 hours. How fast is it going per second?"

Knowing that there were 5280 feet in a mile, I figured it:

$$\frac{\overset{1000}{\cancel{24000}} \times \overset{88}{\cancel{5280}}}{\cancel{24} \times \cancel{60} \times \cancel{60}} = \frac{8800}{6} = 1467 \text{ ft. per sec.}$$

"Well, then down at the equator a spot on the earth is moving from west to east with a speed of nearly a third of a mile a second. At the poles—no motion. In between, more or less according to latitude. Don't a pitcher ever feel the difference whether he is pitching against or with this motion, or whether he is playing in Atlanta or Boston?"

"No—what are you—stringing me?"

"Well, why shouldn't he? You've been finding out that it takes a good deal of energy to get up a velocity like that, even in a base ball."

I thought for a few minutes and then said: "Why, I don't see that the pitcher has got anything to do with that. The ball in his hand is going with the earth. *It has got that velocity already.* All he has got to do is to put that ball across the plate as hot as he can."

"That's the idea," said Uncle Pegleg. "It is *change* of velocity that takes energy. The earth is going around the sun like blazes besides spinning on its axis, but it don't bother us because we've got the motion and there's no sudden change in it. You get onto the elevator sometime with that big wrench and get one of the boys to start you up quick, and notice how much heavier the wrench gets for an instant. It has to be gotten into motion at thc speed of the elevator and your arm has got to furnish the pull to get it to moving. That's the pull that comes on you in addition to the real weight. Then let the boy stop quickly, and see how light the wrench gets. It has got the momentum of the elevator's motion. It wants to keep right on going and tends to lift itself when the elevator stops. But, and this is the important thing, it is only when the *rate of motion is being changed* that you notice anything of this kind. When the elevator settles down to running steady everything on it partakes of its motion and is just the same in the elevator as it would be on a stationary floor. When you are in a railroad car you feel the part of the car you are sitting against pushing you forward when the train starts quickly, and you have to cling to it or push backward on something to bring yourself to rest with the train when it stops suddenly; but after the motion becomes uniform you are just the same with reference to the train as you are now with reference to this room, and the movement of the train with reference to the earth wouldn't bother you any more than does the movement of the earth with reference to Mars. Two boys could toss a ball and it would be no harder or no different for the boy who tossed it toward the engine than for the one who tossed it toward the rear. So that when you make a test of a nozzle, for instance, in the hold of a vessel or on a platform car, you make it just the same as you would if you were glued to the land and you don't care whether the ship is going to Europe or to China, or whether the platform car is standing still or going like the wind, so long as its motion is uniform."

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*See page 84, May 17 number of Power.

I saw then that he had got the reaction-nozzle thing straightened out.

"You take a certain amount of water," he went on, "and you put a certain amount of energy into it by making it into a jet going at a certain velocity. In order to get that energy all back you've got to bring the water back to the velocity that it had when you started. If it was at rest with reference to the earth to start with you have got to bring it back to rest with reference to the earth. If it was in a tank on a moving car you have got to bring it to rest with reference to the car. If you start with it on the moving car and put in enough energy to get it into motion with a certain velocity *with reference to the car* you don't have to take out enough energy to bring it to rest *with reference to the ground* to get your energy back, do you?"

I began to see what was coming.

"You can work from any base you want to," he said, "but you must use the same base for the finish as for the start. If you start with the water at rest with reference to the train you must bring it back to rest with reference to the train. If you start with the water at rest with reference to the ground you must bring it back to rest with reference to the ground. If you start with the water in motion with reference to your base you must subtract the energy due to its velocity from that of the jet velocity if the initial velocity were in the same direction or add it if it were in the opposite direction, for it is the *change* in velocity which requires energy.

"Now in this case (Fig. 1), you take water from the tank, stationary with reference to the ground, you put a certain amount of energy into it with the pump forming it into a jet. This energy will be $\frac{w\,v^2}{2\,g}$, as I explained to you the other day, where w is the weight discharged per second, v is the *change* in velocity and g is that gravity thing, about 32 feet per second. As the water starts from rest, v is the velocity of the jet. Suppose it is squirting toward the west with a velocity of a hundred feet a second and the bucket is running toward the west with a velocity of 50 feet per second. Then the jet will overtake the bucket with a velocity of 100 — 50 = 50 feet per second and the bucket will simply reverse it and throw it backward with the same speed. Well, if it is traveling west in the bucket at 50 feet a second and is thrown eastward with a speed of 50 feet a second, its speed east will just offset its speed west and it will drop dead with no motion relative to the tank where it came from.

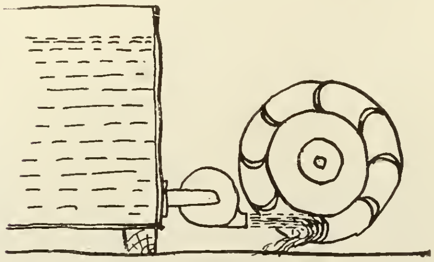


FIG. 1

"Now, let's take that tank on the car. Suppose the car to be running 50 feet per second eastward with reference to a fixed point on the *earth*, and the jet to be squirting westward with a velocity of 100 feet per second with reference to the *nozzle* or the *car*. If we are going to compare the energy put into the jet with the energy taken out of it we must refer the velocities from which those energies are figured to the same base. Eh? Well, let's refer them all to the earth.

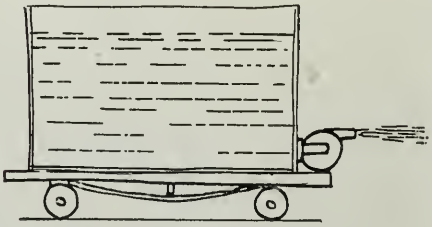


FIG. 2

"To start with, the water in the tank on the car is moving eastward with a velocity of 50 feet per second. The pump puts an amount of energy into it represented by $\frac{w\,v^2}{2\,g}$. If the car runs at one-half the velocity of the jet or through $\frac{v}{2}$ feet per second, the energy exerted upon the car will be as you figured it:

force × space = energy

$$\frac{wv}{g} \times \frac{v}{2} = \frac{wv^2}{2g}$$

and the car will have got back all the energy that the pump put in."

"But it hasn't brought the water to rest," I remonstrated. "The water is still in motion with regard to the earth."

"So it was to start with," returned the old man. "You found it in the tank going at the velocity of the train and you leave it going at the same velocity in the other direction. The amount of energy stored in it is the same in both cases. It is the energy that was stored in it when the train was started up overcoming its inertia like the elevator overcomes the inertia of the heavy wrench. It's just the same when we get through with it as when we took it, and has nothing to do with the pump energy or jet energy that we are considering."

"Then it isn't true that a reaction nozzle like that has to run at the full velocity of the jet to get the energy all out, is it?" I asked.

"Yes, it is, because if the thing is going to run for any time it has got to take new water aboard, and the force to get it into motion has got to be charged up against the reaction, so the net force and the net energy won't figure out as you figured them. I'll tell you about that some other time."

It has been stated by Mr. Parsons that steam turbines should not be applied to vessels of lower speed than 18 knots, and the experience of the Southern Pacific with the "Creole," running between New York and New Orleans, bears this out. Originally fitted with turbines designed to drive her at 16 knots, the vessel was found to be extravagant on fuel. She was recently taken to Cramps' yard, and fitted with reciprocating engines, which, with the original boiler plant, drove the ship at 16.55 knots from port to port, the boilers steaming easily. Much interest attaches to the 16-knot Government collier, which will use turbines, with the Melvin & MacAlpine reduction gear interposed between the turbines and the propeller. Some months ago this gear was illustrated in POWER.

Steam Turbine in Sawmill Plant

By Warren O. Rogers

> *For the general run of sawmill plant power is usually furnished by water or by a miscellaneous collection of steam machinery. The present plant is an exception, for it is equipped with a modern 500-kilowatt turbine generating unit.*

In the little lumber town of Ferguson, S. C., is located one of the largest sawmills in the South, that of the Santee Cypress Lumber Company. The steam plant consists of a cross-compound, 17 and 34 by 36-inch Buckeye steam engine, which drives the sawmill, using a steam pressure of 125 pounds per square inch. In the same engine room is a small Cameron & Barkley center-crank engine, belted to a Sprague direct-current generator of 50 kilowatts capacity. This unit is not in use at the present time, as the electric energy for lighting purposes is obtained from a turbine unit. There is also an 8x14x9x8-inch steam-driven Bury air compressor, and a three-ton Remington ammonia compressor driven by a vertical steam·engine.

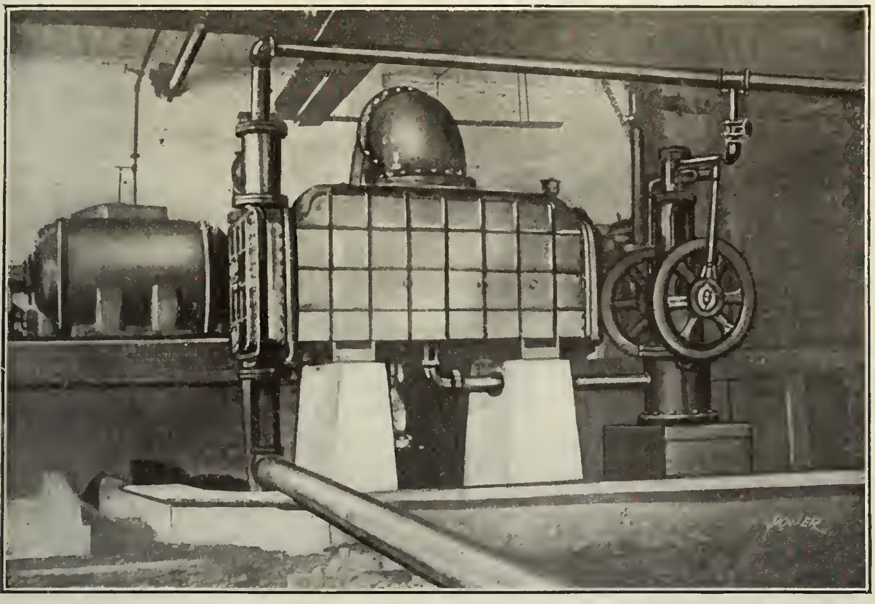


Fig. 1. Arrangement of Condensing Outfit

Just outside of the engine room, in a new addition to the plant, is located an Allis-Chalmers turbine to which is coupled a 500-kilowatt, three-phase, 60-cycle generator.

The exciter for this unit consists of a 6x6-inch American blower engine, which is direct connected to an Allis-Chalmers direct-current 120-volt generator. There is also a motor-driven exciter set which consists of a 15½-horsepower, three-phase 60-cycle motor, direct connected to a direct-current generator designed for 120 volts.

The steam turbine is connected to a Wheeler condenser mounted on concrete piers, as shown, so that the top of the condenser rises to a hight considerably above the top of the turbine. The steam connection between the bottom of the turbine and the top of the condenser resembles in form the letter S. An Edwards steam-driven air pump is connected to the condenser, as shown in Fig. 1. The condensing water is obtained from a pump pit located at the farther end of the building. In this pit are placed a steam- and a motor-driven centrifugal pump, the steam-driven unit being held in reserve.

Fig. 2 shows a view of the top deck of the boiler room, the furnaces being of the Dutch-oven type. Two methods of supplying the boilers with fuel are employed. The bottom firing floor is arranged as in the ordinary wood-working steam plant, regulation furnace doors being supplied for the purpose of firing the larger pieces of waste, bark, etc., by hand. The upper deck receives the sawdust and finer waste material from the mill by means of the conveyer system shown in front of the boilers, and from which a spout drops down over an opening in the top of the Dutch-oven furnaces. By this method the fine wastes are handled automatically, the only firing labor in the boiler room being that of pushing the accumulated material through the firing hole into the furnace. There are eight 125-horsepower and two 250-horsepower Casey-Hedges return-tubular boilers. These boilers are equipped with metal Dutch-oven furnaces.

At present the steam-turbine unit is not loaded to its full capacity, so that it is not running at its greatest efficiency. The installation of such a prime mover, however, shows that even at a sawmill plant economy is the watchword, and that the latest development in power-plant machinery is demanded.

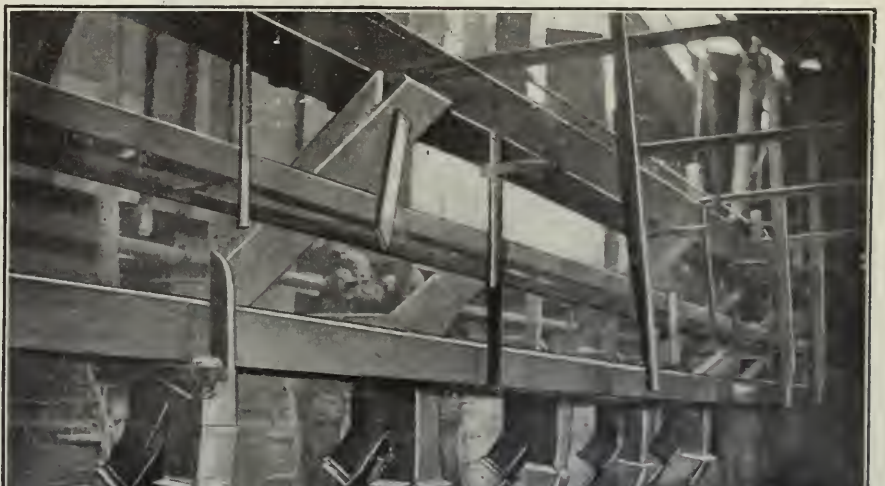


Fig. 2. Method of Feeding Sawdust to the Boiler

Automatic Damper on Air Flue

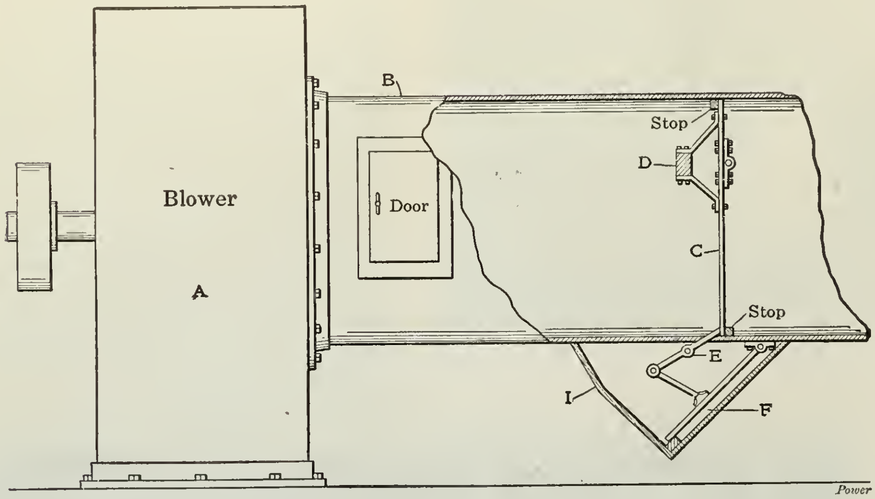
By J. G. Ould

Some time ago the writer was called upon to solve a problem concerning the ventilating apparatus in a building. The method of doing the work is set forth in the hope that it may be of help to someone in a similar predicament.

The building is equipped with an indirect-heating system in which fresh air is passed over steam coils and then forced into the various rooms. There is also a suction system of ventilation. The following remarks have to do with two of the suction and one of the heating fans, all of the Sturtevant type:

The two suction fans are situated in pent houses on the roof, and are placed over large air shafts into which suction ducts are led. One fan is 60 inches in diameter, and is belted to a 20-horsepower motor; the other is 84 inches in diameter and is belted to a 25-horsepower motor. The principal work of the smaller is to exhaust the foul air from two amphitheaters used as lecture halls, while the larger draws air from various sources.

The building is of fireproofed steel construction, and has heavy concrete and tile floors, which no doubt absorb considerable heat, thereby acting as reservoirs.



Automatic Damper on Heating Fan

The building is used in the daytime only, and consequently steam is shut off at night. This is true of the large suction fan, except that it is also shut down between noon and 1 p.m. The other suction blower is run only when the lecture halls are in use, which averages from five to seven hours per day.

When the suction fans were first put in, there were no dampers in the air shafts; so they were open to the outside air at all times. This naturally took a large amount of heat out of the building, not only in the day time, but at night as well, when the building was not in use and the fans were shut down.

The problem, then, was to install dampers in the ducts which would automatically open and shut when required; so that when the attendant started his motor, the damper would open, and when the motor was shut down, the damper would close of its own accord.

This proposition was under consideration for several days, the question being "how might it best be accomplished?" We had various sources of power at command, but finally hit upon the simple plan of using gravity in conjunction with a damper hung out of center, which proved to work very well without any attention on the part of the attendant.

The damper for one exhaust fan was made of a sheet of galvanized iron 41x42 inches and hung with its shaft 6 inches off center. On the end of the shaft is a lever, which allows the damper being kept open in summer time when we wish the heat to escape. The damper is placed on the outlet side of the fan and is just heavy enough to stay shut against the natural draft of the flue, but as soon as the fan comes up to speed the damper is held open by the upward rush of air. It gradually settles back on the stops, as the fan slows down.

The other exhaust duct needed a somewhat different treatment on account of there being a large beam through the middle of the duct, which prevented the use of a large swinging damper as in the first case. This damper was made somewhat after the plan of a window shutter, except that the hinges are at the bottom of the slats instead of in the center as in the shutter. The slats are of galvanized sheet iron, with edges turned and wired, and are supported on a steel rod which acts as a shaft. These slats are connected to a rod which in turn is connected to a small lever running through the duct.

The damper in the heating duct required still different treatment. In this case the fan draws its air supply through a horizontal flue, the inlet to which is exposed to the prevailing winter winds, and, unlike the foregoing cases, the damper is on the intake side of the fan.

The style of damper used on the exhaust flues could easily be opened by a strong wind blowing into the duct, if installed on this side of the fan, and if the temperature of the incoming air happens to be below freezing, supposing the fan to be shut down and the steam cut off the heating coils, any water remaining in the coils would freeze and burst the heaters. Even if the coils did not freeze, a zero temperature brought in through a 54-inch hole would soon lower the temperature of the room.

The type of damper finally adopted to meet these conditions is shown in the illustration; *A* is the fan housing, *B* the air inlet, *C* the overhung damper, *D* the weight, *E* the latch of the lock, and *F* the piston which actuates the latch to unlock the damper. The damper is shown closed and locked.

Its operation is as follows: As soon as the fan speeds up, it exhausts the air in the duct between the damper and the rotor and the outside air pressure forces up the piston *F*. This unlocks the damper, which with the suction on one side and the current of air on the other, begins to swing about its shaft. The overhung weight *D* then comes into play and with the aid of the air current keeps the damper steady and horizontal.

When the fan stops, the damper slowly returns to its original position, the weight *D* now helping to close it and force it over the latch, which as soon as the damper passes, springs up into place and locks.

This scheme has worked automatically for about four years without giving trouble of any kind.

Two reinforced concrete tanks have been built as part of a concrete-block chimney in France. The chimney is 180 feet high, the diameter at the summit 7 feet 9 inches and the tank capacity 33,000 gallons.

Mechanical Refrigeration

By F. E. Matthews

An elementary treatment of commercial systems with simple illustrations to show the principles of operation.

Practical mechanical refrigeration may be said to date back to 1855, in which year the ammonia-absorption machine was invented by Carré, and the Perkins ether compression machine, patented in 1834, was simultaneously commercialized by two different inventors in two different countries—Professor Twining in America and Mr. Harrison in Australia. It is a rather strange coincidence that the development of the two systems so commonly employed today, i.e., the absorption and the compression systems, should have been commenced the same year. Generally speaking, while both the absorption and the compression systems have been employed in connection with both the direct expansion and the brine systems for the production of both medium and extremely low temperatures, brine has been used more commonly in connection with the absorption machine and direct expansion in connection with the compression machine. The former has been employed more frequently for extremely low and the latter for comparatively low temperatures. The absorption machine is generally conceded to be more economical than the compression machine when producing extremely low temperatures, under which condition, because of the lightness of the gas under the correspondingly low pressures, the compression machine loses greatly in both efficiency and capacity. At the present time, however, it is problematical whether the substitution of the higher-efficiency gas engine as a prime mover for the inherently low-efficiency steam engine will not put the compression system in position to produce as economical results as the absorption system, even under the disadvantage of low temperatures. Since the compression system is in far more common use today, and since a large number of the parts of the two systems are common, the details of construction of the compression machine will first be considered.

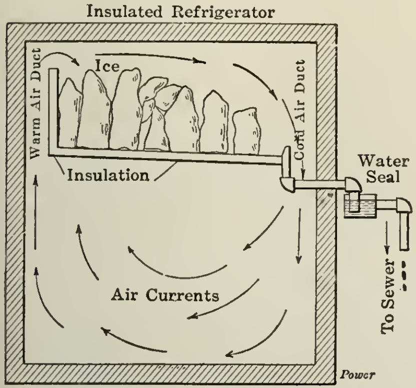


FIG. 1. REFRIGERATOR COOLED BY ICE BUNKER

The function of a refrigerating apparatus, whether it be an absorption or a compression refrigerating machine, or simply a bunker full of ice, is to provide a heat-absorbing medium which, after it has absorbed its fill of heat from the products to be cooled in the cold-storage rooms, may be removed from the coolers so that the heat absorbed may also be removed. After its removal from the coolers, this medium may be divested of its heat, after which it may be allowed to return to the coolers to absorb more heat, as in the case of ammonia or brine circulated through coolers; or it may be thrown away and a new supply introduced, as in the case of cooling by the use of ice.

A diagram of the simplest arrangement of an ice bunker for cooling a cold-storage compartment is shown in Fig. 1. The ice in this case is stored on a water-tight floor over the compartment to be cooled, the cooling being effected by a natural circulation of air up over the ice and down through the cold-storage compartment. Movement of air without the use of a fan driven by an outside source of power is comparatively slow, owing to the limited power available in the difference in weight of the ascending column of warm and the descending column of cold air. Since both the bunker and the room below are completely filled with air, the sinking of air in one place displaces air in another place. The warmer air is forced up to the bunker where it in turn becomes gradually cooled, finally flowing down by gravity into the room below.

Bunkers for ice, or other cooling means, are usually constructed in the form of a tank with one side removed, as shown in Fig. 1, ample spaces for ducts being allowed between the sides of the tank and the sides of the room to permit the air to circulate freely. Were the left side of the tank also removed, it is obvious that the tendency would be for the cold air to flow off of the ice in both directions. This would give rise to conflicting currents of air which would check the circulation, and for the same rate of air circulation would require a greater difference in temperature between the air in the bunker and that of the cooler below.

To prevent the cooling of the rising column of air in the warm air ducts as well as that lying next to the floor in the bunker, which not only retards circulation by reducing the difference in temperature, but may also precipitate moisture when the warm air approaches saturation, the air-duct walls and bunker floors should be insulated. When operating with properly insulated bunkers and liberal ducts, the circulation should be sufficiently rapid to carry any moisture-saturated air that may enter the cooler from the outside or that may have become saturated through contact with stored products to the ice chamber before its moisture can be precipitated by contact with the cold surfaces of the cooler. When cooled in this way the excess moisture is precipitated in the ice bunker and flows away with the melted ice. Where the warm air ducts or the bunker floors are not insulated, excess dampness can be prevented only by the use of some such deliquescent salt as calcium chloride. This is the case when the air of the cooler is not allowed to come in contact with the melting ice but is chilled by contact with the metallic floors of overhead bunkers or the walls of tank bunkers.

Where salt is used so that the resulting temperatures are below freezing, the precipitated moisture is frozen on the heat-absorbing surfaces and this difficulty is avoided, but in time this accumulation has to be removed, thus giving rise to other difficulties.

Exceptionally well insulated coolers have been constructed in which it is possible to maintain a temperature as low as from 38 to 40 degrees Fahrenheit in compartments cooled by natural circulation and ice without the use of salt. The average cooler, however, is not capable of producing such favorable temperatures, and the present-day demand for lower temperatures makes it neces-

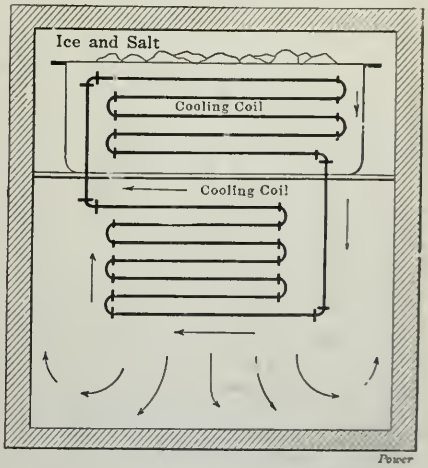


FIG. 2. GRAVITY BRINE SYSTEM

sary to resort to other means. The addition of salt allows the natural circulation and ice system to satisfy a few such requirements, and the further addition of a fan to force the circulation may allow it to include a few more. For widening the application of ice cooling, the system illustrated in Fig. 2 has been devised. This system consists essentially of a tank for holding the ice and salt in the proportion required to produce the desired temperatures, and a continuous pipe circuit, a part of which is located in the cold-storage compartment where it absorbs heat and the other part in the ice tank where it gives up the heat to the ice or freezing mixture of ice and salt, as the case may be. The pipe is filled with brine of just sufficient density to insure against freezing. The brine in the coil of pipe located in the tank, becoming heavier, flows down into the coil located in the cold-storage compartment, causing the warmer, lighter brine to pass upward and take its place in the ice tank. Heat is conveyed from the cold-storage compartment to the ice chamber by the natural circulation of the conveying medium as in the preceding case, but in this case the medium is a liquid while in the preceding one it was a gas. An abnormal rise in temperature of the cooler increases the velocity of the brine circulation and consequently increases the refrigerating capacity of the system. By the use of this system it is possible to prevent the air of the cold-storage compartment from becoming contaminated by contact with melting ice and unsanitary ice bunkers.

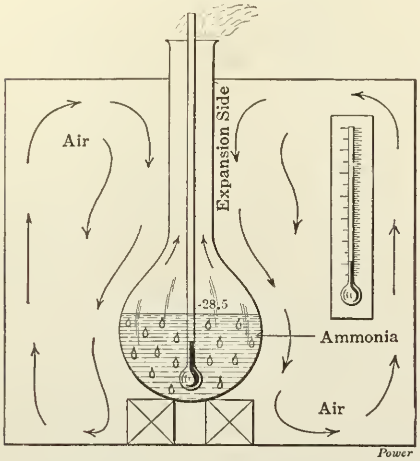


FIG. 3. ILLUSTRATING DIRECT-EXPANSION SYSTEM

The process of mechanical refrigeration when the refrigerating medium is a condensable gas, or more accurately speaking, a liquid having a sufficiently low boiling point under atmospheric or other conveniently produced pressures, to produce the desired temperatures, may be explained as follows:

Fig. 3 represents a flask partly filled with a refrigerant such as anhydrous ammonia. Since the pressure on the refrigerant in the open flask is only that of the atmosphere, it will boil at —28.5 degrees Fahrenheit. As this temperature is far below that of the surrounding air under usual conditions, heat will pass from the air into the refrigerant; the former would be refrigerated and the latter heated up to the boiling point if it were not already boiling. At the boiling point the absorption of a definite amount of heat from the air effects the evaporation of a definite amount of the liquid; the anhydrous ammonia, absorbing heat directly through the walls of the containing vessel from the surrounding air, operates just as it and similar refrigerating media do in the direct-expansion refrigerating system. This cooling effect is hastened by a marked circulation of air around the flask, occasioned by the greater specific gravity or weight of the cooled film of air lying next to the flask, which flows down and away at the bottom of the flask, allowing the warmer air to rise and take its place at the top.

The brine system has its elementary counterpart in such an arrangement as that illustrated in Fig. 4, in which the flask containing the boiling anhydrous ammonia is immersed in a second containing vessel, such as a large beaker, filled with brine, a solution of sodium chloride (NaCl), calcium chloride (CaCl), or other salt in water. The low first cost, together with certain physical and chemical characteristics, has practically limited the commercial brine system to the use of either sodium or calcium chloride solutions. As in the preceding case, the boiling refrigerating medium absorbs heat from the surrounding medium, in this case brine which we will assume is of such a nature as not to freeze at —28.5 Fahrenheit. Since this solution does not freeze at the temperature of the evaporating refrigerant, and the latent heat of fusion of the liquid is not extracted, an evaporation of only a comparatively small amount of the refrigerant suffices to cool the surrounding solution almost down to —28.5 Fahrenheit. This cooling effect is hastened by a marked circulation within the solution itself, set up by the difference in specific gravity of the colder part of the liquid lying next to the flask, which sinks to the bottom of the beaker, allowing the warmer part to flow in and take its place.

The brine in the beaker having become colder than the surrounding air, heat flows from the air to the brine, just as in the preceding case it flowed from the air to the ammonia. It will be noted, however, that difference in temperature, and accordingly the flow of heat, between the air and the brine can never be so rapid as that between the air and the ammonia, except in the limiting case in which the brine and the ammonia are of the same temperature, when unfortunately the ammonia would have no cooling effect on the brine. If the same amount of the same refrigerant be placed in flasks of the same shape and size in both the foregoing experiments, it will be found that the brine employed in the second case will assume a temperature intermediate between that of the boiling ammonia and that of the surrounding air, and were there any way to measure the cooling effect produced on the air, it would probably be found that less heat would be absorbed in the latter than in the former case, notwithstanding the fact that the heat-absorbing surface of the beaker is greater than that of the flask. A very crude idea of this difference might be gained by surrounding the vessels used in each experiment by boxes made of the same size and material and noting the temperature of the air within.

The flasks containing the ammonia employed in the above experiments correspond to the expansion pipes immersed in and employed for the purpose of cooling the brine in the brine system described later, and the beaker containing the brine to the brine pipes immersed in and employed for cooling the air in the cold-storage compartments. If the flask of ammonia employed in the second experiment had been immersed in water, as shown in Fig. 5, instead of brine, as shown in Fig. 4, the water which freezes under atmospheric pressure at 32 degrees Fahrenheit, will give up first its specific then its latent heat to the anhydrous ammonia boiling under atmospheric pressure at —28.5 Fahrenheit, with the result that the water will first be cooled from its initial temperature to 32 degrees Fahrenheit, then be frozen at 32 degrees. If sufficient ammonia still remains and heat is not absorbed too rapidly from the surrounding air, the ice will be cooled several degrees lower than 32 degrees Fahrenheit; in fact, that part lying nearest the flask may be chilled to the temperature of the evaporating liquid, —28.5 degrees Fahrenheit. If a pound of anhy-

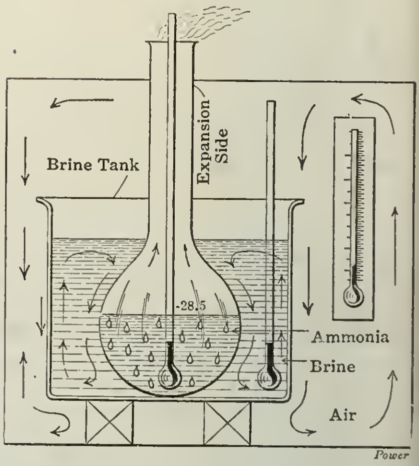


FIG. 4. BRINE CIRCULATING SYSTEM

drous ammonia be evaporated from and at —28.5 degrees Fahrenheit in the flask, the amount of heat which it will absorb will be 555 B.t.u., which is its latent heat of vaporization. If now it be assumed that this heat is all drawn from three pounds of water at an initial temperature of 73 degrees Fahrenheit, it will just suffice, assuming no losses, to cool it down to the freezing point and then to freeze it at 32 degrees Fahrenheit, 42 B.t.u. for each of the three pounds being required to overcome the specific heat of the water in cooling it through 41 degrees, and 144 B.t.u. for each pound of water at 32 degrees Fahrenheit being required to overcome the latent heat of fusion.

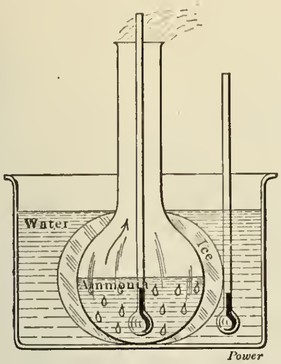


Fig. 5. Elementary Plate-ice-making System

Three pounds water × drop in temperature from 73 to 32 degrees × specific heat of water (unity) + three pounds water × latent heat of fusion of water (144 B.t.u.) = latent heat of vaporization of the ammonia (555 B.t.u.) required to cool and freeze the three pounds of water.

$$3 \times 1(73 - 32) + 3 \times 144 = 555.$$

If carried out in an ice plant instead of a laboratory, the apparatus requisite for the above experiments would probably appear more nearly as in Fig. 6. In this illustration the glass flask is replaced by an iron flask, or drum, such as is commonly used for shipping ammonia, carbon dioxide and other liquefied gases. The liquid from this flask is allowed to "expand" or escape in a small stream through a valve *V* and a pipe leading to an expansion coil. After traversing this coil, the vaporized ammonia escapes to the atmosphere. This outlet, however, is provided with a valve V_1, which, when closed, diverts the ammonia through a second pipe connecting through two outlets, each provided with a valve, the one leading to a tank of water and the other to an ammonia compressor. If diverted into the water, the ammonia vapor will be absorbed and be recoverable in the form of aqua ammonia, a principle which is employed in the absorption type of refrigerating machine. The ammonia vapor may be conducted to an ammonia compressor and suitable condenser, where it will be condensed and recovered in the form of anhydrous ammonia, a principle which is employed in the compression type of refrigerating system. The several parts used in commercial refrigerating systems to take the place of those illustrated in Figs. 3, 4 and 5 are represented diagrammatically in the figures that are to follow. Fig. 7 is such a conventional representation of an ammonia flask or receiver, a controlling or expansion valve *V* and a container of the boiling liquid ammonia, or expansion coil.

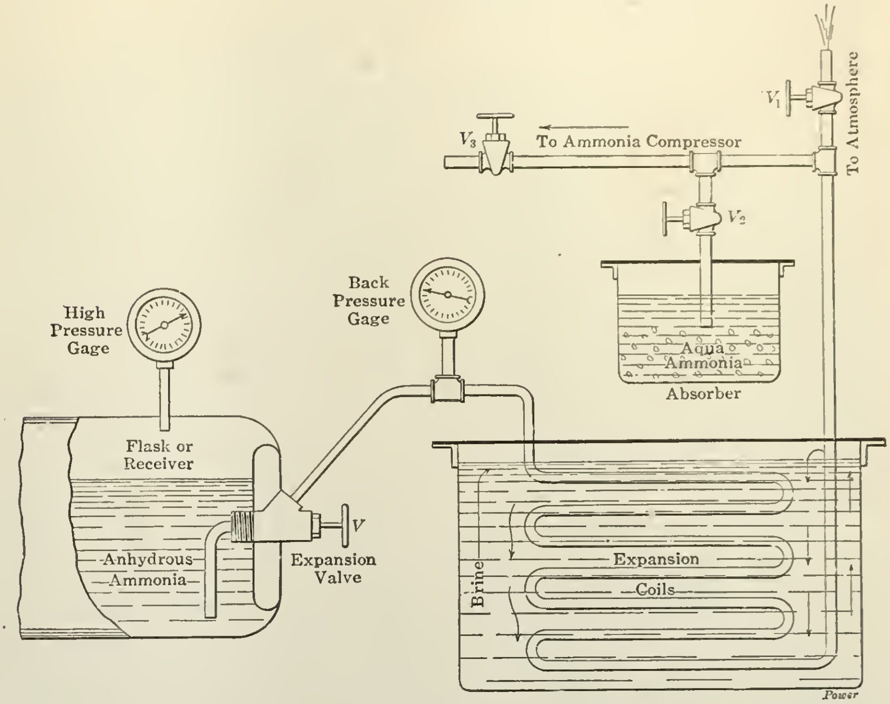


Fig. 6. Elementary Brine System

Were it not for the initial cost of the refrigerating medium, this elementary refrigerating system in which the ammonia is allowed to escape to the atmosphere after evaporation might find commercial application, but since anhydrous ammonia commands a practically fixed market price of $0.25 and upward per pound, according to the distance from point of production, such a system would be eminently impracticable.

A commercial compression refrigerating system consists of a set of pipes, or other containing vessels, in the cooler, in which the refrigerating medium absorbs heat at a low temperature from the products to be refrigerated; a second set of pipes, or other containing vessel, outside of the cooler in which the refrigerating medium gives up its heat to a secondary cooling medium, such as water or air, at a comparatively high temperature, and a compressor.

The pipes, located in the cooler, through the walls of which the ammonia absorbs heat from the objects to be refrigerated, are erroneously called expansion coils, and those through the walls of which the ammonia gives up its heat to the natural cooling medium, water, condenser coils.

The compressor is a mechanical means of circulating a heat-absorbing medium through the cooler, and of removing from this medium the heat absorbed *en route*. Heat is absorbed in the cooler from the atmosphere, products, etc., by virtue of the fact that the refrigerating medium is lower in temperature. This same heat must be similarly removed from the refrigerating medium by introducing a secondary medium materially lower in temperature. Since there is no secondary cooling medium available of a lower temperature than the refrigerating medium, even at the temperature at which it returns from the cooler after having absorbed large quantities of heat, it becomes necessary to raise the thermal level of the heat in the refrigerating medium sufficiently to allow it to gravitate into the cooling medium. This is effected in the absorption machine by the direct application of heat in the "generator," and in the compression system by the performance of work in the "compressor."

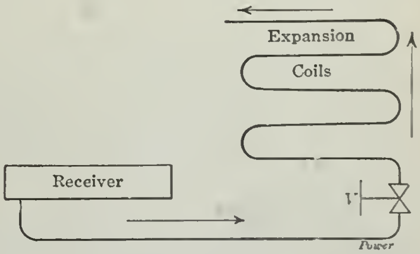


Fig. 7. Conventionalized Diagram of Fig. 6

In the operation of a compression system, almost any gaseous working medium might be employed. In practice, however, the list is limited to only such gases as are capable of being liquefied under ordinary natural temperatures and not too high mechanically produced pressures. Judging from the relative number of commercial installations employing the different media, one may assume that under the average conditions anhydrous ammonia comes nearer to fulfilling all the requirements of a practical working medium than any other.

The system previously described and illustrated in Fig. 3 consists of a single member, i.e., a containing vessel for the working medium. The refrigerating process might have been carried out just as well had there been no outer containing vessel and no second liquid. In that case the heat would pass from the air surrounding the test tube, directly to the ammonia, just as the air of a cold-storage compartment is cooled by "direct expansion," a system which is differentiated from the brine system by the location of the "expansion coils" containing the boiling or expanding ammonia, in direct contact with the atmosphere of the cold-storage rooms.

The commercial system corresponding to the one illustrated in Fig. 4, in which the evaporation of the ammonia cools the surrounding water, which in turn cools the surrounding air, is the "brine system," which takes its name from the fact that the cooling effect of the refrigerating medium is expended on a more or less uncongealable solution, such as sodium chloride ($NaCl$) or calcium chloride ($CaCl$) brine, in which the cooling solution circulated through a secondary system of coils in the cold-storage compartments is made the vehicle for conveying the refrigeration to, or, more properly speaking, the heat from, the atmosphere in the cold-storage rooms.

In order to make the systems illustrated in Figs. 3, 4, 5 and 6 commercially practicable, means must be provided for converting the gasified anhydrous ammonia back into the liquid state. In other words, after the ammonia has evaporated in the expansion pipe coils and has absorbed its full complement of heat, much as the sponge sucks up its fill of water, the heat and water with which the ammonia and a sponge are respectively charged must be squeezed out before they can again perform the function of absorption. In the case of the sponge, this is accomplished by the simple application of pressure. In the case of ammonia, pressure must not only be applied, but, since heat can be made to flow only from a relatively warmer to a relatively cooler substance, some cooling medium must be provided as well. The two cheapest and otherwise most convenient natural cooling media to be obtained are air and water. On account of the low specific heat of the former and the fact that it is usually at a higher temperature than water found in the same locality, we are practically limited to the use of water.

Before the ammonia can liquefy, all of the heat absorbed in the cold-storage rooms must be squeezed out, or rather be induced to flow out of the ammonia into the cooling water. From this it appears that were the water sufficiently cold, i.e., colder than the gaseous ammonia returning from the cooler, the flow would take place without the increase in pressure and temperature, and there would be no need for a compressor. On the other hand, if so cold a medium were available, it would be used directly for absorbing heat in the place of the ammonia or the ammonia and brine, as it would then be sufficiently cold to induce a flow of heat from the products to be refrigerated.

Since the pressure and consequent temperature of the cold ammonia gas returning from the expansion pipes must be raised before a heat transfer can be made to take place between it and cooling water at ordinary temperatures, the use of a compressor or suitable gas pump, in the case of the compression system, and some other means of heating and increasing the pressure of the refrigerating medium, in case of the absorption system, must be employed. In either system there must also be provided a suitable cooling chamber or condenser in which the cooling water can be brought in sufficiently close proximity to the refrigerating medium to allow the necessary heat flow from the hot ammonia to the cold water to take place.

Fire and Water Tube Boilers

By William Westerfield

Notwithstanding claims to the contrary, soot does not affect the efficiency of the fire-tube more than that of the water-tube boiler. Fire-tube boilers are less prone to prime, and, other things being equal, they are as efficient as the water-tube type.

In this article it is not the writer's purpose to speak in condemnation of any class of equipment, but to call attention to the fact that the representations made in favor of any class of power-plant equipment, or any other for that matter, by those engaged in the building or selling of it, must not be given too much weight. The same human frailty that leads the ordinary worker to imagine himself to be worth more than he really is, leads the builders and sellers of equipment to believe in the merits of their goods more fully than is really warranted, and to overstate them. It also leads them to see theoretical advantages, and to overlook practical disadvantages in the things they are building or selling. The situation in which they are placed, as advocates for something in which they have a lively personal interest, naturally leads them to take a more or less one-sided view, so that they are prone to give all the facts favorable to their output, but omit all that are not favorable. This treatment, when it is taken "as directed" makes a strong case for a weak proposition.

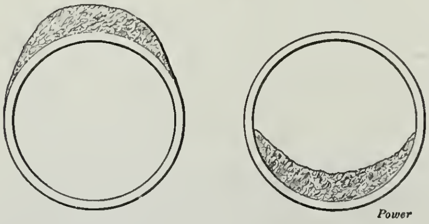


FIG. 1 FIG. 2

ASSUMED FORMATION OF SOOT ON TUBES

I have in mind a claim made for the water-tube boiler, which in theory seems all right, but which in practice does not stand up. It is claimed that in the fire-tube boiler the soot accumulating in the tubes has a greater insulating effect than it has in the case of the water-tube boiler. This is usually illustrated as in Figs. 1 and 2. In Fig. 1 is shown the tube of a water-tube boiler, with the accumulation of soot and ashes on the top of the tube, where, according to the claim, it does not interfere with the draft as it would in the fire-tube boiler. This might be true, but as a matter of fact it is not. In the first place, there is no excuse for allowing the tubes in any type of boiler to get into the condition shown. It may be that boilers are run that way in some "Arkansaw" sawmill plant of 10-horsepower capacity, but in any plant operated by an engineer, never. Fig. 2 is employed to show how the accumulation forms in the fire tube, showing a tube half full of ashes. If this condition is permitted, then we must admit that the draft will be seriously interfered with, and the steaming capacity of the boiler considerably reduced, no matter of what type it may be.

But these illustrations fail to illustrate, as well as the descriptions fail to describe the actual conditions. Any engineer knows that soot does not hunt for the top or bottom of a surface to the extent indicated. On the contrary it sticks whereever it lands, whether it is top or bottom. If you open the smoke-box door of a fire-tube boiler you will find that up to a certain point the accumulation will be of almost uniform thickness all around the inner surface of the tube. When it is allowed to accumulate, say, for a week, it is likely that the mass would fall to the bottom surfaces, and fill up the tubes as indicated. But why take an abnormal condition for the purpose of illustration, when it is not likely that condition will exist in one case in a million? And for the same reason, the soot accumulation does not form on the top of the water tube and leave the lower surfaces clean as indicated, but forms almost evenly all over its outside surface.

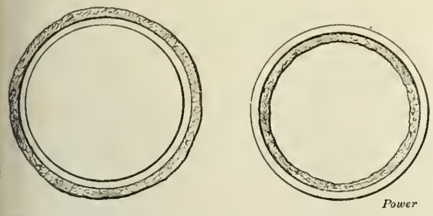


Fig. 3. Fig. 4

Actual Formation of Soot on Tubes

The manner in which the accumulations really form is shown in Figs. 3 and 4 respectively. The fire-tube boiler has the advantage of being easily reached in every part of the tube and shell surfaces, inside and outside. The scraper used for the inside of the tubes of fire-tube boilers is familiar to all engineers and firemen. For the outside of the tubes the writer has used scrapers of the form shown in Fig. 5. These are made of two lengths for convenience, the shorter one usually about 12 and the other about 18 inches long, and by their use all parts of the outside surfaces of the tubes may be reached and cleaned easily and thoroughly. The inside edge of the curved part of this scraper is made sharp in order to enter the scale more readily and remove it. These may be made in any shop where there is a forge and hammer.

In order to get good results with any type of boiler, it is imperative that the heating surfaces be kept clean, inside and outside. When it is understood how easy it is for the firemen to blow the soot and ashes from the boiler flues or tubes with the apparatus now furnished for that purpose, no excuse can be found for allowing them to fill up, or to accumulate sufficient soot and ashes seriously to affect the steaming of the boiler. The writer has never seen a water-tube boiler that could be cleaned in this manner as quickly and as easily as the old return-tube or marine boiler of the Continental type. In five minutes the fireman can open the smoke-box doors and blow the accumulation from every tube in a large boiler of either of the above types, and this repeated twice in twelve hours will keep the tubes in very good condition in so far as soot is concerned. But he has yet to see the water-tube boiler that can be cared for in this respect so quickly.

In the fire-tube boiler the blower is used from the front, where plenty of working room must be provided for other reasons; hence no extra room need be provided to operate the cleaning apparatus. In the water-tube boiler settings, the openings are usually located in the sides, as well as front and rear, so that it is necessary to allow considerable space between settings to enable the men to get between them when using the blower.

In Doctor Bell's excellent work on "Electric Power Transmission," on page 330 there is a table of recorded boiler performances. The best performance with any fuel is stated to be 13 12 pounds of water evaporated per pound of fuel, for a return-tube, and 13.01 for a water-tube boiler. On Cumberland coal the performance of the return-tube boiler was 12.07 pounds evaporation per pound of fuel, and that of the water-tube boiler, 10.79 pounds per pound of fuel. Quite likely there are other records that show the water-tube boiler in a better light, but the preceding ought to go far toward convincing anyone that the race of the "old reliable" that our forefathers got along with is not fully run.

In the writer's opinion, the water-tube boiler is especially to be recommended for large installations and for high pressures. For medium-size and small power plants, and for pressures up to, say, 150 pounds, the writer prefers the fire-tube boiler. They are more easily cleaned, can be cared for by men of less skill than is required for the water-tube boiler, and are less liable to give trouble when unskilfully handled. They are not so likely to prime when forced, due to the fact that having a larger relative volume of water from which to generate steam, the ebullition is not so active, hence there is less tendency to throw up spray to become entrained in the steam current to the outlet. This is one great advantage in itself. In the water-tube boiler, the ebullition being very active and the steam bubbles being formed so rapidly and filling so much of the water space in the boiler, the water level is caused to rise abnormally high when the boiler is being forced, and as soon as the forcing ceases the level will fall abnormally low.

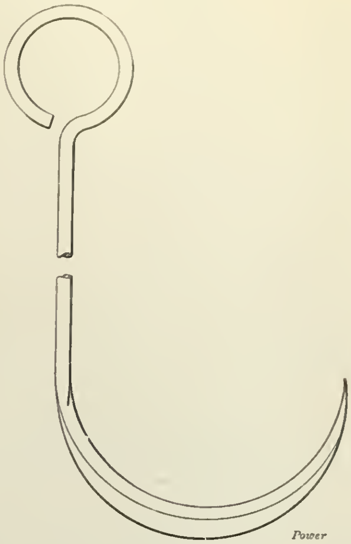
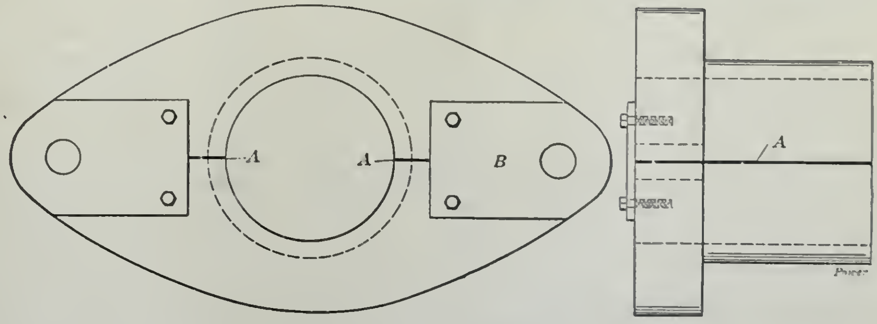


Fig. 5. Tube Scraper

Split Stuffing Box Gland

While looking around a power plant in Chattanooga, Tenn., the writer noticed the method, illustrated herewith, of making a split gland in a case where it was found necessary to devise some method of permitting the glands to be taken from the piston rod of a pump without taking the pump apart.

The gland was of the ordinary type sawed through the center lengthwise, as shown at *A*. In order to secure the two sections after being placed on the piston rod, two iron reinforced pieces were made and drilled as shown at *B B*. The larger holes were for the passage of the gland studs, and the two smaller holes for the two bolts which screw into holes drilled and tapped in the face of the gland. By removing two bolts and the stud nuts, the gland can be easily removed.



Gland and Plates

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Installing Induction Motors

By R. H. Fenkhausen

The Connecting Wires

When considering the installation of an electric motor, the first problem encountered is the calculation of the supply circuit. In calculating the proper size of wire to use for a given induction-motor installation, a more careful consideration of the factors involved is necessary than would be required in a direct-current installation.

In the first place the relation of the supply voltage to the motor voltage requires careful study, because the torque developed by any induction motor is proportional to the square of the voltage and a very small difference in voltage results in a marked difference in the torque of the motor, especially at starting. Most commercial induction motors will operate on voltages within 10 per cent. of their rating, but with impaired operating characteristics. To enable a closer selection of voltage one large manufacturer puts out two lines of motors, one for 200-400 volts and one for 220-440 volts, which allows the choice of a motor suitable for voltages of 180-240 and 360-480. When the voltage of the line is below the motor voltage the wiring to the motor must be calculated for as small a drop as practicable in order that the available voltage will not be greatly reduced by excessive line losses. If the line voltage is higher than the motor voltage a drop of 5 to 10 per cent. is permissible in the wiring and results merely in the loss of that much power, which may be offset by the saving in copper, especially for large motors or long distances.

The power factor of an underloaded motor is improved by a reduction in voltage, which fact should be considered when installing a motor large enough for future extensions. A three-phase motor with its windings normally delta connected may be connected star- or Y-fashion until the load increases to a value requiring the full horsepower, and an improvement in power factor secured. The torque will, of course, be reduced but it will be sufficient for the light load.

For induction motors not exceeding 25 horsepower in capacity, the familiar direct-current formula,

TABLE 1. FULL LOAD CURRENTS PER PHASE.

Average Power Factor and Efficiency.

	110 Volts.			220 Volts.			440 Volts.			550 Volts.			
H.P.	1-Phase.	2-Phase.	3-Phase.	1-Phase.	2-Phase.	3-Phase.	1-Phase.	2-Phase.	3-Phase.	1-Phase.	2-Phase.	3-Phase.	H.P.
2	23.80	10.10	11.54	11.90	5.05	5.77	5.95	2.52	2.89	4.76	2.02	2.30	2
5	52.30	22.92	26.50	26.15	11.46	13.25	13.07	5.73	6.63	10.46	4.58	5.30	5
7½	68.75	34 42	33.70	34.37	17.21	19 85	17.19	8.60	9.93	13.75	6.88	7.94	7½
10	90.60	45.30	52.40	45.30	22.65	26.20	22.65	11.32	13.10	18.12	9.06	10.48	10
15	132.80	66.40	76.80	66.40	33.20	38.40	33.20	16.60	19.20	26.56	13.28	15.36	15
20	175.20	87.40	101.30	87.60	43.70	50.70	43.80	21.85	25.35	35.04	17.52	20.26	20
25	219.00	109.60	126.70	109.50	54.80	63.40	54.70	27.40	31.70	43.80	21.90	25.34	25
35	321.00	160.50	185.80	160.50	80.20	92.90	80.00	40.10	46.40	64.20	32.10	37.15	35
50	428.00	214.00	247.20	214.00	107.00	123 60	107.00	53.50	61.80	85.60	42.80	49.44	50
75	656.00	328.00	379.10	328.00	164.00	189.50	164.00	82.00	94.70	131.20	65.60	75.82	75
100	848.00	424.00	487.20	424.00	212.10	244.10	212.20	106.10	123.40	169.60	84.80	97.44	100
125	1067.00	533.50	613.90	533.50	266.90	307.50	266.90	133.40	155.10	213.40	106.70	122.78	125
150	1300.00	650.00	751 00	650.00	325.00	376.00	325.00	162.50	187.40	260.00	130.00	150.20	150
200	1696.00	848.00	974.40	848.00	424.00	487.20	424.00	212.00	244.00	339.20	169.60	194.80	200
250	2134.00	1067.00	1228.00	1067.00	533.50	613.90	533.50	266.90	307.50	426.80	213.40	254.60	250
300	2600.00	1300.00	1502.00	1300.00	650.00	751.00	650.00	325.00	376.00	520.00	260.00	300.40	300

TABLE 2. MOTOR WIRING.

For a three-phase motor, use the current per wire of a corresponding two-phase motor, and determine the size of wire as though for that motor.

Approximate Loss in Per Cent., at Different Standard Voltages.				Current per Wire × Distance in Feet for Ten Standard Wire Sizes.									
110 V.	220 V.	440 V.	550 V.	No. 0.	No. 1.	No. 2.	No. 3.	No. 4.	No. 5.	No. 6.	No. 8.	No. 10.	No. 12.
1%				5,300	4,200	3,320	2,630	2,090	1,650	1,310	825	520	325
2	1%			10,600	8,400	6,640	5,260	4,180	3,300	2,620	1,650	1,040	650
3	1½			15,900	12,600	9,960	7,890	6,270	4,950	3,930	2,475	1,560	975
4	2	1%		21,200	16,800	13,280	10,520	8,360	6,600	5,240	3,300	2,080	1,300
5	2½	1¼	1%	26,500	21,000	16,600	13,150	10,450	8,250	6,550	4,125	2,600	1,625
6	3	1½	1¼	31,800	25,200	19,900	15,780	12,540	9,900	7,860	4,950	3,120	1,950
8	4	2	1⅝	42,400	33,600	26,560	21,000	16,700	13,200	10,480	6,600	4,160	2,600
10	5	2½	2	53,000	43,500	33,200	26,300	20,900	16,500	13,100	8,250	5,200	3,250
	6	3	2⅜	63,600	51,400	39,840	31,550	25,080	19,800	15,720	9,900	6,240	3,900
	8	4	3¼	84,800	67,200	53,120	42,000	33,400	26,400	20,960	13,200	8 320	5,200
	10	5	4	106,000	84,000	66,400	52,600	41,800	33,000	26,200	16,500	10,400	6,500
		6¼	5	132,500	105,000	83,000	65,750	50,200	41,200	32,750	20,625	13,000	8,125
		8	6½	169,600	134,400	106,240	84,000	66,800	52,800	41,900	26,400	16,640	10,400
		10	8	212,000	168,000	132,800	105,200	83,600	66,000	52,400	33,000	20,800	13,000
			10	265,000	210,000	166,000	131,500	104,500	82,500	65,500	41,250	26,000	16,250

$$\frac{22\,I\,D}{V} = c.m.$$

nay be used and the result increased by ne size of the Brown & Sharpe gage to over the inductive drop, etc. In this ormula $c.m.$ = circular mils of wire ross-section; I = amperes per phase at ull load; D = length (in feet) of one wire, V = the drop in volts, and 22 is the resistance per mil foot of a two-wire circuit, allowing for sag or other irregularities.

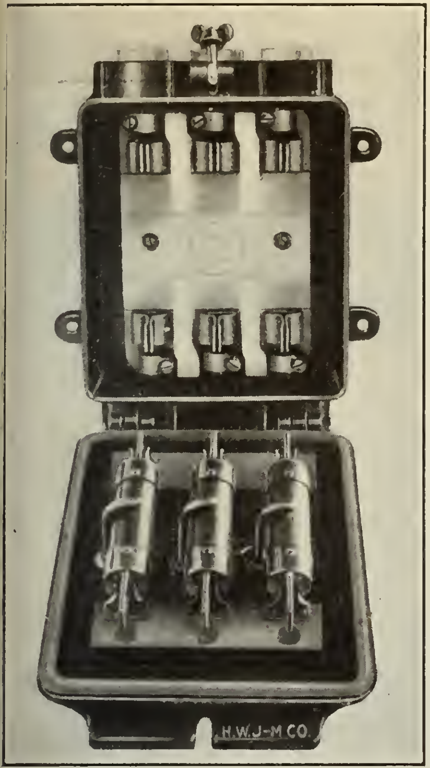


Fig. 1. Inclosed Fuse Box

The full-load current in amperes is usually given on the motor name plate, but may be taken from Table 1. This should be increased by 50 per cent. to allow for overloads, before applying the formula. Table 2 will serve to reduce the calculation required in most cases. This gives the relation between the distance × amperes per wire and the approximate loss in percentage of the voltage at the motor, for wires of standard sizes from No. 0 to No. 10.

For example, suppose a 15-horsepower two-phase motor is to be located 150 feet from the nearest point on a 220-volt local main, and it is desired to keep the loss down to 3 per cent. maximum, because the voltage is not always right up to normal. From Table 1 the full-load current is 33.2 amperes per wire, and adding 50 per cent. for overloads gives practically 50 amperes maximum. The distance × current = 150 × 50 = 7500. Starting at 3 per cent. in the 220-volt column of Table 2, follow that line into the right-hand section of the table until the nearest number to 7500 is found; that is 7875, and it is in the No. 6 column, so that No. 6 is the size of wire that would be used for direct current. Taking one size larger, to allow for inductive drop, gives No. 5 as the size to be used in the case assumed.

The value of the secondary current for multiple-speed motors should be obtained from the manufacturers of the motor before attempting to install the secondary wiring between the motor and the controller.

For very large feeders supplying a large number of motors, it is preferable to use several small cables in parallel rather than one large cable, because the inductive drop in one large wire supplying a load of low power factor is considerably greater than that in several smaller wires of equal total cross-section. This feature does not become important, however, until the size of the large conductor is greater than No. 00 or 000 Brown & Sharpe gage. The inductive drop in stranded cables is greater than that in solid wire of equivalent cross-section, due to their greater diameter, and also to the helical twist given the stranding. The difference is so small, however, that it is negligible except for long transmission lines.

In alternating-current conduit work all the wires of a circuit must be in one pipe, or else the drop will be prohibitive and heating of the conduit will result. For open work the different wires of a circuit should be located as close together as circumstances and voltage will permit, as the inductive effect is lessened by reducing the distance between the wires.

Overload Protection

Nearly all modern motor-starters for use with squirrel-cage induction motors are provided with two sets of taps, one for running and one for starting.

The fuses in the running connections should be rated about 25 per cent. larger than the full-load current of the motor. This will allow the fuses to blow when the motor is overloaded over 25 per cent. for a long period, but as a fuse will carry 50 per cent. overload for several minutes there is no danger of the fuse opening the circuit from brief overloads which the motor could easily carry without harm.

The starting current of a squirrel-cage motor is several times the full-load running current, but owing to the fact that the starting period is of short duration and a fuse takes an appreciable time to reach the melting temperature, the starting fuses need not be as large as they would have to be if they blew the moment the overload occurred.

Where circuit-breakers are installed instead of one set of fuses it is preferable to use them in place of the running fuses rather than the starting fuses, in order that the time-element feature of the fuses may be utilized during the starting period. Ordinary circuit-breakers, such as are usually installed for motor protection, open instantly when the current for which they are set is exceeded, and this lack of time lag makes the necessary current setting too high for proper protection during the starting period, but is not a serious objection when running. A few manufacturers mount the running fuses directly on the starter, but for the reasons given above the connections of these fuse blocks should be changed to the starting position and larger fuses installed for use during the starting period.

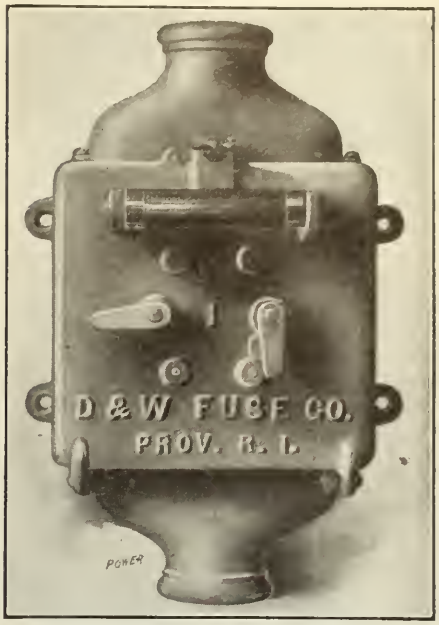


Fig. 2. Bell Mouths on Fuse Box

Another important advantage obtained by the use of a circuit-breaker in the running position is that if a careless operator should throw the starter to the running position before the motor is up to speed, that will not cause a delay such as would be the case were it necessary to replace the blown fuses; the circuit-breakers may be closed and a fresh start made before the motor has come to rest.

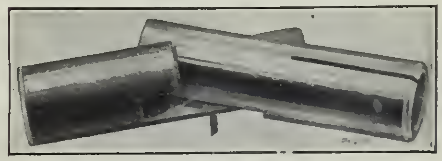


Fig. 3. A Good Form of Connector

Although the average employee outside of electrical workers regards a circuit-breaker with awe and will not tamper with it for any price, there are many cases where a machine operator will raise the setting of the breaker so that he will not be troubled to reclose it when opened by the careless handling of his starter. To prevent this practice, I have adopted the expedient of sealing the adjustment screws of all circuit-breakers with lead seals similar to those used on meter cases.

PROTECTION OF LIVE PARTS

All live parts of a motor installation should be protected so that workmen cannot come into contact with them accidentally or carelessly. This applies equally to high- and low-voltage circuits, because it is not always the actual shock which causes injury in accidental contact with a circuit. The natural recoil which follows even a small shock is often the cause of a man's fall from an elevated platform or of his coming into contact with gears, belting, etc., which result in serious injury.

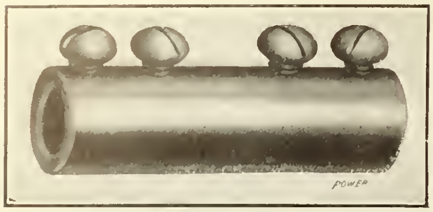


FIG. 4. A POOR FORM OF CONNECTOR

The switch should preferably be of the oil-break type. A convenient arrangement is an oil switch with automatic overload and no-voltage release magnets.

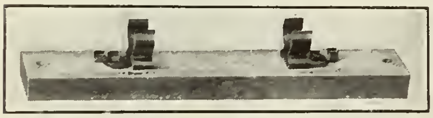


FIG. 5. CARTRIDGE FUSE HOLDER

The fuses should be of the cartridge type inclosed in an iron box, as indicated in Figs. 1 and 2, with clamps by means of which the fuses may be withdrawn for inspection or renewal when the box is opened as shown in Fig. 1. This not only permits the safe renewal of fuses by any person, but also serves as an additional switch in case of trouble.

Where fuses and circuit-breakers are both installed, the fuses should always be on the line side of the circuit-breaker, which will allow repairs to be made on the circuit-breaker when the fuses are out, and also allows the fuses to protect it in case of a flash-over due to lack of oil or other causes.

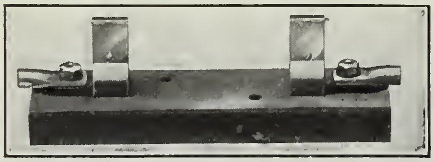


FIG. 6. HOLDER FOR LARGE FUSES

Where conduit work is installed, bell mouths, such as those shown in Fig. 2, should be used at all fuse boxes and circuit-breakers. This makes the conduit job continuous and protects the wires from mechanical injury. Where it is necessary to resort to open work, as in most types of auto-starters, a condulet with a porcelain cover having the required number of holes should be used to terminate the pipe.

The leads from auto-starters and motors should not be permanently soldered to the connecting wires. Knife-blade connectors of the type shown in Fig. 3 should be used instead; their extra expense will be more than offset in the time saved by their use whenever any of the apparatus requires change of connections or replacement while under repair. Instead of taping up the connectors, a better, neater and more accessible job will be made if a piece of fiber tubing, such as the casing of a blown fuse, be slipped over each connector and held in place by one or two turns of tape at each end. Connectors of the type shown in Fig. 4 are disapproved by the underwriters and should never be used on permanent work, and switches, fuse blocks, etc., having binding posts of the type where the wire is held in a drilled hole by the pressure of a screw point should be avoided. The contact secured is so small that the screw easily loosens under vibration and an arc starts between the wire and the binding post. Several cases have come under my observation where this arc has increased, unnoticed, until the heat of the wire ignited the insulation; the liability to cause a serious fire this way is obvious.

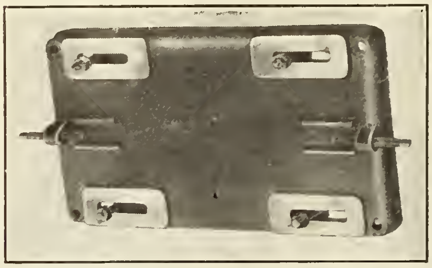


FIG. 7. MOTOR "BASE" FOR CEILING MOUNTING

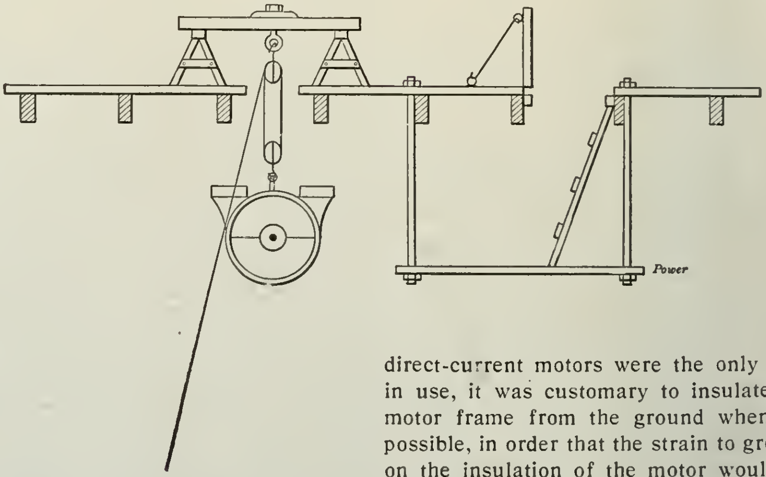


FIG. 9. METHOD OF PLACING INVERTED MOTOR

For currents not exceeding 30 amperes, the underwriters approve washer and screw connections of the type shown in Fig. 5, with spring jaws to take fuses having ferrule terminals. For currents exceeding 30 amperes, the soldered lugs of the type shown in Fig. 6 must be used at all switches and fuse blocks, and for currents exceeding 60 amperes knife-blade terminals to fit jaws of the type shown in this figure are required on all fuses.

A few years ago motors used to be equipped with exposed terminals, but now nearly all motors are furnished with flexible leads and connectors of the type shown in Fig. 3.

FIG. 8. DRIP PAN FOR CEILING MOTOR

This greatly reduces the liability of accidental short-circuits, and any strain on the terminals is prevented from injuring the interior connections of the motor by a clamp of insulating material applied to the flexible cables.

LOCATION OF MOTORS

When motors are mounted on machine tools for direct connection or gearing, or on structural-steel foundations attached to a steel-frame structure, the motor frame will, of course, be grounded. In the early days of the electric drive when direct-current motors were the only kind in use, it was customary to insulate the motor frame from the ground whenever possible, in order that the strain to ground on the insulation of the motor would be minimized and also to permit the temporary operation of a grounded motor. The superior quality of insulation used in modern motors makes the windings, etc., very unlikely to become grounded, and it has therefore become standard practice to ground the frames of all induction motors, a special ground wire being run when motors are mounted on a wooden foundation. Common humanity demands this in the case of high-voltage machines, but even on low voltages a shock may result in serious injury, as explained in the previous article.

Another reason for the grounding of induction-motor frames is to prevent the accumulation of static charges which occur on an insulated motor, due to its static capacity with respect to the ground. Varying-speed crane motors of the wound-rotor type appear to be especially liable to this trouble, and several cases have come under my observation, when these motors were mounted on a wooden-frame crane, where the crane could not be operated until the frame of the motor was grounded. The shocks only occurred occasionally, but the workmen would not handle the crane until the ground wire was installed. If a direct ground on the frame is not desired 500 to 1000 ohms of dead resistance may be connected in the ground wire, which will allow the static charges to escape, but the motor frame may have a dangerous potential above the earth, in case of an actual ground in the windings.

The location of a motor will naturally be governed by local conditions, but a clean, light and well ventilated place must be chosen, if repairs are to be kept down to the minimum. Lack of ventilation reduces the load the motor can carry and a dark, dirty or inaccessible location renders it liable to be neglected. When the saving in floor space renders it advisable to place the motor on a side wall, or a better arrangement of belting can be made by installing the machine on the ceiling, some means of ready access should be furnished or the motor is sure to be neglected. For ceiling installation a special bedplate is required, as shown in Fig. 7, with a drip pan (Fig. 8), held in place by an eyebolt, to prevent surplus oil from dropping to the floor or machines below.

The easiest way to install a ceiling motor is to tap a hole in the bottom of the frame, if one is not already there, and cut a small trap in the floor above, through which the tackle may be dropped to hoist the motor into place, as represented in Fig. 9. For very high ceilings another trap may be cut to one side, to give access to the motor for inspection, and a working platform hung under this trap, as in Fig. 9.

LETTERS

The Three-wire versus the Two-wire Generator

At several engineers' clubs during the past month, I have heard the subject discussed as to whether the three-wire direct-current generator with collector rings is better than the two-wire generator with a motor balancer. Personally, I believe the three-wire machine to be the better for the following reasons:

With the three-wire generator there is a saving in that only one machine has to be bought instead of two or three; lower maintenance, because there are fewer parts to be cared for; and higher efficiency, because one machine has smaller losses for a given output than two or three machines.

Some engineers argue that the fluctuations in voltage, due to a continually changing load, cannot be overcome without a motor balancer. I have five electric elevators running from a 250-kilowatt generator and drawing 1000 amperes. Sometimes the load of all the elevators is thrown on at once, but the voltage drop is not noticeable on either the lights or the engines. This shows that a three-wire generator can handle a sudden load just as well as a compensating set. At times I have seen a drop or rise in voltage, but this was through no fault of the generator; it was due to the governor not being sensitive enough to respond to the sudden change.

Furthermore, when the circuit breaker goes out, the place is in total darkness until the compensator is built up again. On a three-wire machine, the pilot lights remain lighted and it is only necessary to pull the main switch, throw in the circuit breaker, then close the main switch, and the load is on. I believe this to be the best and simplest way, but would welcome other opinions upon the subject.

Walter N. Smith.

Chicago, Ill.

Overheating of Copper Plated Dynamo Brushes

Concerning the burning of brush faces on generators, of which there has been some discussion lately, an experience which I once had with this kind of trouble may be of interest. The machine was a 150-kilowatt direct-current compound-wound generator, having six sets of brushes, four in each set, and delivering 250 volts at 200 revolutions per minute. When this machine was started some of the brushes became white hot in spots near the tips of the brushes. It was at first thought this was due to poor contact between the brushes and the commutator but examination proved the contact good. The burning would change from one brush to another until nearly all were destroyed in the center, which took about three days. Sometimes a brush would burn when carrying about one-third of the load and would not burn at full load. Different grades of brushes were tried but that did not help the trouble any. There was no sparking except at the brushes which were heated. The holders were of the box type and had the usual shunt or "pigtail" connections to the brushes.

While watching the machine one day it occurred to me that perhaps the vibration of the brushes in the holders might improve the contact between the holder and the copper plating of the brush and lower the resistance there so much that the brush would carry more than its share of the current. Scraping off the copper plating on the brushes proved this reasoning to be sound and corrected the trouble. The machine has been running with full load for three years since the copper plating was scraped off and the same brushes are in use yet. The pigtails are of ample size to carry the current without depending on contact between the brushes and the holders. They are clamped on and the plating was not removed under the clamp. I have sometimes tried a single copper-plated brush on this machine and it immediately becomes white hot at the contact end.

P. L. Werner.

McKeesport, Penn.

An Unbalanced Air Gap and a Powerless Motor

A day or two ago I had an experience with a 35-horsepower three-phase induction motor which was new to me. The moter is in an out of the way department and an employee came running to the engine room to say that the motor had "gone up." I hurried to the spot and found that the foreman had shut off the current and the motor was at a standstill. I was told that the motor had made a noise like a circular saw and lost speed. I got up on the suspended platform on which the machine was mounted and found the bearing on the belt side so hot that the babbitt had started to run. Pulling off the belt, I turned the motor shaft by hand and found that it turned freely, and apparently without the rotor touching the stator at any point. I put the compensator handle at the "starting" position and the motor ran at about 15 revolutions per minute and hummed a great deal. Next I tested all the fuses and the compensator fingers and found them all right; tests for open circuits and grounds also failed to show anything wrong. But the motor simply would not run normally.

Then I took the rotor out and found that it had rubbed at one spot on the stator, but had done no harm. After the burned bearing had been babbitted and fitted to the shaft again, the motor was reassembled, the current turned on and away she went as smoothly as a top. Now the question is why would not the motor run at first? All I did that would affect its operation at all was to put the rotor in the center of the stator bore, equalizing the air gap all around. I wish someone would explain why lowering one end of the rotor until the air gap is nearly all on one side at that end apparently robbed the motor of all its power.

R. Manly Orr.

Brantford, Ont.

Gas Power Department

Elementary Lectures on the Gas Producer

By Cecil P. Poole

Peat

As stated briefly in the last lecture,[*] peat is merely decomposed vegetable matter, consisting chiefly of decayed moss and water plants. Most of the peat is found in bogs, those in fairly high altitudes containing decayed swamp moss and small forest plants while those in low altitudes, located near rivers and creeks, contain decayed grasses, leaves and water plants. As would be expected, fresh peat is so moist that it is useless for fuel; it must be dried out considerably before it will burn readily, and in order to be of commercial value as a fuel it has to be put through an elaborate process of drying and pressing.

Even when dried as far as practicable, peat contains a large proportion of moisture in comparison with lignite and coal. Two typical analyses of air-dried peat are given in Table 2 herewith. The "heat values" given at the foot of the table are the numbers of heat units liberated by the combustion of one pound of peat.

TABLE 2. TWO PEAT ANALYSES.

The figures are percentages, by weight, of air-dried peat.

	Per Cent. Bohemian.	Per Cent. Irish.
Carbon	39.26	53.18
Hydrogen (available)†	3.88	5.54
Nitrogen	1.36	1.91
Moisture	26.26	25.12
Ash	17.27	9.93
Unknown	11.97	4.32
Heat value	68.40	94.15

Dried peat uncompressed is very porous and ignites readily at about 375 degrees, Fahrenheit. It burns in the open air with a long, smoky flame. Because of the expense of drying and pressing it, and the high cost of transportation (due to its bulkiness), peat is not widely used for fuel.

Lignite

Lignite is commonly called "brown coal" because of the color of most lignites. There are, however, black lignites as well as brown. Table 3 gives representative derived analyses[‡] of lignites, which show the wide range of moisture and ash contained in the fuel. The North Dakota sample, for instance, contained over 35 per cent. of moisture and 9.3 per cent. of ash, whereas the Wyoming peat contained only 9.4 per cent. of moisture and nearly 21 per cent. of ash.

> **Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.**

The "heat value," given at the bottom of each column of the table, is the number of heat units that are "liberated" or "evolved" by burning one pound of lignite.

The percentages in the body of the tables are the proportions, by weight, of the various substances in the lignite. For example, the Montana lignite showed 59.08 per cent. of carbon; that means that in each pound of the lignite there was 0.5908 pound of carbon, or in 100 pounds of the lignite there were 59.08 pounds of carbon. Some of this was solid carbon, called "fixed" carbon, as explained in the last lecture, and the remainder was gaseous carbon, combined with hydrogen to form hydrocarbon gases, but this feature of the carbon content need not be taken into consideration just now, beyond the fact that the hydrocarbon gases which contain the gaseous portion of the carbon also contain a portion of the available hydrogen.

Bituminous Coals

From the name, the student would naturally suppose that bituminous coals contain bitumen, but they don't. In early times, when the scientific knowledge of geology in general, and coals in particular, was very imperfect, it was thought that certain soft coals contained bitumen because they felt resinous to the touch and burned very freely and with a long flame. Hence, they were called bituminous, and the name has clung to all soft coals above the grade of lignite.

There are two general classes of bituminous coals, known as "caking" and "noncaking." Caking coal, when thrown on a fire, swells up and the lumps fuse together, or "cake," in a sort of spongy mass which must be broken up frequently to allow the draft to get through and the fire to get at the particles of coal. Noncaking coal does not behave that way.

Table 4 gives the derived analyses and heat values of twelve bituminous and three semi-bituminous coals which are fairly representative of these grades. A glance at the proportions of carbon and hydrogen will show that the ratio of carbon to hydrogen is considerably higher in semi-bituminous than in bituminous coals. A still further increase in this ratio carries the classification up to anthracites.

Anthracites

Anthracite is a hard, lustrous coal, composed almost entirely of carbon. Semi-anthracite comes between anthracite and semi-bituminous coals. It is mostly lumps, as it comes from the dealer, but the lumps can be crumbled by a good sharp blow, whereas "straight" anthracite is almost as hard as rock and is not easily fractured or crushed. This distinction, of course, is meant to apply to grades not very near the technical dividing line between the two classes. A semi-anthracite which lacks only 1 or 2 per cent. of coming within the anthracite class cannot be distinguished by sight or touch from an anthracite that is just over the dividing line between the two classes.

TABLE 3. ANALYSES OF SOME REPRESENTATIVE LIGNITES.

The figures in the table are percentages, by weight, of the lignite as received.

Where found	California.	Colorado.	Montana.	North Dakota.	Texas.	Wyoming.
Carbon	47.18	57.46	59.08	41.87	39.25	54.91
Hydrogen*	3.87	4.00	4.14	2.85	3.08	3.88
Sulphur	3.05	0.55	1.73	0.48	0.79	0.59
Nitrogen	0.66	1.15	1.33	0.69	0.72	1.02
Moisture	18.51	18.68	11.05	36.78	34.70	22.63
Ash	15.49	5.99	10.97	5.09	11.20	4.50
Unknown	11.24	12.17	11.70	12.24	10.26	12.47
Heat value	8,525	10,143	10,539	7,204	7,056	9,650

*Free hydrogen and that combined with gaseous carbon, forming hydrocarbons.

Heat Value of Coal

It is impossible to compute the actual heat value of coal from any analysis, because of the imperfection of all known methods of analysis, but a reasonably close idea of the heat value may be obtained from the derived analysis by simple calculation, as follows:

Add the percentages of carbon, hydrogen and sulphur and multiply their sum by 170, for lignites and soft coals. Deduct from this product 10 times the percentage of moisture.

For semi-anthracite, use 160 instead of 170 as the first multiplier.

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[*] The fourth of the series, which appeared in the issue of August 16.

[†] All hydrogen except that contained in the moisture.

[‡] See page 1476, issue of August 16.

For anthracite, use 150 *instead of* 170 *as the first multiplier.*

These rules are based on the assumptions that the total combustibles in lignites and soft coals average about 17,000 heat units per pound, those in semi-anthracite about 16,000 and those in anthracite about 15,000; and that the moisture in all fuels absorbs about 1000 heat units per pound when it is evaporated into steam. As the proportions are expressed in percentages, the heat values must be divided by 100 in order that the product of the two shall be correct. That is, multiplying 80 per cent. by 170 gives the same result that would be obtained by multiplying 0.80 by 17,000; and if coal contains 80 per cent. of combustible, each pound of it will contain 0.80 pound of combustible.

Expressing the rules in the shape of formulas:

For lignite and soft coal—

$$170\,F - 10\,M = \text{B.t.u.}$$

For semi-anthracite—

$$160\,F - 10\,M = \text{B.t.u.}$$

For anthracite—

$$150\,F - 10\,M = \text{B.t.u.}$$

In these formulas $F =$ the sum of carbon, hydrogen and sulphur percentages; $M =$ percentage of moisture, and *B.t.u.* = heat units per pound of lignite or coal.

The chart, Fig. 13, gives the gross heat for different percentages of total combustibles. In using it, it is necessary to subtract 10 times the moisture percentage from the result obtained from the chart.

EXAMPLES

The Colorado lignite referred to in Table 3 contained

	Per Cent.
Carbon	57.46
Hydrogen	4.00
Sulphur	0.55
Combustible	62.01

The heat liberated by burning these combustibles is approximately

$$170 \times 62 = 10{,}550$$

B.t.u., and as the lignite contained 18.68 per cent. of moisture, the heat used up in evaporating this into steam would be about

$$10 \times 18.68 = \text{practically } 187$$

heat units. The estimated net heat of the lignite, therefore, is about

$$10{,}550 - 187 = 10{,}363$$

B.t.u. per pound. The test, however, showed 10,143 (see Table 3), so that the estimated heat value is too high by

$$10{,}363 - 10{,}143 = 220 \text{ B.t.u., or}$$

$$\frac{220 \times 100}{10{,}143} = 2.17$$

per cent.

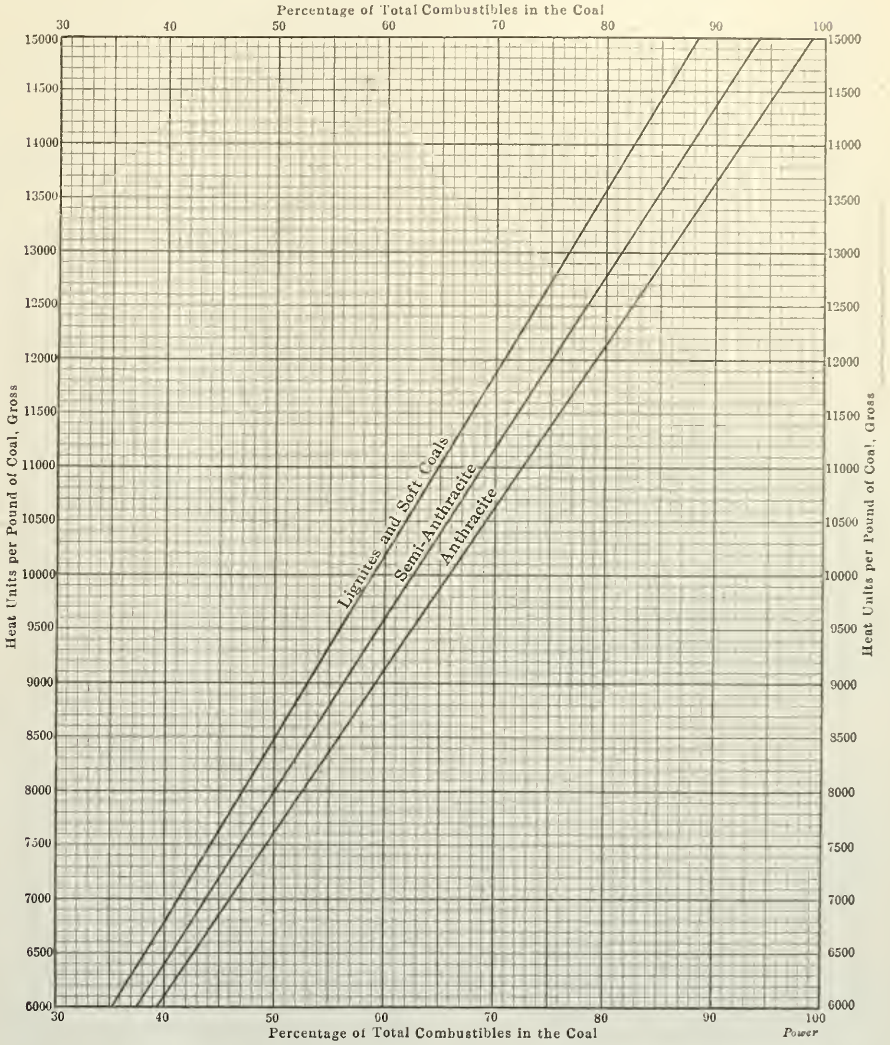


FIG. 13. RELATION BETWEEN TOTAL COMBUSTIBLE AND GROSS HEAT IN COALS

TABLE 4. ANALYSES OF REPRESENTATIVE SOFT COALS.

Bituminous.	Carbon.	Hydrogen.*	Sulphur.	Nitrogen.	Moisture.	Ash.	Unknown.	B.t.u.
Alabama	68.69	4.47	1.01	1.54	3.36	12.43	8.50	12,350
Arkansas	75.31	3.93	1.28	1.53	2.19	11.63	4.13	13,464
Illinois	60.74	4.00	1.32	1.15	12.91	11.64	8.24	10,804
Indiana	60.34	4.10	2.50	1.18	11.40	13.40	7.08	11,061
Indian Territory	62.34	4.24	3.96	1.36	6.24	13.21	9.45	11,228
Iowa	59.82	3.90	5.03	0.94	8.24	16.00	9.07	11,027
Kansas	61.88	4.23	6.27	0.92	4.18	17.91	4.61	11,642
Kentucky	65.29	4.51	3.52	1.40	7.92	10.06	7.30	12,022
Missouri	57.00	4.05	5.25	0.94	8.33	19.36	5.07	10,586
Ohio	72.03	2.93	2.19	1.07	8.25	6.72	4.81	13,591
Pennsylvania	76.15	5.26	2.52	1.56	1.52	6.42	6.57	14,762
West Virginia	78.00	5.09	0.90	1.54	1.75	6.34	6.38	14,107
Semi-bituminous.								
Arkansas	76.44	3.56	1.99	1.37	2.36	12.08	2.20	13,259
Pennsylvania	83.94	4.21	0.91	1.27	0.59	6.04	3.04	14,753
West Virginia	82.87	4.59	0.65	1.68	1.53	5.05	3.63	14,807

*Free hydrogen and that contained in hydrocarbons.

The Texas lignite contained

	Per Cent.
Carbon	39.25
Hydrogen	3.08
Sulphur	0.79
Combustible	43.12
Moisture	34.70

The gross heat is about

$$170 \times 43.12 = 7330 \text{ B.t.u.}$$

The heat absorbed by evaporated moisture is about

$$10 \times 34.7 = 347 \text{ B.t.u.}$$

The estimated net heat in the lignite, therefore, is

$$7330 - 347 = 6983$$

B.t.u per pound. The heat value by test was 7056, so that the estimated value is 7056 — 6983 = 73 heat units, or 1.03 per cent., too low.

The Illinois bituminous coal of Table 4 contained 60.74 per cent. carbon, 4 per cent. hydrogen, 1.32 per cent. sulphur and 12.91 per cent. moisture. Total combustible, 66.06 per cent.

$$170 \times 66.06 = 11{,}230 \text{ B.t.u.}$$

for the gross heat.

$$10 \times 12.91 = 129$$

B.t.u. lost in steam, and the estimated net heat is about

11,230 — 129 = 11,101

heat units per pound. The test showed 10,804, so that the calculated value is 2.75 per cent. high.

The Pennsylvania bituminous coal (Table 4) contained

	Per Cent.
Carbon	76.15
Hydrogen	5.26
Sulphur	2.52
Combustible	83.93
Moisture	1.52

The gross heat is about

170 × 83.93 = 14,268 B.t.u.,

and subtracting the heat lost in the steam (10 × 1.52 = 15) gives

14,268 — 15 = 14,253

heat units per pound, net, which is 3.45 per cent. less than the test figure of 14,762.

TABLE 5. ANALYSES OF REPRESENTATIVE HARD COALS.

	Pennsylvania Semi-Anthracite.	Colorado Anthracite.
Carbon	72.65	86.46
Hydrogen*	2.50	1.62
Sulphur	0.74	0.29
Nitrogen	0.77	0.75
Moisture	5.41	3.45
Ash	15.78	5.90
Unknown	2.15	1.53
B.t.u.	12,047	13,470

*Free hydrogen and that contained in hydrocarbons.

The West Virginia semi-bituminous coal contained

	Per Cent.
Carbon	82.87
Hydrogen	4.59
Sulphur	0.65
Combustible	88.11
Moisture	1.53

Gross heat of combustibles:

170 × 88.11 = 14,979

heat units, and the steam absorbs about 10 × 1.53 = 15 heat units, leaving

14,979 — 15 = 14,964

heat units per pound, which is 1.06 per cent. more than the test showed (14,807).

A Pennsylvania semi-anthracite contained

	Per Cent.
Carbon	72.65
Hydrogen	2.50
Sulphur	0.74
Combustible	75.89
Moisture	5.41

Gross heat = 160 × 75.89 = 12,142 B.t.u.
Steam absorbs 10 × 5.41 = 54 B.t.u.

Net heat per pound.........12,088 B.t.u.

The heat value obtained by test was 12,047 B.t.u. per pound.

A Colorado anthracite contained

	Per Cent.
Carbon	86.46
Hydrogen	1.62
Sulphur	0.29
Combustible	88.37
Moisture	3.45

Gross heat = 150 × 88.37 = 13,255 B.t.u.
Steam wastes 10 × 3.45 = 35 B.t.u.

Net heat, estimated 13,220 B.t.u.

The heat value obtained by test was 13,470 B.t.u. per pound.

The foregoing examples, of course, were figured in detail. The chart does not give readings as closely as the estimated values are computed, but it will serve very well to get at a rough approximation of the gross heat value, from which to subtract the heat used up in evaporating the moisture.

Some Points in the Practical Operation of Gasolene Engines

By A. L. Brennan, Jr.

The ignition system of any internal-combustion engine is a fruitful source of "trouble" and should be investigated whenever an engine stops or refuses to run without an evident cause. The wiring plans of different engines vary somewhat, but the general principle is the same in all of them, so I shall not go into this subject at all, but will assume that the engine is wired correctly.

The vibrator or "trembler" spring of an induction coil should be adjusted to the least possible tension consistent with reliable firing, and it will be found necessary to make slight changes from time to time, particularly when running on battery current, because as the battery voltage diminishes, the tension on the vibrator spring should be reduced. All contact points must be clean, flat and in line, to insure proper contact. Pitted points can be readily cleaned and "squared" by using a very fine flat file.

Before starting an engine the following details should be given careful attention. Fill all grease cups and lubricators. If the engine oils from the base, see that there is a sufficient quantity of clean oil in the base. Turn on the circulation water to the jacket, open the discharge outlet if necessary, turn on gasolene, close the switch in the ignition circuit and crank. If the engine refuses to start, any one of the following conditions may be the cause: A loose terminal or a disconnected wire, usually to or from the battery; damp cylinders; battery plug not in position; a carbonized or broken spark plug; dirt in the gasolene pipe; mixture faulty; water in the carbureter; the check valve in the carbureter not working; inlet valve stuck; poor compression; leak in the water jacket, allowing water to get in the cylinder; engine flooded with gasolene; battery run down; timer slipped on the shaft; muffler or exhaust pipe clogged.

This is a rather formidable list of troubles, but only a few of them occur in every-day operation. The best thing to do first is to look over the electrical system for a broken connection, and a good way to test for this is to close the switch, turn the engine over slowly and note if the coil vibrator buzzes (assuming that the jump-spark system is used). If the vibrator works, then the trouble cannot be in any part of the primary or low-tension circuit, but there may be trouble in the high-tension wiring, spark plug, or the secondary winding of the induction coil. To test the plug it will have to be removed from the cylinder and laid on the cylinder, with the wire attached to it as before; then turn the engine over and note if there is a good spark at the plug points when the vibrator buzzes. If the plug shows a large even spark of a bluish tinge, the trouble is not in the electrical system.

On the other hand, if the vibrator on the induction coil does not buzz when the engine is turned over, one or more of the following troubles exists: A broken or disconnected wire between the coil and the engine; a short-circuit or a broken wire in the battery or generator circuit; contact points out of adjustment or in bad condition, failing to make good contact; binding posts on the coil broken or loose; the coil is wet or otherwise damaged; vibrator spring broken or vibrator stuck; battery weak or exhausted; wiring put back wrong. If the coil seems to work all right but the plug fails to fire, the seat of trouble is either in the high-tension ground wire or in the plug itself. If it be in the plug, it may be due to the parts being loose; the porcelain being broken, carbonized, or short-circuited by dampness.

Irregular explosions in the cylinder are usually caused by a weak battery, but may be due to broken-down insulation on wires, causing a partial short-circuit; a dirty or broken spark plug; wire terminals not being securely fastened; too much lubricating oil, or an inferior quality which carbonizes readily; vibrator points on the coils being out of adjustment or pitted; contacts in the timer being worn; carbureter check valve not working right; carbureter float not moving freely; water in the carbureter; gasolene feed pipe being clogged.

When an engine fires regularly but does not develop its normal power, the cause may be insufficient lubrication of either the bearings or the piston; the crank shaft being out of line; a poor grade of gasolene; wrong proportioning of the mixture: low compression due to stuck or worn rings, wear of the cylinder wall, a loose spark plug or loose valves or valve cages; insufficient lift of the exhaust valve: exhaust port partly clogged; exhaust cam out of place; too much tension on the air valve of the carbureter; inlet-valve spring out of order.

Muffler explosions are due to misfiring, the unburned mixture being delivered into the hot exhaust passages where it is ignited; misfiring may be caused by the exhaust valve being hung up or the stem being bent and preventing the valve from seating properly; by the mixture being too weak, or by an insufficient or retarded spark.

Readers with Something to Say

Dimensioning Pipe Bends

When the average engineer desires to get some pipe bends made, he naturally endeavors to make a sketch of what he requires. Many times, especially when the work requires anything but a 90-degree bend, the measurements given on the sketch submitted to the pipefitters are not definite enough. It is absolutely necessary that the angle between the straight runs of pipe be given in some manner, and that the ends of the pipe be definitely located.

It is quite common to see a sketch like Fig. 1 with the dimensions *A* and *B* given only. Now, if it is tried to make a full-sized drawing on the floor from these two dimensions, it will soon be found that the bend may occur at several different points. By the addition of dimension *C* from one end of the pipe to the intersection of the center lines projected, the bend is definitely located and the direction of the pipe determined.

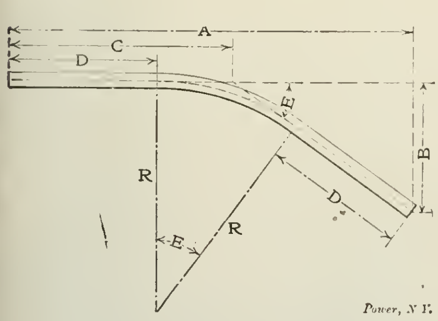


Fig. 1

It is often thought that if *A*, *B* and the radii *R* are given, these are all the dimensions necessary, but this is not so. Dimensions *C* or *D* will complete the requirements. Unless all the dimensions are taken with care, there is liable to be some difficulty in making a sketch to satisfy all these conditions. Dimensions *A*, *B* and *C* are the best to work from, with the radius *R* if the job is particular. If the angle *E* can be measured with a protractor, then the dimensions from the intersection of the straight runs to the ends of the pipe on both legs are the best ones to give.

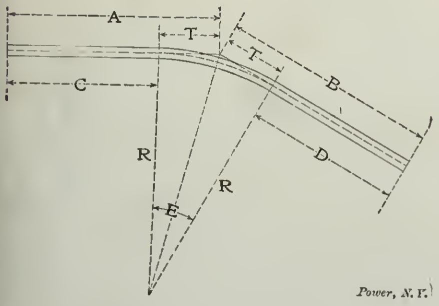


Fig. 2

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

The pipefitter must know at what point to start to bend the pipe, on what radius to bend it, how much pipe to bend, and the angle that the straight ends make with each other. His final check must be the angle and the length from the intersection of the center lines to the end of the pipe on each leg. This dimension is called the center-to-end length.

Suppose, as shown in Fig. 2, there are given the center-to-end distances *A* and *B*, the angle *E* and the radius *R*. In order to get the length of the straight ends *C* and *D*, *T* must be calculated and subtracted from *A* and *B*. Connecting the center point with the center of the arc of the bend will give two equal right-angled triangles.

The ratio found by dividing the perpendicular by the base of a right-angled triangle is called the tangent of the angle opposite the perpendicular. Most handbooks of engineering have tables which show this ratio corresponding to different angles. The angle to be considered is one-half of the angle *E*. Looking in a table of tangents the decimal number corresponding may be found, and as before stated this will be equal to *T* divided by *R*; or in other words, *T* is equal to *R* times the decimal found in the table. Subtracting *T* from *A* and *B* will give the straight lengths *C* and *D*.

The length of pipe to be bent must now be determined. It is known that the diameter of a circle times 3.1416 gives the circumference, and since the radius is one-half the diameter, the radius times 6.2832 equals the circumference, or the length of an arc of 360 degrees. Therefore the radius times 6.2832 divided by 360, or 0.01745, gives the length of an arc of one degree and unit radius. To get the length of any arc of a given angle and radius, multiply the radius by 0.01745 and by the number of degrees in the angle. By adding together dimensions *C* and *D* and the length of the arc, or bent part of the pipe, the amount of pipe to be cut will be obtained.

Willis M. Wright.

New Haven, Conn.

Sand Erosion

Some years ago, on investigating a case of so called electrolysis, I found a condition of affairs as shown in the accompanying photograph, and traced the cause to sand erosion or sand blast, due to the rapid flow of water and sand against the pipe.

Since then, I have had occasion to notice this same occurrence in a number of instances on water-works piping, and also to a more limited extent when any material leak occurred on steam piping laid in sandy soil. The results were far more rapid when water was present than when only steam escaped.

Effects of Sand Erosion

My conclusions are that it is not wise to lay steam pipes in the earth where a leaky connection is possible, and that the very best of workmanship is necessary for the installation of both steam and water pipes where laid in sand or gritty earth.

Henry D. Jackson.

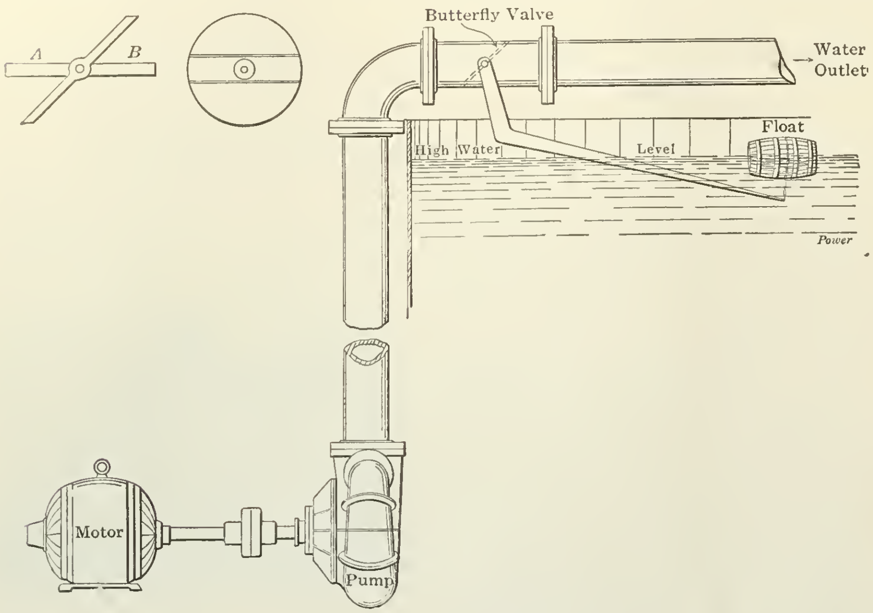
Boston, Mass.

Water Supply Control

Frequently in power houses where a continuous water supply is needed, the rate of consumption varies considerably. With such conditions there is a risk of having at times an empty or an overflowing tank, or else it is necessary to have a man in the pump house to watch the water level.

If conditions will allow a centrifugal pump being used, the scheme shown in the accompanying illustration will solve the difficulty. The float is made of a common oak keg painted to prevent cracking, and the butterfly valve is one that any machinist can turn up on a lathe.

The method of turning a butterfly valve may not be familiar to all, hence a short description may be of service. The casting or plate must have false stubs as *A* and *B*, which, when centered and placed on the lathe, allow it to be turned to the required diameter. These ends are then chipped off. The valve should be a fairly tight fit but not absolutely water tight as a small leakage will not matter. The valve may be fastened to the shaft by set screws.



Float Attached to Discharge Pipe

A butterfly valve has as much pressure above the center as below, hence it may be termed a balanced valve. Thus no undue pressure is thrown on the lever connecting to the float. This is true in whatever position it is placed.

Supposing the supply pipe to the tank, or, in other words, the discharge pipe from the centrifugal pump, to extend to an indefinite hight, the pump would pump as high as its capacity would allow and there the water would remain. This is practically the same as when a valve is placed at the end of a short discharge pipe, and no harm will result.

The operation of the scheme is obvious and I hope some brother engineer may profit by it.

Frank S. Bunker.

Vallejo, Cal.

An Emergency Decision

Some years ago I took a position with a new steel mill which was under construction and began by fitting up the first battery of five vertical water-tube boilers. Before it was possible to get a feed pump in commission I was called upon for steam with which to start the blowers on a battery of gas producers. There were a couple of new locomotives at hand, which had not been put into service, so an injector was taken from one of them and connected to the feed pipe at the rear of one boiler.

When the boiler was filled and fired I noted at what pressure the injector would work, finding that it refused to handle any water until the steam had reached 110 pounds. The boiler was rated at 300 to 500 horsepower and the size of the injector was only 1¼ inches. But as the full rating of the boiler was not required, it was possible, by leaving the injector on all the time, to keep sufficient water in the boiler. A night and a day fireman were put on and for several days things went all right. The gas producers were put into service and the open-hearth furnace started; they had reached the point where to cut off the gas was out of the question.

As the feeding apparatus for the other boilers could not be put into commission for some time, I began to get anxious, and cautioned the firemen to maintain the pressure so as to keep the injector going at all hazards. I lived about a mile from the plant, and was awakened a few nights after this by hearing the low-water alarm at the plant. Getting into a few clothes, I started for the plant at a full run, meeting the fireman, who was on his way to notify me. Arriving at the plant I found the injector stopped with the pressure gage on the boiler showing 80 pounds. Both furnace doors were wide open, but were choked with live coal and clinkers, which the fireman had attempted to withdraw. There was not a drop of water in the gage glass. I thought of the damage that would result from stopping the blowers and determined to raise the steam pressure in order to get the injector into operation. I accordingly pushed the fire back onto the grates, opened the dampers and began nursing the fire. In a few minutes it was white hot and the steam pressure began to rise, though at the time it seemed an age. In the meantime, I cooled the injector and when there was sufficient pressure it fairly shrieked as it caught the water. The boiler was kept going for a couple of weeks before another was ready to relieve it, but a thorough examination failed to locate a single injury. It is not known how low the water was in the boiler, but the cause of the trouble was undoubtedly due to the fireman having gone to sleep. He disappeared just after notifying me of the trouble and I have never seen him since.

T. J. Bloss.

Chattanooga, Tenn.

Improving a Brake

The brake on our hoisting engine, located at the shaft of a mine, had always given more or less trouble. On the down trip it took strenuous work on the part of the engineer to set the brakes tight and quick enough to keep the cage under control.

Fig. 1 shows the kind of brake used.

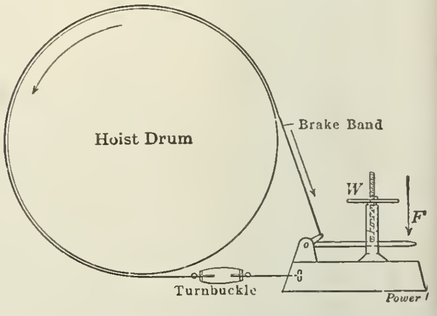


Fig. 1. Original Band Connection

It was of the common band type, lined with hardwood blocks and tightened by means of either the handwheel *W* or the foot-lever *F*. The arrow denotes the direction of rotation of the drum when the cage was being lowered, and it will be seen that the pull on the brake strap in tightening was in a direction opposite to that of the drum. This made it very difficult to set the brake.

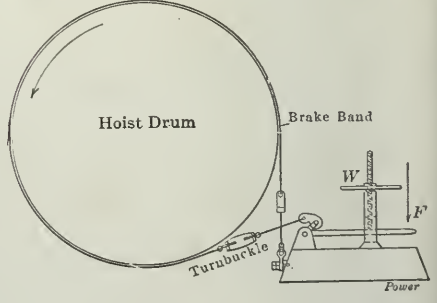


Fig. 2. Band Connections Reversed

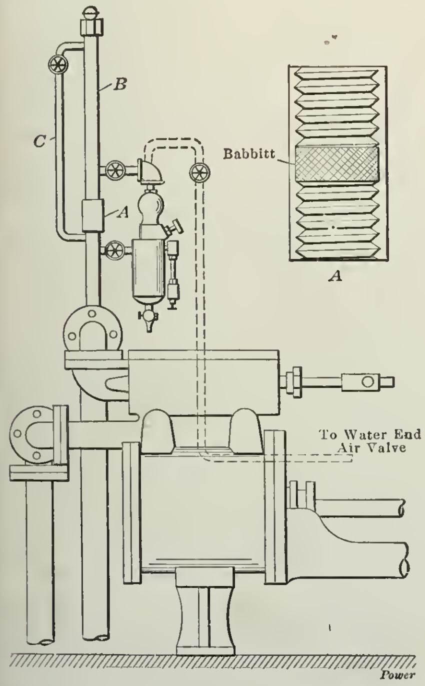
It was decided to overcome this trouble by reversing the connections of the brake strap, so that the pull in setting the brake would be in the same direction as the rotation of the drum. This was done as shown in Fig. 2, and proved so satisfactory that a slight pressure of the foot on the lever *F* was as effectrve as screwing down on the handwheel formerly was.

J. A. Carruthers.

Bankhead, Canada.

Piping a Lubricator

I was once asked to help doctor a boiler feed-pump lubricator, located on a standpipe rising from a low steam pipe, as shown in the sketch. It appeared that when the condensed-steam pipe was connected in the usual manner, the lubricator could not be made to supply the desired amount of oil. At the time I visited the plant, this pipe was connected to the water end of the pump, as shown by the dotted lines. Naturally the excessive water pressure soon emptied the lubricator, no matter how little the regulating valve was opened. The remedy applied was that of cutting a 12-inch length from the 1-inch standpipe, and inserting the galvanized coupling *A*. The middle portion of the coupling was filled with melted babbitt, which was made to adhere firmly to the sides. Then the remaining piece *B*, 18 inches in length, was screwed in place. The two chambers formed by the babbitted coupling were connected together by the 1/8-inch bypass *C*, and the lubricator piped as shown, the top connection being made above the coupling as near to it as practicable and the other connected below it. The water of condensation in pipe *B* is at all times sufficient to maintain a uniform flow of oil to the feed pumps, and the lubricator has given entire satisfaction since making the change.



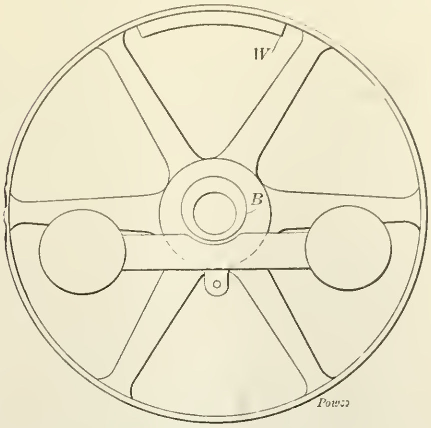
Original and Improved Arrangement of Piping

H. P. Everette.

Boston, Mass.

Flywheel Out of Balance

The engine was of the high-speed type and caused considerable annoyance by excessive vibration. At slow speeds it would run smoothly but upon speeding up to its normal speed the flywheel began to run out of true and the shaft began to spring. We placed a scribe to the hub of the flywheel and it traced the path *B*, shown in the sketch. This led us to believe that the flywheel was out of balance. We accordingly removed part of the weight *W* and experienced no more trouble.



Flywheel Showing Mark Made by Scribe

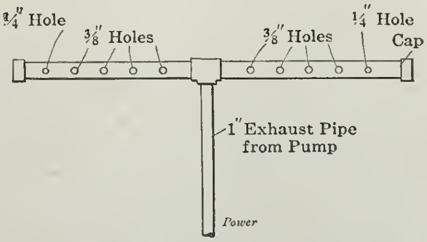
C. R. McGahey.

Sheffield, Ala.

Preventing the Formation of Clinkers

Upon first coming to the plant at which I am at present employed, I was troubled with large clinkers forming on the grates, especially next to the bridgewall and along the sides of the furnace. I was about to run a jet of live steam under the grates when the thought came to me of using the exhaust from the pump, instead of allowing it to escape through the roof.

I accordingly drilled a hole through the front and center of the furnace, close up under the grates, also drilling several holes in two pieces of gas pipe long enough to reach underneath the grates. To give best results this pipe should be at least ¼ inch larger than the exhaust from the pump and the holes next to the ends should be smaller than the others, as shown by the accompanying sketch.



Pipe Placed under Grate

Since installing this I have not been troubled with large clinkers and there is condensation enough in the steam on its way to the discharge pipe to dampen the ashes, so that when the ashes are pulled out there is no dust flying about.

Horace B. West.

Geneva, O.

Erecting a Boiler

Some months ago our company purchased a 172-horsepower Babcock & Wilcox boiler located in an abandoned mill about three-quarters of a mile from our power house. The work of removing and reërecting this boiler devolved upon the writer, and a description of how it was accomplished may help someone having a similar task to perform.

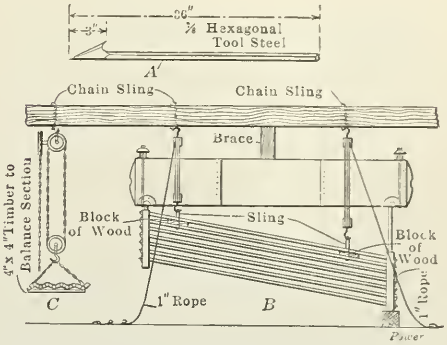


Fig. 1. Method of Removing Sections

With six negro laborers, a wagon, chainfalls, ropes, hammers and other necessary tools, I proceeded to the mill. After starting several men to work tearing out the brickwork, cautioning them to save as many whole bricks as possible, for use again in the new setting, with the others I commenced taking down the safety valves, steam pipe and doors, loading them onto the wagon as fast as they were removed to be hauled to the power house.

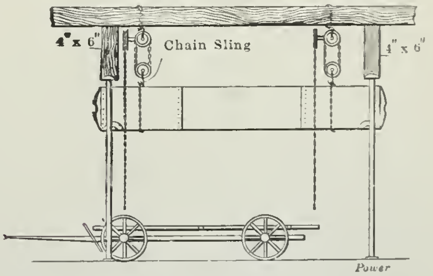


Fig. 2. Lowering Steam Drum

This did away with frequent handling and at the same time kept the space around the boiler clear, allowing the men to work to better advantage.

At the power house the bricks were dumped on the ground, and a negro boy, at 75 cents per 1000, cleaned and stacked them. The iron work was brushed off with a steel brush and given a coat of graphite paint.

When the brickwork was torn down to within 2 feet of the foundation, we braced the boiler from both sides with 2x4-inch scantlings. After all the brickwork had been removed, the nipples holding the mud drum were cut out, the mud drum was removed and blocking placed under the rear end of the headers, to steady them while cutting out the vertical risers, to allow for removing the sections.

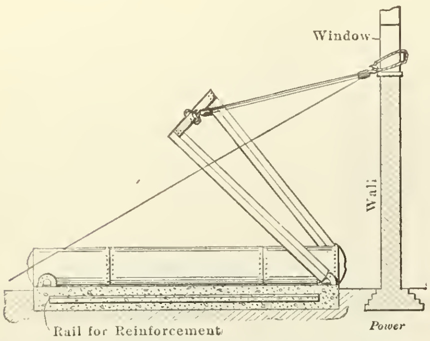


Fig. 3. Raising the Columns

At *A*, Fig. 1, is shown the form of tool used in splitting the nipples, care being taken not to cut the header or drum. At *B* is shown the manner in which the sections were removed, one section at a time; the slings being slipped along the tubes until the latter could be swung out to where the chainblock could be attached. Then the section was turned halfway around, as at *C*, ready to be lowered into the wagon.

Fig. 2 shows how the steam drums were lowered onto the wagon after all the sections had been removed.

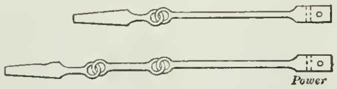


Fig. 4. Special Expanding Tools

The drums were hauled to within a few feet of the foundation which previously had been put in and were set in position ready to be hoisted when the supporting columns were up. Fig. 3 shows how these columns and girders were raised. In setting the columns care was exercised to have them perfectly level and in line with one another.

The drums were hoisted into place in a manner similar to that in which they were taken down, and the cross pipes were bolted on to help hold them in place.

The tube sections were now raised and held by chainfalls while the new nipples were expanded into place. In expanding these nipples they were given an extra flare to insure against pulling out. Fig. 4 shows the special form of expanding pins used to get up through the handholes into the cast-iron headers.

As each section of tubes was placed in position the baffle bricks were fastened in place with a stout piece of string, the cast-iron pieces, which hold them from slipping, being put in and bolted together after all the sections were in place.

Next, the mud drum was blocked in place and the nipples expanded into it; after which the front was put on, the boiler closed and subjected to a hydrostatic pressure of 240 pounds per square inch, the boiler-feed pump being used for this purpose. With the exception of a few leaks around the handhole plates, the boiler withstood the test all right. After the setting was completed we kept a slow fire under it for several days to dry out the brickwork.

W. F. Cox.

Natchez, Miss.

Clogged Vent Pipe

An uptodate plant had been installed in a new office building, the dedication of which was marked by a large banquet in the evening. Consequently all the lights and fans were in use, which threw a heavy load on the power plant. The equipment included five Westinghouse engines of various sizes, four of which were in service on this particular occasion.

While the banquet was in progress and everything was apparently running smoothly in the engine room, there suddenly arose the smell of burning oil. Immediate investigation showed that the crank pins of two of the engines were running hot, and a speedy change was made to the spare engine. This had barely been accomplished when the other two engines began to smoke at the cranks. For a while it looked as if a shutdown were inevitable, for every time an engine was started the crank pins began to heat. Water was poured into the crank cases by the bucketful, but would not remain there.

In this type of engine the crank case has a vapor vent which must be kept open, or the accumulation of vapor will create sufficient pressure to force the water out through the overflow. Upon the water in the lower part of the case floats the oil into which the cranks splash at each revolution. In this case the vents from the crank cases of each of the five engines connected with the main vent pipe which led to the flue. Here was the cause of all the trouble—the opening into the flue was found to be stopped up. This caused excessive pressure in each crank case and forced the water out, thus placing the level of the oil out of reach of the crank.

Edward T. Binns.

Philadelphia, Penn.

Clogged Suction Pipe

On taking charge of a 400-horsepower plant some years ago I was confronted with a complicated system of piping, the details of which it is not necessary to go into. The receiver tank rested horizontally on the floor. The two feed pumps rested on a foundation raised 8 inches above the floor beside the tank. The 2½-inch suction pipe was tapped into the bottom of the tank near one end, dropping about 3 inches to an elbow, and running 24 inches beyond the end of the tank to another elbow, finally terminating in a tee between the pumps. Just below the tank and past the ell was located a 2½-inch globe valve. The tank had not been opened for cleaning for over a year and the only opening to the atmosphere was through a ¾-inch vent pipe and a 1½-inch pipe with a stop cock was used for putting soda into the feed water.

One day the pump began to kick and the water began to lower in the boilers. The gage glass on the tank showed the water level to be near the top. The second pump was started but acted the same as the first. I concluded that the trouble was in the valve, and managing to supply the boilers with cold water from the small pump in the engine room, I took off the bonnet of the globe valve and allowed the water to drain from the tank. I saw at a glance where the trouble was and taking a packing hook picked out a handful of waste from the globe valve and pipe. How it came in the valve is still a mystery to me, as it was good strong waste showing very little sign of decay. It evidently had been put into the tank through the 1½-inch pipe.

R. A. Cultra.

Boston, Mass.

Frost in the Engine Room

I am troubled a great deal during the winter months with frost accumulating upon the walls and ceiling of the engine room. The building is a frame structure, the outer wall of which consists of one ply of boards, one ply of sheathing and a covering of galvanized iron; the inside is sealed with ordinary lumber. The frost collects on the ceiling and is apt to fall on the generators and engine and cause trouble.

We have arrived at the supposition that the strong draft under the boilers displaces the air in the engine and boiler rooms, causing a suction of cold air through all the crevices in the walls.

We thought it might be possible to overcome the trouble by installing a cold-air duct from the outside to the ash pit. Opinions as to the advisability of this will be welcomed.

Charles Fenwick.

Saskaskia, Can.

Questions Before the House

Where Was the Westinghouse Governor?

In the June 14 issue Mr. Westerfield mentions the case of a runaway engine which was equipped with a Westinghouse governor. I should like to ask Mr. Westerfield where this plant was located.

Terrell, Texas. W. C. Drake.

Installing Globe Valves

In the June 21 issue, under the above heading, I mentioned an experience with globe valves on the bottom blowoff pipe of two boilers. The valves were of the angle type, with the pressure on top of the disk. The stem broke on one valve, leaving it closed, and on the other valve the thread on the stem became stripped and left this also closed. This plant was not located in Massachusetts.

Exceptions were taken to this part of my contribution by Mr. Smith in the July 26 issue. The threads on all valve stems wear out, but on none more quickly than where the pressure is on the top or back of the disk. In the case in question, I secured a 6-foot crowbar and a piece of board and put the block on the wheel. I then drove the end of the bar into the brickwork and pressed the valve shut in less than two minutes in one case, fastening the free end of the bar with a rope and weight until the valve could be repaired. In another case I happened to have some long ⅜-inch bolts. Putting the heads in a vise, I bent hooks on them to go around the valves. Slipping the points through the wheel on the stem and placing washers on each bolt, I screwed up the nuts and had the valve clamped shut in a short time. But I have not seen the engineer who could open one of these valves which had the pressure on the top of the disk when the disk broke from the stem. I have used eight different types of stop cocks and fifteen different types of valves under both high and low pressures and have not found any which were perfectly satisfactory on a blowoff pipe under all conditions. A valve that is suitable for use under 200 pounds pressure with superheated steam is too small and too heavy for a pressure of 65 to 85 pounds. I have used the babbitt-faced disk mentioned by Mr. Smith and in two instances came to grief.

I wish to call Mr. Smith's attention to the acts of 1909 relative to the operation and inspection of steam boilers in Massachusetts. The section dealing with blowoff pipe and fittings states "Globe valves shall not be used." Referring to feed piping, it states that "When a globe valve is used on a feed pipe, the inlet shall be under the disk of the valve." These laws apply to boilers installed after May 1, 1908.

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Perhaps Mr. Smith uses an iron-body blowoff valve having a heavy piston or disk about 2½ inches long, with a babbitt face at the bottom. With the long disk practically closing the port, and the babbitt face pressed against the seat, the valve would be practically steam and water tight. The only pressure at the seat, other than that caused by the screw in closing the valve, is from leakage past the disk. Therefore, the full boiler pressure does not come on the valve seat, but strikes the disk at right angles, leaving no more boiler pressure on the top of the disk than there would be on the bottom. Hence the valve can be opened very easily. This form of blowoff valve works very satisfactorily under pressures of 100 pounds or less, but cannot be classed as a globe valve on account of the shape of the disk.

R. A. Cultra.

Boston, Mass.

I do not advocate valves set with the pressure on top of the disk. A short time ago I was called upon to open a 10-inch valve having the pressure above the disk, and even with the help of a 24-inch wrench applied to the wheel, I could not budge the valve. Of course, I could have used a little more force, but probably would have pulled the stem out of the disk.

The angle valves on the blowoffs of one of our boilers are also set in this way, and they are continually leaking at the glands, notwithstanding the fact that they are packed every year.

W. H. Plowman.

Philadelphia, Penn.

Long Run of Pumping Machinery

Answering the request of Edwin D. Drefus in a recent issue of Power, I would say that we have a 21 and 42x24.5 x36-inch compound-condensing Gaskell crank and flywheel pumping engine, rated at 7,500,000 gallons per twenty-four hours. This engine ran from June 1, 1901, to July 10, 1902, a total of thirteen months and ten days without a stop. This long run was necessary because all the other equipment was being removed, and a new 10,000,000-gallon pump was being installed. The Gaskell pumping engine was installed in 1873, and had been in service nineteen years when this record was made. It was thoroughly overhauled before starting in 1901 and never gave any trouble during this long endurance test; in fact, when it was shut down it was found to be in perfect working order. During this period an average of 5,000,000 gallons was pumped every twenty-four hours. The Holley system is used at this plant, and the pressures are 60 pounds for domestic purposes and 100 pounds on the fire lines.

John Ackerman.

Auburn, N. Y.

Rim Bolts

In the June 5 issue, Mr. Mulham inquires whether the rim bolts in a wooden flywheel are in tension or in shear. A flywheel such as described and traveling at its customary speed would have, due to centrifugal action, a force tending to pull out the arms and the rim. If the rim were perfectly rigid there would be no forces acting on the bolts at all. But as the rim is actually more flexible than the arms, centrifugal force will cause the rim to assume an elliptical form due to its being held at two opposite points and a tension will be brought upon the rim bolts by the tendency of the rim to assume a circular form.

John French.

Washington, D. C.

In reply to Mr. Mulham's letter in the issue of July 19, I offer the following: Both, the men who say the bolts will be in shear, and those who say they will be in tension, are right. The force acting in a direction along the radius of the wheel will equal $\frac{MV^2}{R}$ where M is the mass, i.e., the weight divided by 32.16, the acceleration due to gravity, V is the velocity of the rim, and R the radius of the rim. This is the force which places the bolts in tension and is present while the wheel is revolving. The shear upon the bolts varies directly with the acceleration of the wheel if running without a belt and if running with a belt there is the additional tension of the tight side of the belt. Hence, when the wheel is getting up to speed, even if there is no belt, there will be a tension on the bolts varying as the square of the speed and there will be a shear upon the bolts which will remain nearly constant if the increase in velocity of the wheel is constant. When the wheel is running at constant speed with no belt the shear will be negligible except for a small amount due to the windage of the rim. With a belt, the shear will depend upon the tightness of the belt and the load placed upon it. The belt will hold the rim against the arms and the only stress upon the bolts under the belt will be the shear.

P. Arthur Tanner.

St. Louis, Mo.

The question as to the nature of the stress imposed upon the bolts holding the rim of a built-up flywheel to the arms, has been raised by Mr. Mulham in the July 19 issue. If the nuts were drawn up very tight on the bolts, a tensile stress would be created. If the arms were pocketed a shearing stress could not under any circumstances exist; a shearing stress is possible only where the arms are not pocketed and when the bolts are not strong enough to make the adhesion between the contact surfaces great enough to resist the tendency of the rim to slip. Possibly the fact that the belt pulls on the rim may have had some influence in creating the idea of a shearing stress, but in a properly designed wheel the strength of the various parts is such that the effect of the belt pull is negligible. Inasmuch as there is no shearing stress imposed on the bolts in a properly designed flywheel rim having pockets or recesses for the arms to fit into, there is but one type of stress which the bolts must resist, namely, tensional. Naturally, the centrifugal force of the rotating rim increases this stress on the bolts.

Wesley McArdell.

Brooklyn, N. Y.

Induced Draft Trouble

Referring to Mr. Faulthier's letter in the July 12 issue, concerning trouble with an induced-draft system, I believe that his fan has a greater capacity than the smokestack, consequently causing a reaction in the current. If he were to reduce the speed of the engine he would obtain better results under the circumstances, as there is a limit to all things.

J. S. Turner.

Big Rapids, Mich.

Replying to Mr. Faulthier's letter in the July 12 issue on "Induced Draft Trouble," I would say that, in my opinion, the trouble could be remedied by putting an elbow on the discharge pipe so as to discharge upward in the smokestack; this would act somewhat as an ejector. The probable cause for the blower not working under present conditions is that it discharges downward and the smoke just rolls around in the base of the stack, the blast from the blower being strong enough to act as a wall and prevent the smoke from passing up the stack. It should be remembered that a blower will not keep discharging into an air-tight receptacle.

Frank Gartmann.

Sheboygan, Wis.

Runaway Engines

In the July 19 issue, Mr. Trube cites the case of a runaway engine and tells how it was brought under control again by closing a stop valve in the steam line. His remarks show that the engine was not properly adjusted or it would not have run away at all under the conditions.

An engine should be run with the throttle valve wide open all the time. It is the duty of the governor to regulate the speed of the engine, regardless of the variations in pressure or load, and the throttle valve should never be used for this purpose.

Mr. Trube states that when the steam pressure was low the throttle valve had to be opened wide in order to keep the engine running up to speed, and as the steam pressure was raised the speed increased so that the throttle valve had to be closed somewhat; but as the throttle valve refused to work, the speed continued to increase until the engineer closed the stop valve, the regulation for the balance of the week being done by the stop valve. The engine was a cross-compound Corliss. I wonder where the governor was, or if it was regarded as only an ornament and had been discarded? It is quite plain that the engine was not correctly adjusted, the governor apparently being of no use whatever.

From the old type of plain slide-valve engine, governed by a throttling governor, to any of the modern types of engines, all should be operated with the throttle valve wide open. If an old slide-valve engine does run away, because of the governor belt having broken or jumped off, the quickest way to stop the engine is to grasp the governor balls and raise them, thus instantly closing the governor valve; this can be done quicker than any throttle valve can be closed.

In a single-eccentric Corliss valve gear the latest point of cutoff is 45 per cent. of the stroke and the earliest is zero, with all variations in the cutoff, from zero to maximum, under the full control of the governor. When the governor fails to work, the safety cams are brought into service, preventing the steam valves from being opened, and the engine stops. It is a very unusual defect that can cause a properly adjusted Corliss engine to run away.

Had Mr. Trube's engine been correctly adjusted, the governor would have taken care of the speed during the variation of pressure, and no runaway would have occurred.

H. W. Benton.

Cleveland, O.

The letter appearing in the July 19 issue entitled "Runaway Engines," is, no doubt, a step in the right direction. The recommendations, however, are hardly adequate and by the expenditure of a few more dollars a complete installation can be made whereby the safety of both employee and equipment is made independent of the presence of mind of the engineer.

In the first place, not every man will rush forward and shut down an engine when it is running at a speed which is liable to burst the flywheel at any instant. Moreover, the chances are that, by the time the engineer reaches the stop valve the damage will have been done, especially if the increase in the speed of the engine has not been noticed immediately. It is obvious, therefore, that a mechanism for tripping the valve that will operate automatically will prove the best safeguard against accidents of this kind. Such a mechanism should not be dependent upon the regular engine governor for its operation; any device of this kind, if attached to the governor, would be of no greater value than the governor itself, as has been proved time and again.

An arrangement that has been found quite satisfactory is that of having an additional governor attached to the engine shaft by means of either a belt, chain or gears; although the latter method is not always advisable. The governor should be arranged so that it will trip a quick-closing valve when the engine reaches the upper speed limit. A common method of operating this system is by means of an electric contact connecting with a solenoid, which closes the valve. The necessary current can be obtained from batteries or the generator, if the engine be connected to one. A tripping lever should also be provided to shut down the engine in case the belt or chain breaks. Practical demonstrations of installations of this kind have shown that it is possible to have the steam entirely shut off within four seconds after the engine reaches the speed limit.

Such a system is not very elaborate, and there are few engineers who will not agree that it is a good investment.

Everard Brown.

Pittsburg, Penn.

Weakness as Disclosed by Boiler Explosions

In several descriptions of boiler explosions that have appeared in recent issues of Power, a point has been made of the fact that the metal had torn in the solid plate rather than at the joint or having sheared the rivets. This fact was stated to indicate that the plate had been damaged or deteriorated to such an extent that it was weaker at the torn section than at the joints. It is a well known fact to those who have made a study of boiler explosions, that aside from the point of initial rupture, the breaks in a boiler structure, due to an explosion, do not follow the lines of least resistance, by any means. Frequently a line of rupture will take a girthwise course, although a cylindrical vessel is twice as capable of resisting rupture in this direction than it is longitudinally; this is without considering the strengthening effect of the tubes, which is an important factor always present in the horizontal tubular boiler.

The erratic nature of ruptures produced by boiler explosions is conclusive proof of the theory advanced by Zerah Colburn regarding the manner in which such explosions are produced.

This theory is in effect, that the initial rupture, (which, of course, occurs in a section that is least capable of resistance) produces a water-hammer effect in the boiler due to the sudden release of pressure in the steam space, and subsequent swelling of the contained water due to the instantaneous formation of steam bubbles throughout its mass. As is well known, ruptures due to water-hammer do not follow the lines of least resistance, but follow the direction of application of the force. Thus in a pipe line, a water-hammer usually blows out a cap at the end of the pipe or destroys a fitting; most of the failures being in a section that would have been the last to have failed under a steady application of internal pressure, and the same fact is true of boiler explosions. Therefore the nature or direction of ruptures produced by boiler explosions cannot serve to determine the probable relative strength of the different parts.

S. F. JETER.

New Haven, Conn.

Central Station versus Isolated Plants

It is evident from Mr. Marks' letter in the July 12 issue that he does not understand the theory of rate making for electric service. He considers it wrong that the small consumers should have to pay three or four cents per kilowatt-hour more than the average price, while the large consumers, who can afford to operate their own plants and consume but a small proportion of the total output, enjoy the special privilege of paying three or four cents less than the average rate. Mr. Marks errs in assuming that the price per kilowatt-hour should be made the same to all. As this misapprehension is shared by many and hinders a proper settlement of the rate question, the following may be timely:

It is recognized by the electrical interests that it is radically wrong to fix a uniform rate per kilowatt-hour. That which the central stations sell and which the consumers buy is service and not merely a certain number of units of electricity. This service consists of several factors, which vary greatly in their relative proportions; often the cost of energy consumed is small in comparison with some of the other factors. To illustrate this, suppose that a plant is to be installed to supply one hundred consumers with light. To determine the capacity of the plant, the average maximum demand of each consumer must be known. Suppose this to be five kilowatts per hour; then a plant of 500-kilowatt capacity must be installed. Some consumers will not require more than one kilowatt, while others may require 20 kilowatts. It is evident that the investment necessary to care for one may be 20 times that required for another. As soon as the plant is installed, the investment becomes subject to interest, taxes, insurance, depreciation, etc., which are practically constant regardless of the energy used by the consumers, and it is plain that each consumer should pay such a proportion of these charges as the investment made to take care of his needs bears to the total investment; he should pay this regardless of the kilowatt-hours used. This may be charged to "readiness to serve."

After the plant is in operation, there is a certain amount of work that must be done for each consumer, such as reading and caring for meters, keeping the records of his equipment, etc., and making out and collecting bills. All of these are practically independent of the energy consumed and may be termed a bookkeeping charge.

Finally, there is the cost of generating the power. This varies greatly, depending on the efficiency of the equipment and lines, the cost of fuel, labor, etc., but will be the same per kilowatt-hour for each consumer. If a consumer who has many lamps installed uses them but a short time each day, the amount of energy he uses may be no more than that of a consumer who has only a few lamps, but who uses them a much longer time. Suppose the "readiness to serve" charge amounts to $2.50 per month per kilowatt of installation, the bookkeeping charge is $1 per month and the cost of generating power is two cents per kilowatt-hour. If one consumer has lamps amounting to 10 kilowatts and consumes 500 kilowatt-hours per month, he should pay 10 × $2.50 = $25 per month for the "readiness to serve." He should pay $1 for bookkeeping charge and two cents per kilowatt-hour; a total of $36 per month, which is an average of 7.2 cents per kilowatt-hour. Another consumer having lamps amounting to only two kilowatts may use the same amount of energy, in which case he should pay $5 per month for "readiness to serve" charge; $1 bookkeeping charge; and $10 for energy; a total of $16, which is an average of 3.2 cents per kilowatt-hour. The average price paid by both is 5.2 cents per kilowatt-hour. Will Mr. Marks insist that it is just to charge both of the consumers at the same rate?

The public in general refuses to give the matter the study necessary to enable them to grasp the principles underlying the making of rates for electric service; hence much of the clamor about unjust discrimination.

I had charge of two plants where an effort was made to introduce this system of charging, but found it very difficult. One month each consumer, who would be benefited by the change, was billed on the "readiness to serve" schedule instead of the regular rate, and one business man came into the office and made a big stir about his bill. I tried to explain the schedule to him, but he would not listen and left, saying, "Young man, this is the last time you will ever get a dollar for keeping books for me." Thereafter he paid at the regular schedule, several dollars more per month than the "readiness to serve" plan would have cost him.

When a central-station management approaches an isolated plant, the cost of serving the consumer having the isolated plant is usually carefully figured and a rate offered that will place the central station on the safe side.

Does Mr. Marks suppose that the central station hustles to get a large consumer, knowing that it will have to hustle for many small consumers in order to make up for what is lost on the large one?

A business concern hustles for the business that pays, and usually avoids the unprofitable customer. It costs more per kilowatt-hour to serve small consumers than large ones; hence they must be charged more. I admit that there is room for a readjustment of rates, but do not believe it should be along the line of a uniform rate to all.

G. E. MILES.

Salida, Cal.

Boiler Flanging

A prominent manufacturer of fire-tube boilers inserts the following clause in his specifications: "Hydraulically flanged heads are not to be used under any circumstances. Heads are to be machine-flanged by the spinning process." The writer does not see why hydraulically flanged or pressed heads are not superior in every way to spun heads. In the first place, the pressed heads of each size are of uniform diameter, making it easy to properly fit the courses to the heads. This uniformity is of consider-are importance where a number of boilers of one size are built together, because, owing to the difficulty of keeping the shell plates and heads separate for each individual boiler in the shop, the person laying out the work in the case of the spun heads of varying diameter, usually endeavors to strike an average for the lot that will best suit all; this usually results in a poor fit for some.

Again, the flange on a pressed head is thicker than the plate from which the head is made; this is a desirable feature and is due to the upsetting of the metal in the flanging process. With the spun head the flange is usually slightly thinner than the plate from which the head is made. The surface on the flange of a pressed head is smoother than that on a spun head, the latter often containing grooves made by the rolls of the spinning machine; this smoothness is important in making tight-head seams without undue caulking. Furthermore, the surface of a pressed flanged head is usually straighter than one which is spun, permitting the use of uniform tube lengths without the necessity of springing the heads into position. The value of this feature after the boiler is constructed is only thoroughly appreciated by the workman who has to put in a new set of tubes, the owner being affected only by the increased cost of the work.

With the multiplicity of diameters called for at the plate mill, the spinning process is by far the most practical for general purposes, but it would seem that a boiler manufacturer who intends to have complete equipment for the manufacture of high-grade boiler work would flange his boiler heads on a press. It seems that the condemnation of the hydraulically flanged head is a subterfuge, intended to mislead the purchaser who is not well informed upon the subject.

F. O. JONES.

Boston, Mass.

Patching Boilers

The proper method of applying patches to boilers is occupying the attention of POWER readers at the present time, the subject being viewed principally from the theorist's standpoint, but with scant consideration for the boilermaker who is to actually do the work. It is proper that safety be given first consideration, and it is aiding this cause to see that the prescribed methods of repair are thoroughly practical.

The majority of patches that are placed on boiler shells are of such moderate dimensions that their shape and the exact spacing of the rivets are not a material factor affecting the strength; in such cases the selection of a method that is most likely to give permanent repair is of paramount importance. I feel perfectly safe in saying that the trouble experienced with patches in 99 cases out of 100 is due to inability to keep them tight; and this lack of tightness is not caused by inability of the patches to resist rupture due to strains caused by the steam pressure. From the standpoint of strength alone, a patch should always be placed on the inside of the shell plate; but there are other important features to be considered. For example, assume that a patch is to be placed on the outer course of a boiler, as illustrated in Fig. 1; it is evident that to facilitate its application such a patch should be placed inside the course, because the ends of the patch can be readily scarfed before it is placed in position. It will then fit smoothly between the shell plates, where they lap over the patch at the girth seam, and a tight joint is assured. If such a patch were placed on the outside of the shell it would be necessary to scarf the edges of the course at the girth seam; this would be difficult and the girth edge of the patch would have to be raised out of level at the corners, which would increase the difficulty of making a tight and lasting job. If the patch were located on an inner course of the boiler, as illustrated in Fig. 2, then it would be necessary to place it on the outside of this course for the same reasons as just given, except that they would apply with greater force, because the scarfing of the corners of the inner sheet, which would become necessary with a patch applied internally, would be a very awkward operation. With a patch of considerable size, where ample room is afforded for working inside the boiler and the extent of the patch lessens the difficulty of having one edge out of level, these points do not apply with such force.

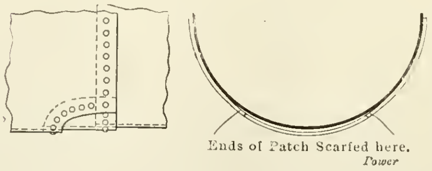


FIG. 1. PATCH APPLIED ON OUTER COURSE

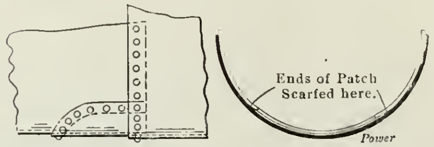


FIG. 2. PATCH APPLIED ON INNER COURSE

Frequently a patch is required on a return-tubular boiler at the point where the blowoff enters. To facilitate draining the boiler it would be desirable to locate such a patch on the outside of the course, but when the patch must include the head seam, as is generally the case, it should be placed inside the shell to permit the making of a tight fit. Usually the deciding factor as to placing a patch on the inside or the outside of a sheet, where it is to comprise a portion of a girth seam, depends upon whether the boiler sheet, to which it is attached, is an inner or an outer course. Where the girth seam is not involved, a patch can usually be put on with equal ease on either side of the sheet.

I do not believe in the generally accepted opinion that the slight depression caused by placing a patch on the outside of a sheet materially increases the liability of scale to collect at such a point. It would seem to be equally consistent to condemn a cylindrical form of heating surface for the same reason and demand a perfectly flat surface, or one convexed on the water side. From the stress that some engineers and inspectors lay on this point it would seem that a whirlpool action, drawing scale from all portions of the boiler, was expected to occur over every slight depression in a boiler plate. The liability of scale to collect on a patch is usually due to causes that produced the initial collection which made the patch necessary.

J. E. TERMAN.

New Haven, Conn.

Unusual Boiler Behavior

In the August 2 issue, Mr. Haeusser describes a boiler experience which he terms unusual. The behavior of the boilers may have been unusual but no boiler could have acted otherwise under the circumstances.

The proper thing for the watchman to have done would have been to fire No. 1 boiler first, because it was the colder of the two boilers. Nevertheless, there was nothing wrong with firing No. 2 first, if he had only shut off the draft on that boiler until the gage showed No. 1 to be steaming well. As it was, No. 1 boiler was being fired at a rate that might be called forcing before the water had reached the boiling point.

When the engines were started the steam pressure dropped because No. 2 boiler could not carry the load alone. Mr. Haeusser pulled the safety-valve chain on the inactive boiler. This lowered the pressure and started action within the boiler.

I have seen boilers behave in the same way. In a battery of boilers some were used all night while the others had the fires cleaned and dampers closed, but the fires were not kept hot enough to maintain the temperature of the water at the boiling point corresponding to the pressure. When the load was thrown on in the morning there was a scramble for steam until all the boilers were hot enough. There was no film of cold air present which might be blamed for the trouble and no safety-valve chains to pull, so we just had to get busy and get the boilers hot.

R. McLAREN.

Berlin, Ont.

The Plotting of Curves*

By F. R. Low

When it is necessary to determine the efficiency of any type of boiler joint it is a great convenience to be able to refer to a diagram which will tell the whole story. Such a diagram is presented in this article and by referring to the familiar expansion curve of the indicator diagram it is shown how these diagrams are plotted.

Mr. Jeter in his lecture on "Riveted Joints," delivered to the club some time ago, described a diagram devised by himself which was in its first form like Fig. 1.

If, for example, the one-inch rivets, for which this particular diagram is drawn, are spaced or "pitched" 2.5 inches apart in an 11/32-inch plate, the horizontal line representing that pitch intersects the vertical line representing that plate thickness at *B*. This point of intersection lying in the lower of the three fields into which the plane is divided by the diagram indicates that the joint may be expected to fail by the tearing apart of the plate as indicated by the sketch in that portion of the field.

If a wider pitch is used there will be fewer rivets to stand the pull, greater stress on each rivet, and more probability that the rivet will crush the plate in front of it, as shown by the sketch in the upper left-hand field. The intersection of the lines representing 3-inch pitch and 11/32-inch plate falls in this field at *A*, indicating that a joint made with 1-inch rivets and these proportions would be expected to fail by crushing. If the plate thickness be increased to 3/8 inch, however, it will not crumple itself nor crush the rivet but will shear it off, as shown by the sketch in the upper right-hand field, in which field the intersection of the lines representing 3-inch pitch and 3/8-inch plate falls at *C*.

If the efficiency of a given joint is wanted it is necessary only to see from the diagram in what manner it may be expected to fail, and compare the force required to produce failure in that way with that required to part the solid plate. With complicated joints which may fail in six or more different ways this diagram, modified to include all possible methods of failure by the simple expedient of limiting the lengths of the lines, as Mr. Jeter explained, becomes of great

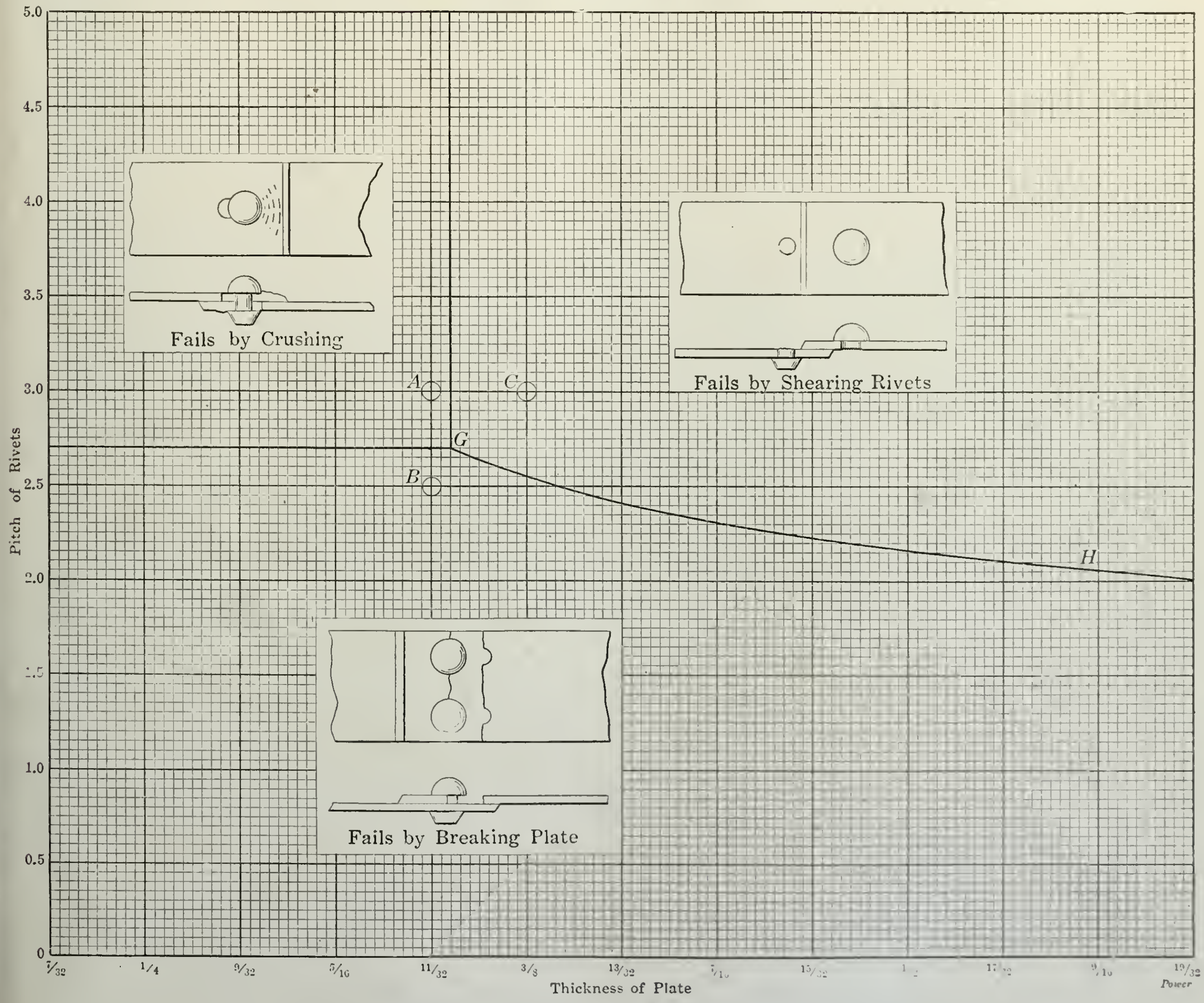


Fig. 1. Diagram Made to Indicate Method of Failure of Joint

assistance and enables one not only to arrive quickly at results which would otherwise require extensive calculation, but to see at a glance the limitations and tendencies of the case.

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*A note to the Modern Science Club, discussing Mr. Jeter's paper on riveted joints.

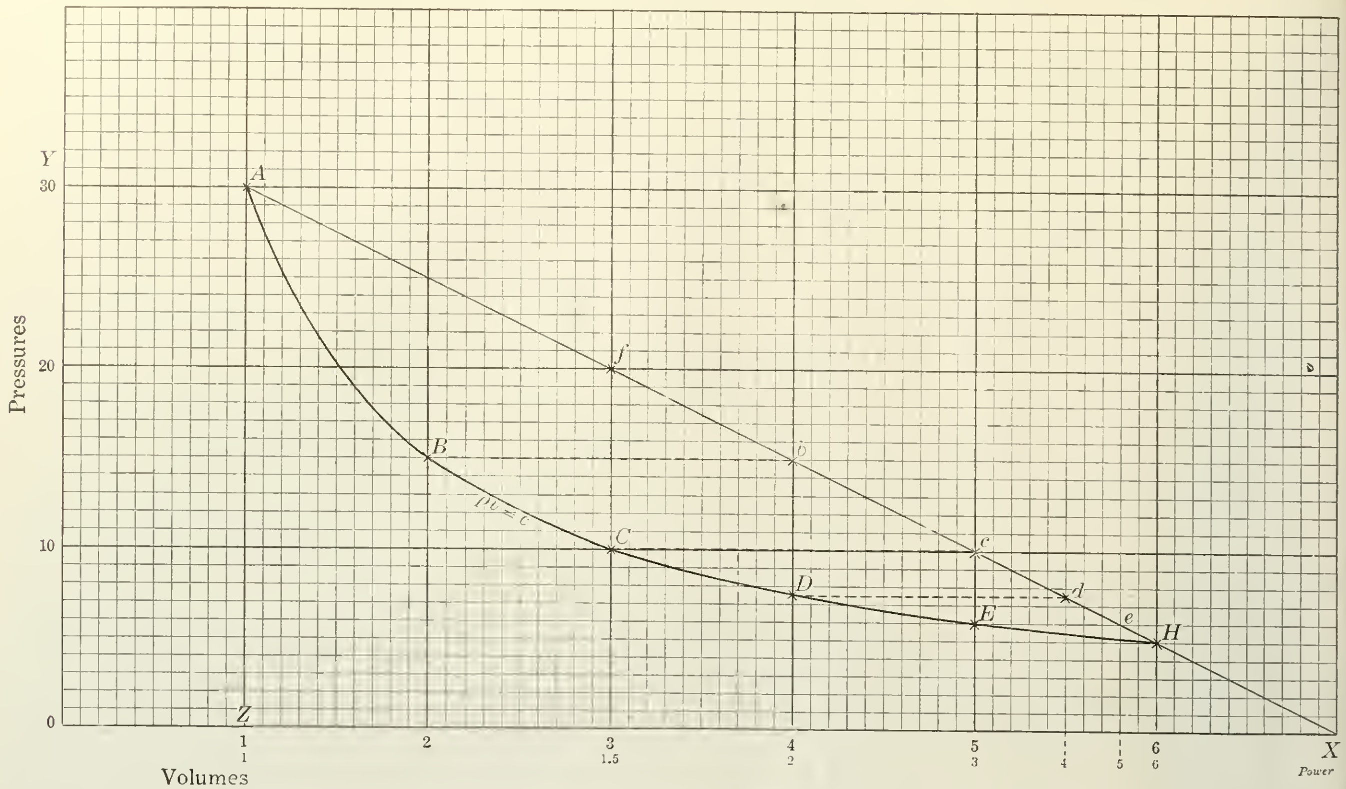

Fig. 2. Ordinary Indicator-diagram Curve and the Straight-line Reciprocal

One of these diagrams has to be plotted for each size of rivet in each type of joint, and for each different assumption that is made of tensile shearing and crushing strengths. This would be a good deal of a task if the line GH separating the plate-failure from the rivet-failure field had to be plotted point by point and traced in the curvilinear form which it has in Fig. 1. Mr. Jeter remarked, however, that by using *the reciprocals* of the horizontal distances in plotting them instead of the distances themselves, the curve became a straight line, and, having the commencement of it at G, it was necessary only to locate one other point upon it and put in the whole line with a straight-edge. Diagrams made in this way for all sizes of rivets and many types of joints with directions for making them were published in Power for January 5, 1909.

It is this particular feature of Mr. Jeter's lecture that I thought it might be interesting to expand; and the object of this note is to show by referring the process to the familiar expansion curve of the indicator diagram more particularly how this is done.

The curve in Fig. 2 is that ordinarily applied to the indicator diagram, where the product of the distances of any point upon the curve from the axes is constant. For instance, the point A is 10 spaces from the OY axis and 30 spaces from the OX axis, and

$$10 \times 30 = 300.$$

The point B is 20 spaces from the OY axis and 15 spaces from the OX axis, and

$$20 \times 15 = 300.$$

In a similar way if the distance horizontally of any point whatever upon the curve be multiplied by the vertical distance from the line OX the product will be 300. It is this characteristic of the curve which makes it representative of that mode of expansion in which the product of the pressure, represented by vertical distances, and the volume, represented by horizontal distances, is constant.

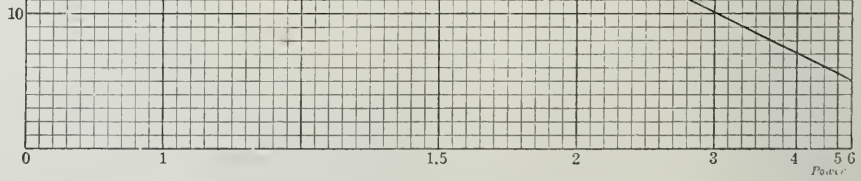

Fig. 3. Plotting the Reciprocal

To lay out the curve set off the *equally spaced* ordinates 1, 2, 3, 4, 5, 6 and upon ordinate 2, where the volume is supposed to be doubled, lay off one-half of the initial pressure; on ordinate 3 lay off one-third; on ordinate 4, one-fourth, etc.

For most purposes of the steam diagram it is better to use the curve and to keep the horizontal distances proportional to the volume or piston movement; but if it were desired simply to make a diagram from which could be measured the volume for any pressure or *vice versa* the curve could, by the simple expedient of using *the reciprocals* of the volumes instead of the volumes themselves, be reduced to a straight line. Such a straight line would be AH, Fig. 2, and the point indicating the pressure for two volumes would lie upon it at *b*, at the same hight as *B*, because, of course, the same hight must be used to indicate the same pressure upon the straight as upon the curved line. The ordinate or vertical line, upon which this point falls now becomes the ordinate for two volumes instead of four, as it was with the curved line, and has been marked 2 in the lower line of numberings for the ordinates. The pressure for three volumes would lie upon the straight line at *c* at the same hight as *C* upon the curve, and the ordinate upon which it lies is marked 3 in the lower line of values. The ordinates for four and five volumes are similarly located. As the straight line *A X* passes through the original point for six volumes its ordinate remains the same.

The point *b* indicating the pressure for two volumes must be one-half as high above the base line *O X* as is the point *A* indicating the pressure for one volume; therefore, it must be one-half the distance of *A* from the intersection *X* of the line *A X* with the base. From *X* to *Z* is 60 spaces; the ordinate for two volumes is one-half of this or 30 spaces from *X*. The ordinate for three volumes is one-third of this or 20 spaces from *X*; the ordinate for four volumes is one-fourth of 60 or 15 spaces from *X*; the ordinate for five volumes is one-fifth of 60 or 12, and the ordinate for six volumes is one-sixth of 60, or 10 spaces, from *X*. In each case the spacing is proportional to the *reciprocal* of the volume, i.e., 1 divided by the volume. The ordinate for 1.5 volumes for example, would be located

$$\frac{1}{1.5} \times 60 = 40$$

spaces from *X*, and the pressure would be that represented by a vertical distance of 20 spaces, as is indicated by the intersection of the line *A X* with that ordinate at *f*. The ordinate for 1 volume would be that under *A*, as before, for

$$\frac{1}{1} \times 60 = 60$$

The diagram will now be like Fig. 3. The spacings of the ordinates instead of being uniform as in Fig. 1, or as the spacing which produced the curve in Fig. 2, are irregular, growing closer as they go to the right. They are not, however, logarithmic like the spacing upon a slide rule. In logarithmic spacing the distance from 1 to 2 will be the same as from 2 to 4, 3 to 6, 4 to 8, etc., and the distance from 1 to 3 the same as from 2 to 6, 3 to 9, 4 to 12, etc. In Fig. 3 the distance from 1 to 2 is 30 spaces; from 2 to 4, 15 spaces, etc.

When the ratio of expansion is constant a steam engine uses a fairly constant amount of steam for each indicated horsepower-hour plus a constant quantity required to overcome cylinder losses.

Suppose the internal losses of a certain engine to be 30 pounds per hour, and suppose it to use 20 pounds additional for each horsepower developed. The steam used per hour will be as follows:

Lb. per H.P.	20	20	20	20	20	20
H.P.	0	1	2	3	4	5
	00	20	40	60	80	100
Cyl. loss	30	30	30	30	30	30
Total steam	30	50	70	90	110	130

In Fig. 4 lay off on the vertical axis *O Y* a scale of total steam consumption in pounds per hour, and on the horizontal axis *O X* a scale of horsepowers. Then set off on the ordinate for 0 horsepower 30 pounds at *a*; on the 1 horsepower ordinate 50 pounds at *b*; on the 2-horsepower ordinate 70 pounds at *c*, etc., and it will be apparent that the points lie in a straight line, for the increase per horsepower is regular, so that *c* is just as much above *b* as *b* is above *a* and for equal horizontal distances the rises (as *d b* and *e c*) will be equal. The total rise above the level of the point *a* will be 20 times the horsepower. For instance, the line *f g* giving the increase in consumption by the imposition of 4 horsepower is

$$20 \times 4 = 80$$

pounds, and the total consumption per hour is

$$80 + 30 = 110$$

pounds.

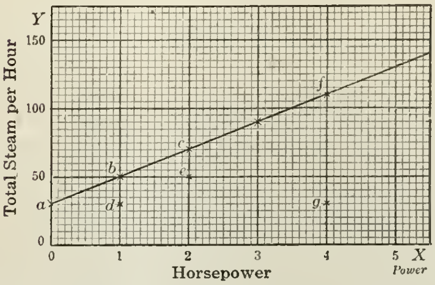

Fig. 4. Steam Consumption per Hour

Calling the total steam per hour *S*,

$$S = 20 \times H P + 30$$

The total steam per hour is the hight on the *Y* axis; the horsepower is the length on the *X* axis. The numerical quantities can have any value at all and the line will remain a straight line so long as they remain constant.

Any relation, therefore, between two varying values *x* and *y* which can be expressed by the formula

$$y = a x + b$$

will be represented by a straight line, and the instant the mathematician sees a formula of this sort he knows that if he plots *x* and *y* against each other the result will be a straight line.

The final term (*b* in this case) may be plus or minus or absent altogether without changing the line except as to the origin. If it is absent the line will commence in the lower left-hand corner at the intersection of the *X* and *Y* axes. Let us see what kind of a formula we have to deal with in the case of the riveted joint.

To find the force required to tear the plate apart as in the lower sketch in Fig. 1, the tensile strength per square inch must be multiplied by the area left between the rivet holes, the area *a b c d*, Fig. 5. This area is the product of the thickness *t* and the length *a b* which is the difference between the pitch and the diameter of the driven rivet. Calling the tensile strength in pounds per square inch *T*, the force necessary to break an element of the joint by parting the plate would be

$$t\,T\,(P - d).$$

The force necessary to shear a rivet is the cross-sectional area of the rivet in square inches multiplied by the shearing strength per square inch, or

$$0.7854\ d^2 s.$$

Making these equal to each other,

$$t\,T\,(P - d) = 0.7854\ d^2 s.$$

Substituting the values of 52,000 for *T* and 42,000 for *s* and solving for *P* produces the following:

$$52{,}000\ t\ (P - d) = 0.7854 \times 42{,}000\ d^2$$

$$P - d = \frac{0.7854 \times 42{,}000\ d^2}{52{,}000\ t}$$

$$P = \frac{0.7854 \times 42{,}000\ d^2}{52{,}000\ t} + d$$

$$P = \frac{0.6\ d^2}{t} + d$$

If a table or diagram of the values of *P* for a given diameter *d* of rivet were being made, the expression 0.6 d^2, which is to be divided by *t*, would remain the same all through the series of calculations for that size of rivet. Represent that constant by *a*. The *d* which is added to the quotient but not divided by *t* also remains constant.

Substituting *a* for 0.6 d^2 gives

$$P = \frac{a}{t} + d$$

which can as well be written

$$P = \frac{1}{t}\,a + d$$

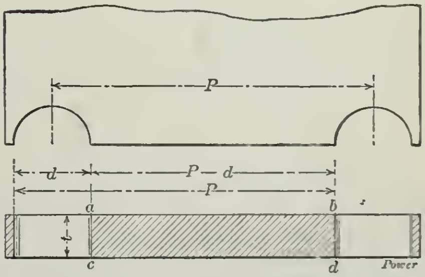

Fig. 5. Section of Plate between Rivets

This, it will be seen, is a straight line formula, *P* the hight on the *Y* axis equals $\frac{1}{t}$ of the distance on the *X* axis, multiplied by a constant with a constant added to it. It is evident beforehand, therefore, that if $\frac{1}{t}$, *the reciprocal* of the thickness, is plotted against the pitch, a straight line will result.

Let us start a plot for a ⅛-inch rivet. If $d = \frac{7}{8} = 0.875$.

$$a = 0.6\,d^2 = \tfrac{6}{10} \times \tfrac{7}{8} \times \tfrac{7}{8} = \tfrac{294}{640} = 0.446$$

Then the operation will be

$$P = 0.446\,\frac{1}{t} + 0.875$$

Thickness.	*Reciprocal of the Thickness.
$\frac{1}{4}$	$\frac{4}{1} = 4$
$\frac{5}{16}$	$\frac{16}{5} = 3.2$
$\frac{3}{8}$	$\frac{8}{3} = 2.67$
$\frac{7}{16}$	$\frac{16}{7} = 2.285$
$\frac{1}{2}$	$\frac{2}{1} = 2$
$\frac{9}{16}$	$\frac{16}{9} = 1.777$
$\frac{5}{8}$	$\frac{8}{5} = 1.6$
$\frac{11}{16}$	$\frac{16}{11} = 1.45$
$\frac{3}{4}$	$\frac{4}{3} = 1.33$
$\frac{13}{16}$	$\frac{16}{13} = 1.23$
$\frac{7}{8}$	$\frac{8}{7} = 1.14$

*To get the reciprocal of a whole number write the number as a denominator with 1 as a numerator. To get the reciprocal of a common fraction turn it upside down.

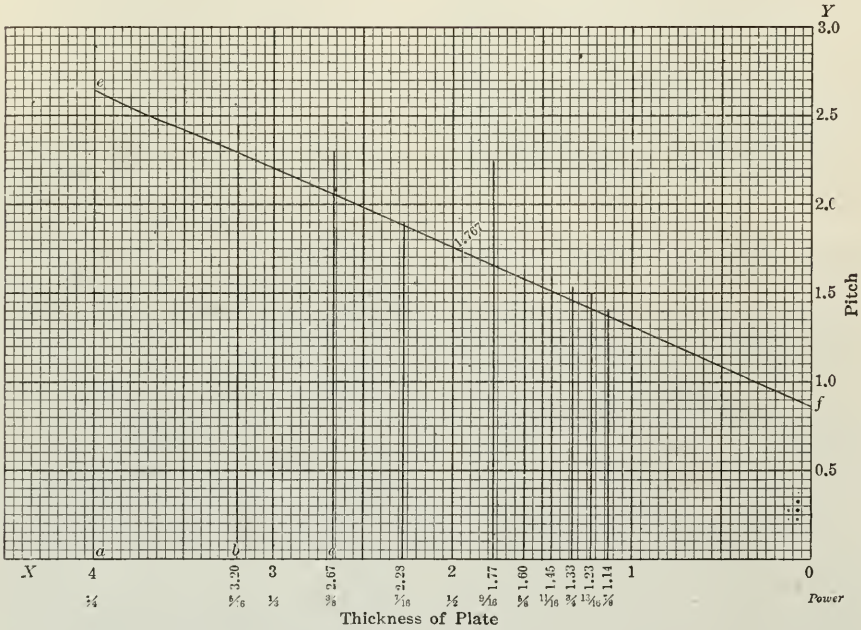

FIG. 6. RECIPROCAL OF THICKNESS OF PLATE PLOTTED AGAINST THE PITCH

The purposes of this diagram are best served by having the Y axis at the right instead of at the left-hand end of the X axis. Lay off the scale of pitches on the vertical axis $O\,Y$ in Fig. 6, and the scale of plate thicknesses along the line $O\,X$. First divide the line into a regular scale, as by the black-faced figures. I have given two of the main divisions to each unit, making each of the finer subdivisions equal to 1/20, or 0.05. Setting off the values of $\frac{1}{t}$ for the plate thicknesses we wish to consider, on this scale we find that the reciprocal of ¼ which is $\frac{4}{1} = 4$ falls upon one of the main divisions at *a;* the reciprocal of 5/16 is $\frac{16}{5} = 3.20$, and this is four of the smaller divisions beyond 3 at *b,* because such division represents 0.05 and it takes four of them to represent 0.20. The reciprocal of ⅜ is $\frac{8}{3} = 2.67$, and the ordinate for this is at *c,* three of the 0.05 divisions and 0.02 more estimated to the left of the 2.5 mark. In the same way the ordinates for the reciprocals of the rest of the thicknesses are located.

The formula is

$$P = 0.446\,\frac{1}{t} + 0.875$$

For a ¼-inch plate $\frac{1}{t} = 4$.

$$0.446 \times 4 + 0.875 = 2.659.$$

For a plate of zero thickness $\frac{1}{t} = 0$, and $0 + 0.875 = 0.875$.

Setting these values off upon the 4 and zero ordinates respectively, the points *e* and *f* are located and the line *e f* may be drawn with every confidence that the values for all of the other ordinates will fall upon it.

For instance, let us calculate that for ½-inch plate. The pitch for this thickness is

$$P = 0.446 \times \frac{1}{t} + 0.875 = 1.767.$$

If $t = \frac{1}{2}$, $\frac{1}{t} = 2$, and

$$P = 0.446 \times 2 + 0.875 = 1.767.$$

The reason that the use of the reciprocal of the volumes reduced the pressure-volume curve to a straight line is easily seen. The formula $p\,v = K$ (here K = a constant) transposes to

$$p = \frac{K}{v}$$

which produces an hyperbola, but if the reciprocal of v be used as a multiplier instead of v itself as a divisor, the formula becomes that of the straight line

$$p = \frac{1}{v}\,K$$

If you want to multiply two numbers together you can add their logarithms. The sum will be the logarithm of the product of the two numbers.

For example, multiply 2749 by 3.1416.

No.		Logarithm.
2749	=	3.4391747
3.1416	=	0.4971509
8636.3	=	3.9363256

You find in a table of logarithms the logs of the two given numbers. Add them, find the new logarithm made by their sum in the table and the corresponding number is the product.

It follows that if you wanted to multiply a number by itself a given number of times you would simply have to add its logarithm together that number of times. For example, to raise 128 to its cube or third power

128	=	2.1072100
128	=	2.1072100
128	=	2.1072100
2097152	=	6.3216300

Easier than this would be to multiply the logarithm by 3, which would, of course, be the same as adding three of them together.

To raise a number to a given power multiply the logarithm by the exponent of that power, the product will be the logarithm of the given power.

The formula, $p\,v$ = *Constant,* gives the curve *a b,* Fig. 7. The formula, $p\,v^2$ = *Constant,* gives the curve *a c.* The formula, $p\,v^{0.5}$ = *Constant,* gives the curve *a d.*

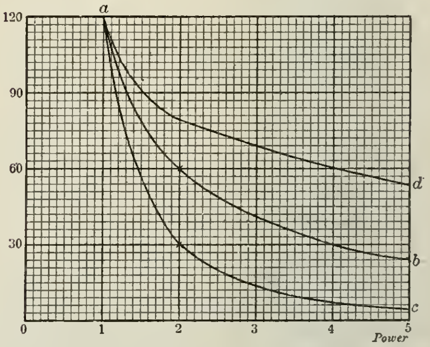

FIG. 7. PRESSURE-VOLUME CURVES

The pressures for the various volumes are gotten as follows:

For the case of $p\,v^2$ = *Constant:*

v	v^2		p		Constant.
1	1	×	120	=	120
2	4	×	30	=	120
3	9	×	$13\frac{1}{3}$	=	120
4	16	×	$7\frac{1}{2}$	=	120
5	25	×	4.8	=	120

The volumes corresponding to the ordinates of Fig. 7 are given in the first column and their squares in the second. The product of these squares and the pressures at the corresponding volume will be 120. The pressures in the third column are therefore found by dividing 120 by the squares of the volumes.

For the case of $p\,v^{0.5}$ = *Constant:* The 0.5 or ½ power of a number is its square root. In the second column below are the square roots of the volumes, which can be found in a table of squares and roots to be found in most engineers' handbooks. The pressures are found by dividing the constant 120 by these roots.

v	$v^{0.5}$		p		Constant.
1	1	×	120	=	120
2	1.414	×	84.9	=	120
3	1.732	×	69.3	=	120
4	2	×	60	=	120
5	2.236	×	53.7	=	120

By varying the exponent you can make the curve go where you please or fit any expansion line as long as it is this kind of a curve. The exponent in $p v =$ *Constant,* is, of course, unity. When the exponent becomes some such number as 1.69 it would be difficult to raise the volume to that power without the logarithms.

The formula then is

$$p v^n = K \text{ and } p = \frac{K}{v^n}$$

Since multiplication is performed by adding the logarithms, the inverse operation, division, is performed by subtracting them; then

$$\log p = \log K - n \log v,$$

which is the straight-line formula again with a simple transposition of its terms.

If, therefore, the volumes and pressures are scaled logarithmically, the curve will become a straight line and may be laid off with a ruler between two determined points and intermediate points measured on the logarithmically-spaced ordinates. Paper ruled logarithmically for this purpose is procurable from dealers in draftsmens' supplies.

The Irish Fireman and the Engineer

By E. A. Dixey

It's hard to get the best of an Irishman when it comes to a duel of wits, and this story, told me by my friend "Mac," third assistant engineer of a transatlantic liner, corroborates the foregoing statement.

"In our boats all the men are known by number. It's not like it used to be in the old tramps I sailed in: 'Here, Murphy, fire up;' or 'Swenson get some more coal on the after plate;' but it's '56 do this,' and '94 do that.' It's only one here and there whose name we know.

"I was standing with my back against the bulkhead one night on the twelve to four watch. Everything was going lovely, a fireman's fair wind, that is to say, a light breeze dead ahead, plenty of steam, good coal and nothing under the stars to bother either me or the men under me.

"Our crew was composed of Irish, Liverpool Irish at that—you know them as well as I do—and one of the trimmers, Murphy by name and 87 by number, was taking a blow under the ventilator at my elbow. Said I:

"'87, your name's Murphy, isn't it?'

"'Yes, sorr.'

"'You come from Liverpool, don't you?'

"'Oi do that, sorr.'

"'Well, I heard a story about a man of the name of Murphy and, by the way, he came from Liverpool, too. Perhaps he's a relation of yours. He died and off he went straight as a dart for heaven, and at the door he was stopped by an archangel with a flaming sword, while Saint Peter pushed back a little sliding wicket, stuck his head out and said:

"'What's your name?'

"'Murphy, sir.'

"'What are you to trade?'

"'Marine trimmer, sir.'

"'Where do you come from?'

"'Liverpool, sir."

"'Shut the gate, Gabriel, we have no use for *him* here,' said Saint Peter.

"Well, with that, Murphy, feeling kind of chilled, owing to the exposed position of the gate of heaven, decided that he would have a try at getting into another and warmer place, which he had always been told was much easier of access. So down he dropped and in a few moments found himself at the gate of hell with 'Old Nick' himself peeking at him through the bars. Murphy took heart as he saw the friendly way the devil was looking at him, and said to himself: 'Shure, he's one of my own trade, but I'm no more than an apprentice alongside of him at the coal-burning business.'

"Well, the devil says:

"'What do you do for a living, young man?'

"'My name's Murphy and I am a marine trimmer, sir.'

"'Where do you come from?'

"'I come from Liverpool, sir,' says Murphy.

"'Is *that* so?' says the devil, in a most sarcastic tone; 'then we've got no use for *you* here.'

"'But, sir,' said Murphy, 'you're not going to send me back to *Liverpool,* are you, sir?'"

"It's a good yarn, Mr. 'Mac,' and as soon as I get a bit more coal out of the bunker, sorr, oi'll tell ye wan about another Liverpool maan, but *his* name was Hennessey,'" said 87.

So I waited till he came out of the bunker again and he started:

"'The man oi'm tellin' ye about was a Liverpool fireman av th' name av Hennessey, and he died just the same as the man ye wor tellin' about, and aff he goes for heaven, hangin' onto the tail av a shootin' star, and by and by he drops aff just at the gate of heaven, stubbin' his toe on the gate post. The noise brings Peter to the gate in a hurry, and he sings out kind av short and snappy:

"'Who are you, phawt's yure name and phwere do yez come from?' all In th' wan breath. And Hennessey, seeing th' jig was up, answers much in the same way:

"'Me name's Hennessey, marine fireman, from Liverpool,' and as he turns to go he hears Peter sing out:

"'Go to hel——p fire down below, they can prob'ly use the loikes o' you.' Th' p in help was a long way afther th' l; ut's the only kind o' profanity they're allowed up there.

"So down he goes, sits down on a pile o' clinkers just outside the dure o' hell, tears a bit aff his sweat rag to tie round his stubbed toe and just then he hears a voice behind him and as he turns he sees the divil lookin' friendly loike through the bars and he says to Hennessey:

"The top o' the mornin' to ye me friend, what's your name?"

"And Hennessey said: 'Me name's Hennessey, oi come from Liverpool, and oi'm a marine fireman to trade, is there a chance for a job?"

"'Come right in,' says the devil, as he swings the dure open; 'come right in and take them four fires in the wing.'

"Well, Hennessey steps inside and over to the wing, and bein' a good man he wants to get a look at his fires. There wor plenty of firing tools, slice bars, prickers, rakes and shovels but high or low he couldn't find a hand rag, and he spent a good deal of time looking for one.

"At last the devil comes over to him and says, sarcastic: "Oi hired ye for a *fireman,* not for a sojer. Now get busy wid yer firin'!'

"'All right, sir,' says Hennessey, 'but I can't fire without a hand rag and I can't find one, so if you want me to get ahead, dig up the rag.'

"With that the divil says: 'Ut's a hand rag ye want, is it. D'ye see that box over in th' corner? Open ut and ye'll find a lot of old chief ingineers in ut, they make damned good hand rags, and be gob it's all they're fit fer.'" And 87 went into the bunker again.

When connecting boilers, exercise great care. See that the pressures on the steam gages agree before attempting to open the steam valves. Open the valves very slowly, and, if provided with small bypass valves use these before opening the large valves. Never connect boilers unless the steam pressures agree. Many explosions have been caused by neglect of these precautions.

Inquiries of General Interest

Heating Feed Water with Live Steam

If the exhaust steam is not sufficient for heating the feed water in a feed-water heater up to 210 degrees Fahrenheit, is it advisable to add live steam for the purpose of reaching the above temperature, or is it more economical to permit the water to enter the boiler at a varying temperature which is sometimes as low as 160 degrees?

J. M. D.

There is no fuel economy in heating feed water with steam drawn from the boiler which the water enters. But there are several good reasons why it is good practice to heat the feed water to the temperature of the steam in the boiler before it is pumped in, chief among which is that by so doing most of the impurities in the water are deposited in the heater instead of in the boiler.

Where the feed pipe passes through the front head to the rear, crosses and turns downward, discharging the water near the shell, the feed water will be so near the temperature of the water in the boiler that no injurious stresses will be set up by unequal expansion of the shell due to local temperature differences.

Pressure in Cylinders

With the throttle open, what pressure per square inch will there be in the cylinder with 100 pounds boiler pressure?

C. F. W.

In order to compensate for friction, heat losses, etc., 85 per cent. of the boiler pressure is commonly taken as the maximum cylinder pressure. This would give 85 pounds pressure per square inch. If the engine does not move and the throttle is open, pressure will increase in the cylinder until it equals that in the boiler, but with the engine moving, 85 per cent. of the boiler pressure may be taken as a fair average of the maximum cylinder pressure.

Raising Steam Pressure

Is it advisable to raise steam pressure rapidly in a locomotive type of boiler?

J. A.

No. As the water absorbs heat slowly, raising the steam pressure rapidly will have a tendency to injure the sheets by unequal expansion, as can be observed by placing the hand on the water leg of a boiler of the firebox type in which steam is being generated from cold water; it will be noticed that the leg of the boiler at the bottom will be cold, while at a distance of perhaps 12 to 18 inches above the sheet will be hot, thus producing an unequal expansion.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Heating Feed Water

How much good does it do to heat the feed water before it goes to the boiler?

J. C. J.

To be made into steam at working pressure the water must be heated from the temperature at which it comes to the boiler room to that of the steam which goes to the engine. All of the heat which goes to the water in the boiler must come from coal burned in the furnace and the more heat there is in the water when it goes into the boiler the less will have to be put into it to make steam. It has been found that there is a saving of 1 per cent. in fuel for each eleven degrees that the feed water is heated before it goes to the boiler.

Combustion

What is combustion, and is air necessary?

J. L. F.

Combustion is burning. It is a rapid combination of oxygen and whatever is burned and is usually spoken of in connection with the fuel used in steam making. As air furnishes the oxygen it is necessary for the process.

Pitch of Steam-pipe Line

Which way should a steam line have a fall, toward the boiler or toward the engine, and why?

F. A. C.

The line should pitch toward the engine.

Stresses in Boiler Seams

Why are the stresses in the longitudinal seams of a boiler greater than on the circumferential seams and why are the longitudinal seams double riveted?

F. A. L.

The force tending to tear the boiler apart along its length is for each inch of length double that tending to separate the shell endwise, and to resist the same pressure the longitudinal seams must be twice as strong as the circumferential ones.

Effect of Improper Air Supply

What is the effect upon combustion if too little or too much air is supplied through the fire?

J. W. A.

If the supply of air is so restricted that not enough is supplied, each atom of carbon will unite with but one atom of oxygen, forming carbon monoxide, which yields only 4500 heat units per pound of carbon consumed. In order to have perfect combustion sufficient air must be supplied so that two atoms of oxygen can unite with one atom of carbon, forming carbon dioxide. Under these conditions one pound of carbon will furnish 14,500 heat units.

Ordinarily, with the fire in proper condition too much air cannot be supplied through the fire. If, however, the fire is thin or has holes in it so that too much air is admitted, the air will cool the gases and carbon monoxide will be formed.

Waste of Fuel at the Safety Valve

What is the estimated waste of coal each minute the safety valve is open?

O. R. D.

In a 3-inch safety valve the estimated waste is 15 pounds of coal per minute.

Location of Fuses

What is the Underwriters' rule for the location of fuses in electric circuits?

R. B.

Fuses must be used at every point where the size of wire is reduced, unless the fuses protecting the larger wire are small enough to protect the smaller wire.

Safe Working Pressure

What is the safe working pressure of a 54-inch boiler, 5/16-inch plate with butt double-strap joint treble riveted; assumed tensile strength of sheet 60,000 pounds per square inch?

V. B. H.

The safe working pressure of a boiler shell, or drum, is expressed by the formula,

$$\frac{Ts \times t \times \%}{R \times F} = W.P.$$

in which

$W.P.$ = Working pressure,
Ts = Tensile strength of material,
t = Thickness of plate,
$\%$ = Efficiency of joint,
R = Radius of shell,
F = Factor of safety.

With the efficiency of the joint 87 per cent. and a factor of safety of 5, the equation by substitution becomes

$$\frac{60{,}000 \times 0.3125 \times 87}{27 \times 4} = 151.7$$

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for July, 1910, *was* 138,000.

August 2.............. 34,000
August 9.............. 34,000
August 16.............. 34,000
August 23.............. 34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

Superheated Steam

The question of superheated steam is again brought to the fore by the Bureau of Steam Engineering in a letter addressed to the users, inquiring into their experiences both as to economic results and the kinds of materials required for valves and other fittings. Perhaps a brief review of the history of superheated steam may be of interest. It was in the early sixties that Engineer-in-Chief Isherwood made exhaustive tests on the "Eutaw" to determine the economy of superheated steam. This followed certain preliminary tests on certain steamers plying on the Chesapeake bay and provided with superheaters in the uptakes, where they got the superheating for nothing and in consequence showed large gains in economy.

Now the "Eutaw" had box boilers with vertical water tubes and they showed so good economy and so far reduced the temperature of the furnace gases that there was no use in putting superheaters in the uptake. So a furnace in each boiler was transformed into a superheater, coal being burned for the purpose in that furnace. This at once touched upon one of the fallacies of the advocates of superheated steam, a fallacy that appears to need emphasis every time that the question is brought up. Isherwood found a gain in steam economy of twenty per cent., but when coal was taken as the basis of comparison the gain was reduced to sixteen per cent., which latter was well worth gaining, and superheating became the order of the day. It is worth while noting that the steam pressure was twenty-five pounds and the steam consumption was twenty-six pounds per horsepower-hour. There was trouble with superheating then as there has been since, but the real reason for the failure of superheated steam at that time was the rise of steam pressure as the progress of engineering enabled engineers to build more trustworthy boilers.

The next campaign was in 1877, when Mr. Dixwell published his results, obtained on an experimental engine in the laboratories of the Massachusetts Institute of Technology. Elaborate arrangements allowed the determination of the heat consumption so that the comparison could be based on thermal units, as should always be done in this work. Again the superheated steam showed a large advantage; measured in pounds of steam the gain was twenty-five per cent., but in thermal units the real gain was nineteen per cent. The engine developed only fifteen horsepower and at fifty pounds gage pressure used thirty-five pounds of superheated steam per horsepower per hour. Again the apparent victory was of no avail, for the simple expedient of raising the steam pressure to seventy-five pounds by the gage enabled the engine to show a consumption of only thirty-three pounds of saturated steam per horsepower per hour. Here again it was found necessary to burn fuel to superheat the steam.

At about the same time Hirn was making his celebrated tests on the interaction of the steam and the cylinder walls, and in particular the effect of superheated steam on that interaction. With an engine which developed one hundred and twenty-five to one hundred and fifty horsepower he obtained a steam consumption of about sixteen and one-half pounds of superheated steam and eighteen pounds of saturated steam at ninety pounds gage pressure. Both results are most creditable. Reckoned in steam the gain from superheating amounted to nearly twenty per cent. This again must be discounted to about sixteen per cent. when computed in thermal units. But other tests by Hirn on an engine with steam jackets showed a steam consumption of seventeen and four-tenths pounds of steam, which, if allowance were made for heat in the jacket drains, would come to nearly the figure given for superheated steam. And in 1884, Delafond was able to get an economy of sixteen and five-tenths pounds of saturated steam per horsepower per hour with the aid of steam jackets. So again the superheated steam won in direct test only to be beaten in practice by a more convenient engineering expedient.

Then the matter rested till Schmidt developed his engine for using highly superheated steam and showed such startling results on ten pounds of superheated steam per horsepower per hour in a compound engine of seventy-five horsepower, with one hundred and seventy pounds steam pressure. The conditions of the tests do not allow a ready comparison with the best practice when using saturated steam, so we may turn to the test made by Professor Schröter in 1904 on a compound steam engine using about one hundred and thirty-five pounds steam pressure. He obtained an economy of nine and six-tenths pounds with superheated steam and twelve pounds with saturated steam. The apparent gain from superheating was twenty per cent., if the steam consumption is taken as the basis of comparison; but if the heat in thermal units per horsepower per minute be taken, as is proper in this case, the gain is about six per cent. This, by the way, is almost exactly the gain in passing from a compound engine with one hundred and thirty-five pounds pressure to a triple engine with one hundred and seventy-five pounds pressure.

And now, for comparison, let us turn to a test made by Professor Miller on the Chestnut Hill pumping engine, which used eleven and two-tenths pounds of saturated steam per horsepower per minute when developing five hundred and seventy-five horsepower. This engine used two hundred and four thermal units per horsepower per minute with one hundred and seventy-five pounds gage pressure, and the compound engine tested by Schröter used one hundred and ninety-nine thermal units. The difference is of little importance to practical engineers; so it may be said that in the last bout the victory lay with superheated steam, but that a more convenient device robbed the victor of his laurels. Lest the question of the use of superheated steam in triple-expansion engines should appear to be neglected, it may be said that tests show that the gain is too small to be of importance.

That appears to be the condition of superheated steam for reciprocating engines. In every trial there is a decided gain from superheating, usually discounted in a discouraging way when reduced to the basis of coal or heat consumption, but still a very marked gain; but equally good results can be had by some other way which practical engineers prefer.

As for steam turbines, there is a strong claim by the builders that superheated steam shows a valuable advantage, with the mechanical advantage of reducing the amount of water in the lower stages of the turbine. There is reason for treating this claim with respect, even if it be not accepted without reservation. But the conservative engineer has reason for waiting before committing himself on the subject.

The Modern Catalog

All successful business men recognize the value of publicity. Dignified advertising has long since ceased to be experimental—today it is a necessity. While it may take many forms, its best exponent is printer's ink—the press, the circular, the catalog.

Much gray matter—and money as well—are expended in the preparation of catalogs, in the subjects discussed, the manner of presentation, the arrangement, subdivision, indexing, etc. In mechanical execution they are fair to behold. The good taste of the typography, the paper, the color scheme, the engraving and the binding, leave nothing to be desired. Such productions are welcomed and preserved.

They appeal to the manufacturer and the engineer, for in them are to be found many of the latest and best data. Recent advances in the arts and sciences, new apparatus and machines, new uses of existing machines, needed data as to output, efficiencies, costs, the establishment of standards, etc., all these are here ready at hand for use.

It is of vital importance both to the sender and the receiver that such publications should attain their highest usefulness. Unfortunately, there is a reason to believe that these monumental productions often fall short and that there is waste to the sender and annoyance to the receiver. The net result is loss to both.

The modern business man, whether manager, purchasing agent or engineer, must be systematic. He is a man of card indexes and filing systems. He must be able at a moment's notice to put his hand upon any one of many thousand scattered and unrelated bits of data. When he receives a large bound catalog, containing data on boilers, engines, pumps, heaters, valves and pipework, and perhaps also machine tools and contractors' supplies, he is at a loss to know what to do with it. If he files it under any one of its divisions it is lost as to the others. True, he may card index each item in it, as some do, but that is an endless job. Often, perhaps generally, it is placed among other similar general catalogs and is more than likely overlooked when a study is being made of a particular subject.

Two additional objections apply. The space limitations of a general catalog preclude exhaustive treatment of any subject; a concise summary is all that can be expected. Furthermore, all live subjects advance; new ideas, new uses and new patterns tread upon each other's heels. Parts of such a catalog soon become obsolete while other parts are still in full usefulness.

Far better is the modern bulletin system, though it lack somewhat of ponderous dignity. The manufacturer can afford to sacrifice something in color scheme and in early English effect if he thereby gains in clearness and uptodateness. That he also reduces costs and improves efficiencies are incidental, but none the less important advantages.

A separate bulletin for each independent subject is issued whenever the situation warrants. It may be as concise or as diffuse as the occasion justifies. It may be replaced at small expense by a later issue when advances in the art require. Best of all, it may be filed with other literature on the same subject, where it is sure to be found when needed. An inquirer about a special article gets only what he asks for; there is no waste on either side.

Some manufacturers maintain regular mailing lists of persons to whom bulletins are sent as fast as issued. Some of them get data along only those lines concerning which they have filed requests. Cards of acknowledgment are inclosed, the signing and returning of which indicate the recipient's continued interest. Many manufacturers furnish a binder in which all their publications may be preserved together if desired, thus forming a perpetually uptodate catalog.

Good progress has been made toward the standardization of sizes of catalogs. Why not go a step further and standardize their form? The large general catalog, with its multitude of more or less unrelated topics, is a relic of the past. All hail to its successor, the unpretentious bulletin, which serves a single master, but serves him faithfully and well.

Electricity Supply in Great Britain

The economy of aggregation at a point where condensing water is available and fuel accessible without cartage, has resulted in the consolidation of plants of moderate size for the generation of electricity into immense central stations, delivering current at high tension to substations for local distribution. Is it possible that there is to be a swing of the pendulum in the other direction?

The Bow Road station of the Charing Cross, West End and City Electricity Supply Company, at West Ham, London, is equipped with Sulzer steam engines. H. W. Kingston, chief engineer of the company, instead of using his substations merely for transformation, is equipping them with oil engines of the Diesel type, also built by the Sulzers. The units at present installed run from 300 to 450 kilowatts each, but there are some in course of construction having a capacity of 1000 kilowatts. These engines use an oil from which the naphtha and gasolene have been distilled and they produce a kilowatt-hour for about three-eighths of a cent, while at the main station the cost is just under half a cent; both of these figures include all operating charges. When the overhead charges are added, the use of the oil engine substations gives a decided advantage in the reduction of the investment in copper.

All theaters and public places are by law required to have two separate systems of supply. The average price collected is a little above one penny (two cents) per kilowatt-hour.

Gas engines are like some men, they must have either a steady load or a good governor to hold their speed within bounds.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Twinvolute Turbine Pumps

Turbine pumps have come into popularity because of their efficiency, compactness and adaptability to all methods of drive.

The Watson-Stillman Company, New York City, holds the exclusive rights to manufacture the Twinvolute type of turbine pump. The difference between this type and the ordinary concentric turbine pump, in which the waterway is concentric with the runner, is shown in Fig. 1. In the Twinvolute the waterway at the edge of the casing varies in size from its fullest width at the outlet to the narrowest point at the opposite side.

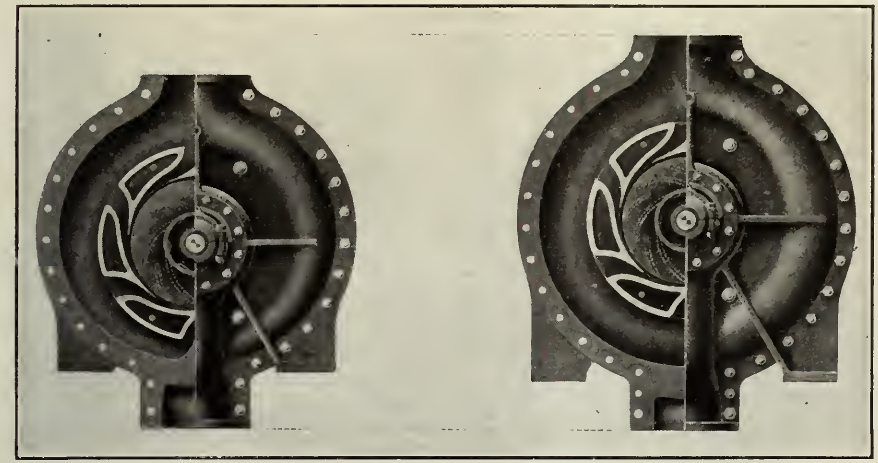


Fig. 1. Relative Sizes of Twinvolute and Ordinary Concentric Pumps

With this design many advantages are secured. It is assumed by the makers of turbine pumps of the concentric-waterway type that when the water leaves the diffusion-vane outlets the water divides and passes in equal amounts around and through the concentric ways. Theoretically this is right, but it has been proved by experiment that the parting is seldom at the mid-point; that is, diametrically opposite to the pump-discharge outlet. In fact, no two pumps have the dividing point in the same place. This is due, probably, to inaccuracies and variations in the smoothness of the inside surface of the pump shell.

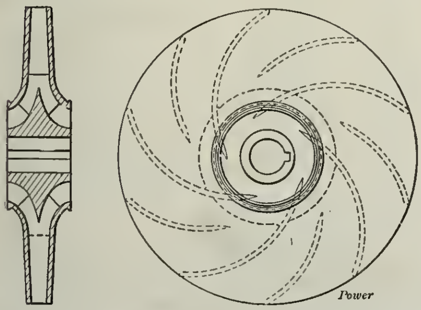


Fig. 2. Impeller of Single-stage Twinvolute Turbine Pump

If the actual dividing point does not correspond with the theoretical, it is obvious that the water must do more work on one side than on the other in reaching the discharge outlet. This results in a drop in the pump efficiency.

The water leaves the discharge outlet of any turbine pump at a predetermined velocity, generally 10 to 12 feet per second. If, when the water passes through the diffusion-vane outlets, it enters a waterway concentric in shape and of an area equal everywhere to that at the discharge outlet, there is a constant increase and decrease in velocity and pressure in the waterway.

The Twinvolute waterway avoids these errors of unequal work, pressure, velocity and objectionable eddies, and adds good features not obtainable with the ordinary construction. As the water leaves the diffusion vanes it is compelled by the construction of the volutes to part at the correct center. Each volute must carry an equal amount of water; hence, the distribution of work is uniform, and every drop of water is compelled to reach the pump discharge by the shortest course. The Twinvolute is so graduated

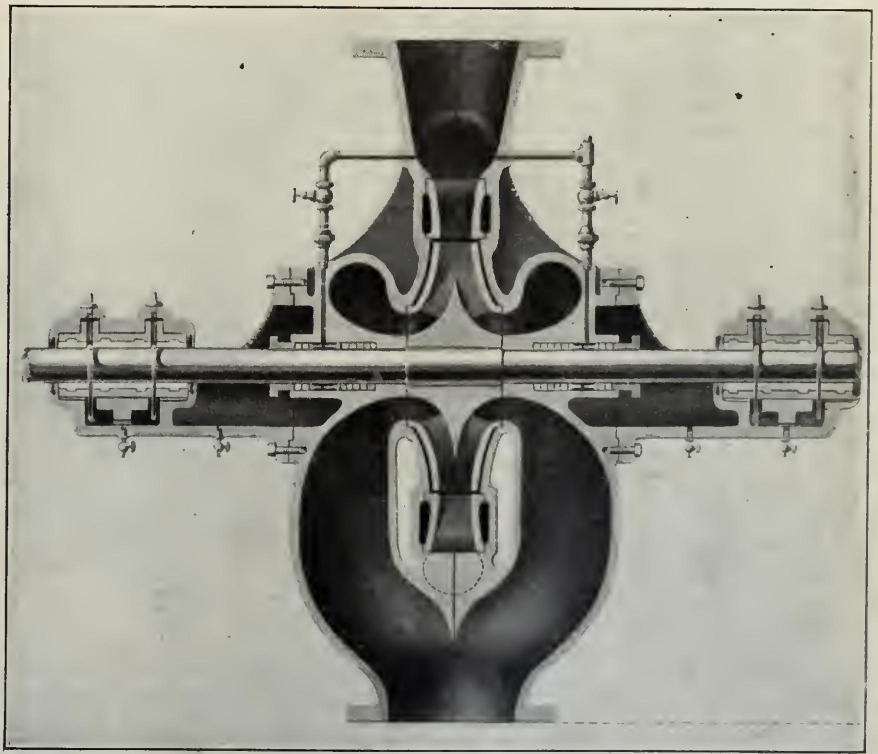


Fig. 3. Cross-section of Single-stage Pump

in sectional area that as the water issues from the diffusion-vane outlets, the additional amounts of water on their way to the discharge outlet are given ample space and variation of pressure at the different diffusion-vane outlets is prevented. The predetermined velocity is, therefore, constant; there are no whirls or eddies and the efficiency must remain at the maximum. Also, with this construction, the size and weight of the pump are materially decreased, thus lowering the first cost. A still further saving is effected by the reduction in the size of the foundations and the amount of floor space required. The internal area or wetted surface is less than with the concentric waterway. The path of travel of a particle of water from the eye of the impeller out through the diffusion vanes and twinvolute to the discharge opening is sufficiently less than in other forms of construction to make a noticeable reduction in the friction loss. Further, because of the twinvolute, the center of gravity is lowered and the vibration reduced. This means longer life for the pump and an absence of noise, which with some types may be troublesome.

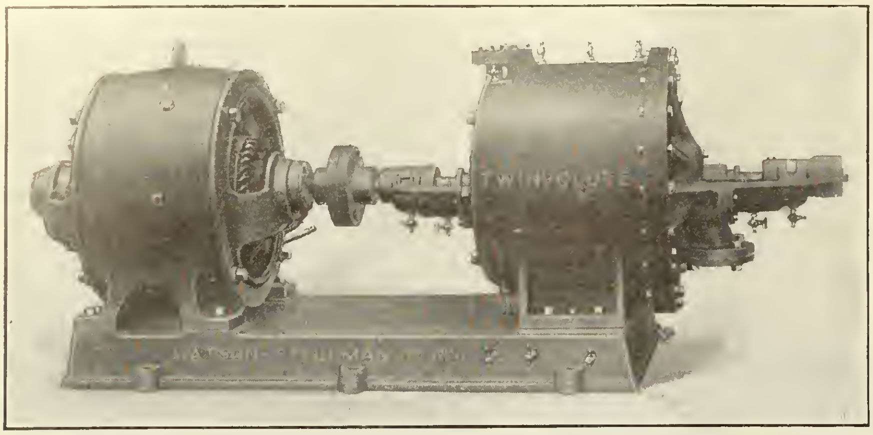


Fig. 4. Motor-driven Twinvolute Pump for Boiler-feed Service

As shown in the cross-section, Fig. 3, water in the suction pipe must divide in the Y passage of the single-stage pump and flow in equal amounts into each side of the impeller.

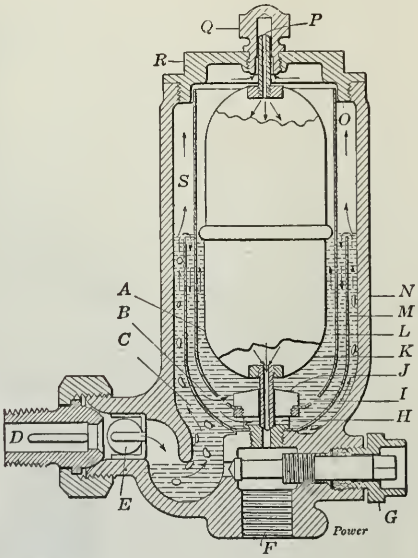
Symmetry is embodied in the impeller design so that as the water enters the eyes on each side, there is equal pressure in opposite directions. The pressure being equalized there is a natural balance and end thrust is avoided.

The divided suction is especially desirable in high-speed and large-size pumps for, by reason of the double openings, the impeller may be made smaller in diameter than in pumps where all of the water must flow into one set of openings. This means a smaller pump and a consumption of less power.

The Watson-Stillman Company manufactures turbine pumps for capacities of from 2 to 150,000 gallons per minute, for suction lifts of from 1 to 25 feet, for heads of from one pound up to that equivalent to a discharge pressure of 500 pounds and for speeds of from 50 to 3000 revolutions per minute. The pumps may be operated by gearing, belts, silent chains, or direct-connected electric motors, gas engines, steam engines, steam turbines or water turbines. The pumps may be used for practically any service, such as heating systems, water works, hydraulic sluicing, condensing systems, storage tanks, boiler feed, fire protection, irrigation and mine drainage.

Monash Radifier Valve

Two great faults that heretofore have been encountered in the erection of vacuum steam-heating systems are claimed to be done away with by the installation of the Monash radifier; that is, the noise caused by the chattering of the float and the removal of the valve for cleaning.



Monash Radifier Valve

The still body of water *A*, see illustration, in which the float is incased, prevents the chattering of the float, due to air or water currents. The seat *J* can be unscrewed and the interior of the valve taken out and thoroughly cleaned without disconnecting the valve from whatever it is attached to.

The valve is attached to the return end of the radiator or heating device by the union nipple and to the return pipe at the outlet *F*, which is connected to the vacuum pump.

The air and water of condensation are drawn by the vacuum from the radiator into the radifier through the inlet *D*, past the ball check *E*, down into the gooseneck trap, where any heavy dirt or foreign matter is deposited, which may be blown or drawn through the bypass valve into the return.

The air and water of condensation pass upward in the chamber *C*, when the air separates from the water and is drawn through the chamber *S* and the opening *O* to the hollow-float guide pin *P*, through the float *K* down to the return through the outlet *F*.

The water of condensation, being drawn up into the body of the valve, rises to the upper edge of the outer shell *M*, flooding the chamber *B* and also gradually filling the chamber *A* by slowly passing through the openings *I*. This opening is of small dimension and prevents the sudden rush of water into the chamber *A*, thus breaking the current or force of the water, and prevents the chattering of the float.

The water gradually rises around the float to a point near the seam, when it raises the float *K*, thereby opening the discharge passage through the valve seat *J*.

While the valve is open the chamber *B* is drained by the water passing through the opening *H* in the valve seat and to the return pipe through the outlet *F*, and since the discharge is over the top of the shell *M*, the oil, grease or floating matter is skimmed and is carried off with the water.

The receding water in chamber *A* keeps the float buoyant until the valve is closed, thereby insuring a noiseless operation of the float.

The phosphor-bronze ball check *E* is held in place by a copper basket and, seating in the union nipple, prevents the vacuum in the radiator reacting on the vacuum in the return when the radiator supply valve is closed.

To clean the passageway of the float, the cap *Q* may be unscrewed and a wire forced through the openings in the float guide stem.

The bypass valve is protected by a locked shield feature *G* which prevents tampering with the valve.

This valve is made by the Monash-Younker Company, 1407-17 W. Jackson boulevard, Chicago, Ill.

The Perfection Calorimeter

The testing of coal, coke and liquid fuels to determine their heat content can be accomplished by anyone of ordinary intelligence and with very little expense by the aid of the instrument described and illustrated herewith.

In Fig. 1 is shown a sectional view of the Perfection calorimeter, manufactured by the Western Meter Company, Davenport, Ia. The metal can *A* is filled with water; *B* and *C* are asbestos cylinders for insulating the instrument from the effects of atmospheric changes while the determination is in progress.

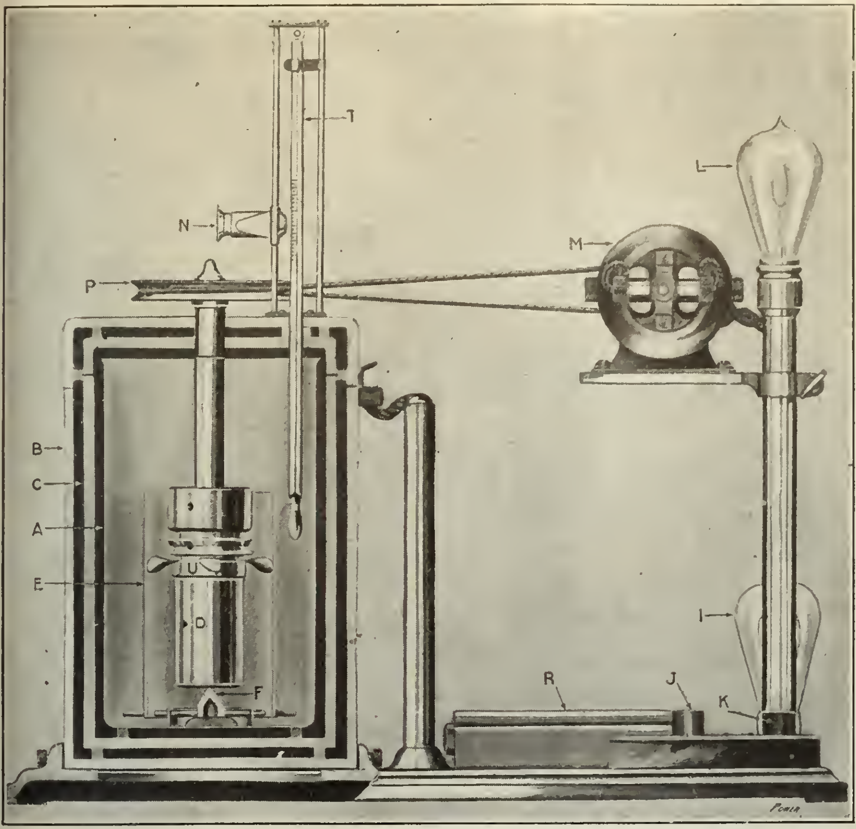


Fig. 1. The Perfection Calorimeter

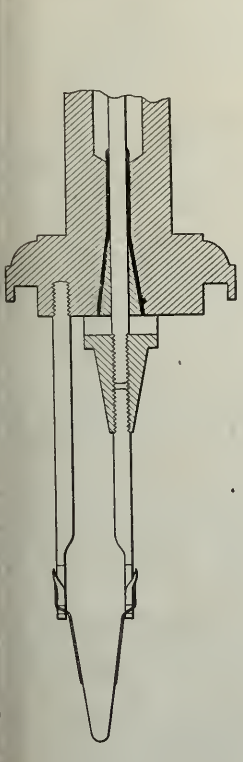


Fig. 2.

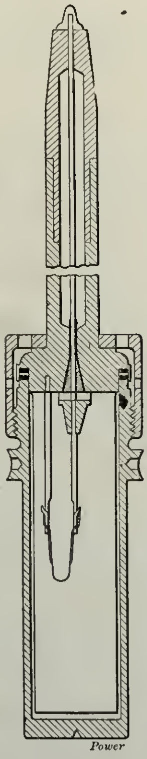


Fig. 3.

The bomb *D* is equipped with an improved combustion chamber in which is placed a weighed quantity of coal or fuel to be tested. The deflector *E* deflects the currents of water, thereby facilitating the equalization of the temperature. The bomb rotates on the pivot *F*. A light which indicates when the charge is ignited is shown at *I*; *J* and *K* are switches for controlling the motor and ignition, and *L* is a light in series with the motor. A telescopic lens *N* is provided for reading the thermometer *T*.

The auxiliary chamber in which the combustion takes place is entirely surrounded by air, so that the sides of the chamber become red hot and cause even the most refractory particles of fuel to be consumed. The instrument is especially adapted for the testing of liquid fuels.

Fig. 2 shows the method of igniting the charge and of fastening the fuse wire *G*. At each end the fuse wire is wound into loops and is attached to hook-shaped lugs. Fig. 3 shows the fuse wire as it appears in the bomb.

When ready to test, weigh out exactly one-half gram of commercially dry coal and place it in an oven to dry. When the coal has been dried it is transferred to the combustion chamber which has been removed from the bomb *D*, Fig. 1. To the coal one measure of chemical is added before the cylinder is replaced in the bomb, the cap adjusted and the whole vigorously shaken. The can *A* is removed from the instrument and filled with exactly 2000 cubic centimeters of water. It is then replaced, the bomb *D* inserted, the cover on the instrument put in place and the thermometer *T* inserted so that the mercury bulb is located at about the center of the can *A*.

The pulley *P* is connected by a small belt to the motor, which rotates the bomb *D* and stirs the water.

The temperature of the water will equalize in about three minutes, it is claimed; the fuel charge can then be ignited by pressing the button *J*, when the heat of the combustion is imparted to the water, and the rise is recorded by the finely graduated thermometer *T*. The rise is then multiplied by a simple constant, and the result is the exact number of British thermal units per pound of fuel tested.

The instrument is complete, mounted on a base 11x22 inches, and is ready for use when connected to any lamp socket available.

The Mechanical Engineers at London

The mechanical engineers left Birmingham on Thursday morning and after visits to Warwick, Stratford-on-Avon and Kenilworth, were conveyed by special train to London, where they arrived in time to be received by President Aspinall of the British Institution at a conversazione at the Institution home. The feature of the evening was a lecture by Doctor Hele-Shaw on "Stream Lines as Affecting Aviation." Stream lines were projected upon the screen by injecting coloring matter into a sheet of glycerin flowing between two glass plates in the lantern, and the area of reduced pressure behind a plane in the stream was very effectively shown by using a less viscous fluid.

On Friday morning a session for the consideration of professional papers was held at the Institution of Civil Engineers, and in the afternoon teas were given by Sir John I. Thornycroft and William Maw. In the evening President Aspinall entertained at dinner, some five hundred covers being laid. The usual line of toasts were drunk, that to the American Society of Mechanical Engineers being proposed by Sir William White and ably responded to by Dr. F. R. Hutton.

On Saturday the party was taken to Windsor and for a sail upon the Thames between Windsor and Marlowe, reaching London by special train in the early evening. A visit to the Japanese exposition on Saturday evening completed a day every minute of which was full of interest.

On Sunday evening a special service was held in Westminister Abbey and an opportunity afforded to see the memorial window to Sir Benjamin Baker, who was an honorary member of the American Society.

It was an exhausted party which dispersed at the end of the program, which, for the Americans, had included two weeks of incessant entertainment and activity. Everybody was delighted with his experiences, and full of appreciation of the generous and spontaneous hospitality which made the visit such a mélange of pleasurable impressions.

BOOK RECEIVED

"Electric Wiring." By Joseph G. Branch. Branch Publishing Company, Chicago, Ill. Cloth, 287 pages, 5⅛x 7¾, 93 illustrations, indexed. Price, $2.

PERSONAL

P. M. Chamberlain and S. S. Howell, both of whom were formerly connected with the Under-Feed Stoker Company of America, have formed a partnership and are engaged in general consulting-engineering work in Chicago. Their offices are in the Marquette building.

A. F. Norcross has succeeded Hubert E. Collins as general manager of the Engineering Supervision Company, 45 West Thirty-fourth street, New York City.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

REVOLVING GAS ENGINE. George Haniquett, Longbeach, Cal. 965,385.

WINDMILL. Henry I. Hawxhurst, Los Angeles, Cal., assignor to Hawxhurst Wind Turbine Power Company, Los Angeles, Cal., a Corporation of California. 965,387.

STEAM TURBINE. Rudolph Schlatter, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 965,412.

COMPOUND STEAM PUMPING ENGINE. Alexander England, Wilkinsburg, Penn., assignor to the Westinghouse Air Brake Company, Pittsburg, Penn., a Corporation of Pennsylvania. 965,573.

ROTARY ENGINE. Richard Lundquist, Laguna de Terminos, Mexico. 965,640.

EXPLOSIVE ENGINE. Jay F. Woolf, Minneapolis, Minn., assignor of one-half to Ellis J. Woolf, Minneapolis, Minn. 965,975.

COMBINED LIQUID AND GAS TURBINE. Carl E. Brockhausen, Chicago, Ill. 964,985.

HOT-AIR MOTOR. Guillaume Mann, Paris, France. 966,032.

CURRENT MOTOR. Edgar A. McClung, Dayton, Wash. 966,041.

ROTARY CURRENT MOTOR. Edgar A. McClung, Dayton, Wash. 966,042.

INTERNAL COMBUSTION ENGINE. Alphonse Butsch, St. Lucia, British West Indies. 966,309.

GAS TURBINE. Leo Samoje, Antwerp, Belgium. 966,363.

INTERNAL COMBUSTION ENGINE. Erik Anton Rundlof, Stocksund, near Stockholm, Sweden. 966,362.

BOILERS, FURNACES AND GAS PRODUCERS

GRATE. Fred Hilburger, Castle Rock, Colo. 965,186.

GARBAGE BURNER AND WATER HEATER. William G. Ross, Chicago, Ill., assignor of one-half to Frank Sutherland, Chicago, Ill. 965,229.

CRUDE-OIL BURNER. Oskar Kirschen, Kolomea, Austria-Hungary. 965,305.

GRATE. John Levey, Chicago, Ill., assignor to John Williamson, Chicago, Ill. 965,589.

OIL BURNER. Arthur H. Light, Los Angeles, Cal. 965,719.

OIL BURNER. William E. Weathers, Frederick, Okla. 965,775.

DIRECT-ACTING PUMPING MACHINERY. John Hutchings, London, England. 965,819.

DISTRIBUTING VALVE. John Hutchings, London, England. 965,820.

CRUDE-OIL BURNER. John A. McCoach, La Cygnes, Kan. 965,831.

STEAM BOILER. Thomas E. Durban, Erie, Penn. 965,883.

BURNER FOR FUEL OILS. Albert W. Thompson, Manchester, N. H., assignor of one-fourth to Harry E. Blanchard, Manchester, N. H. 965,963.

SMOKE-CONSUMING APPARATUS. William Ramsay Marshall, Oldham, and Thomas Hargreaves, Ashton-under-Lyne, England, assignors to Victory Smoke Consumer and Fuel Economizer Company, Ltd., Oldham, England. 966,109.

SPRAY BURNER. Archie T. Rigg, Altoona, Penn. 966,124.

STEAM GENERATOR. William George Hay, Tuebrook, Liverpool, England. 966,201.

FURNACE. John Harrigan, Brooklyn, N. Y. 966,200.

FURNACE. Carl Nordell, Stamford, Conn. 966,233.

POWER PLANT AUXILIARIES AND APPLIANCES

OIL CAN. Luther M. Graves, Philadelphia, Penn., and James Woodward Allen, Lumberton, N. J. 965,578.

HOSE COUPLING. Joseph W. Ferguson, Quincy, Mass. 965,286.

THROTTLE. Jonathan Peterson, Brooklyn, N. Y. 965,322.

PISTON AND PISTON VALVE. Robert Allen, Caversham, England. 965,355.

SPARK PLUG. Robert T. Estes, and John P. Inman, Midland, Tex. 965,380.

TURBINE PACKING. Robert A. McKee, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 965,399.

COMBINED THROTTLE VALVE AND OILER. Lewis C. Bayles, Johannesburg, Transvaal, assignor to Ingersoll-Rand Company, New York, N. Y., a Corporation of New Jersey. 965,554.

IGNITION TIMER FOR INTERNAL COMBUSTION ENGINES. Francis C. Mason, Grand Rapids, Mich. 966,034.

VALVE. Eugene J. McCarty, Clinton, Mass. 966,040.

VALVE MECHANISM FOR INTERNAL COMBUSTION ENGINES. Hermann Steingassner, New York, N Y., assignor, by direct and mesne assignments, to Alfred Hutter, New York, N. Y. 966,367.

VALVE. James P. Haig and Alvin H. Porter, Grand Rapids, Mich. 966,198.

STEAM-POWER APPARATUS. Wilbur C. Anderson, Canton, Ohio. 965,850.

CARBURETER. Paul Bustard, Frederick, Okla., assignor to Eskew H. Archer, Frederick, Okla. 965,867.

SPARKING PLUG FOR EXPLOSIVE ENGINES. Cyrus F. Jones, Oregon, Ill. 965,585.

METALLIC PACKING. William B. Claflin, New York, N. Y. 965,688.

PRESSURE GOVERNOR FOR ELECTRICALLY OPERATED PUMPS. Francis L. Clark, Pittsburg, Penn., assignor to Standard Traction Brake Company, a Corporation of New Jersey. 965,689.

DRAFT REGULATING DEVICE FOR FURNACES. Wilber P. Hall, Pembroke, N. Y.: Alida L. Hall, administratrix of said Wilber P. Hall, deceased. 965,815.

ELECTRICAL INVENTIONS AND APPLICATIONS

COOLING TOWER. Donald Barns Morison, Hartlepool, England. 965,116.

VALVE. Charles Walker, Knoxville, Tenn. 965,129.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., V. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

Vol. 32 NEW YORK, AUGUST 30, 1910 No. 35

THE chief in a large power house asked one of the watch engineers, a young man, to attend to some minor routine work temporarily. The work was a little beneath what the young man thought he was hired for. Besides, he thought that *he* was too good an engineer to be picked out to do this bit of routine.

He considered that it was below his dignity.

Why could not the chief have asked a fireman to do the work? Anyhow, why did he not ask one of the men on the other watches? Those other fellows had an easier time of it than he.

The young man brooded on in this manner until he became an ordinary "grouch." He sulked and did not do as he was told. The result was that he was unceremoniously "fired."

The chief had no intention of imposing on him or showing favoritism to the others. In fact, up to the time of the episode just related, the young man stood highest in the opinion of the chief.

What then was the cause of the young man's downfall? Nothing in the world but the influence which he allowed jealousy to have over his imagination.

This case is not as sad as it might have been. The man is young and intelligent. He probably will discover that it is unprofitable to allow his emotions to blind his reason.

We could go on and recount a thousand and one stories from life, all with the same moral. The well known dog-in-the-manger yarn has the same underlying theme. Because "Tige" could not eat the oats he was unwilling that "Dobbin" should. There are a wonderfully large number of human "Tiges."

Probably no other emotion in the whole complex category of human nature has a greater influence against progress on the part of both the individual and the nation as jealousy. History teems with evidence of this fact. Jealous fear on the part of a few retarded the spread of education for centuries. Innumerable battles have been lost through foolish, childish jealousy on the part of the leaders. Business houses have gone down to ruin simply because the managers would not trust one another and were jealously suspicious of each other's motives.

"A house divided against itself cannot stand."

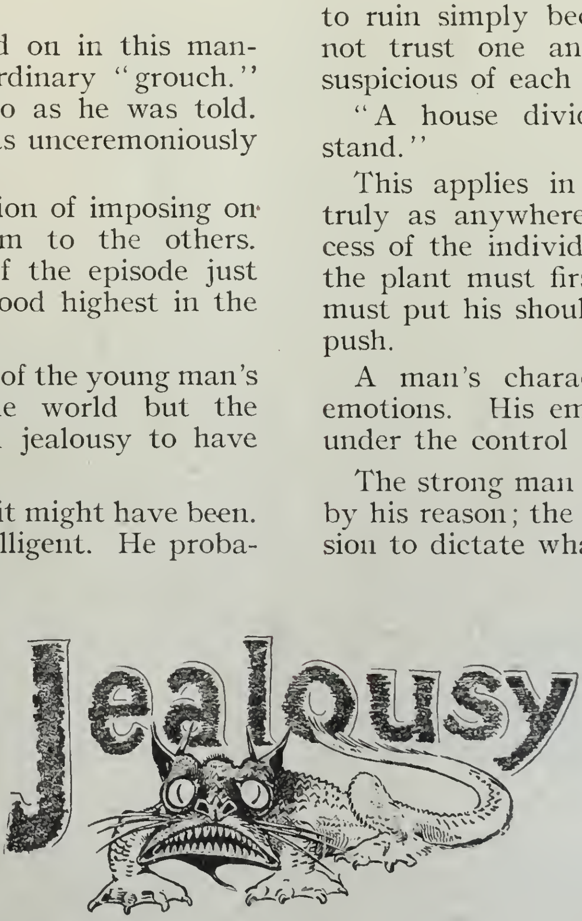
This applies in the power plant just as truly as anywhere else. To make the success of the individual possible, the success of the plant must first be accomplished. Each must put his shoulder fairly to the wheel and push.

A man's character is influenced by his emotions. His emotions are to some extent under the control of his reason.

The strong man strives to rule his emotions by his reason; the weak one allows blind passion to dictate what his actions shall be.

Never yet did an act inspired by jealousy secure a lasting benefit for anyone.

On the other hand, generous motives often have prompted acts which unexpectedly have been handsomely rewarded.



New Power Station at Scranton, Penn.

By F. W. Brady
and C. J. Mason

> *An uptodate, 1000-kilowatt plant supplying direct current for street-railway service. A poor quality of culm is used for fuel, but in spite of this the plant shows a high efficiency.*

The new power station recently erected for the Scranton Street Railway Company was designed, not only to meet the demands for more power, but also for the purpose of eventually replacing the old power station. The old station had met with a number of mishaps which had crippled the service at times and had forced the company to purchase power from one of the suburban companies. An attempt was made a few years ago to increase the output by the installation of an exhaust-steam turbine, which installation was described in POWER of March, 1906. This turbine installation was more or less successful and bridged over some of the difficulties, but it is believed that the whole equipment of the old station will soon be superseded by the more modern one herein described.

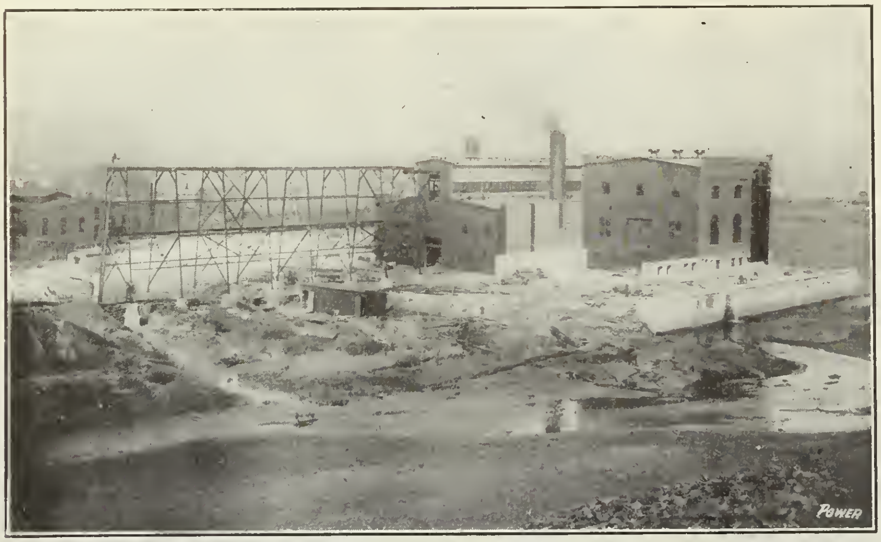


FIG. 1. EXTERIOR VIEW OF PLANT

The new station, built of red brick and steel with reinforced concrete foundations, stands upon an ash dyke at the rear of the old plant and is close to the Lackawanna river, which furnishes water for condensing purposes. This stream is black and turbulent, containing much sewage, and is charged with acid-mine water. At certain seasons it overflows its banks and causes considerable damage to surrounding property. The foundation of the engine-room portion rises above the high-water level of the river, but the floor of the boiler room is lower and there is a possibility of it being flooded should the river rise unusually high.

With the present installation the capacity of the new plant is 1600 kilowatts, which is the same as that of the old one. An extension, when completed, will double this capacity. The foundation for the addition to the engine room is ready to be built upon as can be seen in the general view, Fig. 1. The boiler house requires but three more boilers to complete the original plans. The boiler room is especially roomy, well ventilated and light, and the equipment is arranged to facilitate overhauling and repairs. There is no cellar under the boiler-room floor, but under the engine room there is a basement in which are located the circulating pumps, oil tanks and filters.

BOILERS

The plant contains five Stirling boilers with room for a future installation of as many more. Fig. 2 shows the front of one battery of boilers with the coal bunkers and measuring chutes. Each boiler has 4348 square feet of heating surface and 136 square feet of grate surface, and is rated at 435 horsepower, making a total of 2175 boiler horsepower at present installed. The ratio of grate surface to heating surface is 1 to 32. The steam drums are made in one plate, of open-hearth flanged steel having 54,000 to 62,000 pounds tensile strength, with double-riveted, butt joints. The boilers are allowed a working pressure of 180 pounds per square inch, although the safety valves are set to blow off at 165 pounds.

The furnaces are equipped with Wildt's patent grates. These grates consist of a number of individual rocking grate bars; each bar made up of a number of fan-shaped cast-iron sections strung on two rods and clamped together. The sections have suitable bosses and recesses so that when clamped together they are not only properly spaced but also form one rigid

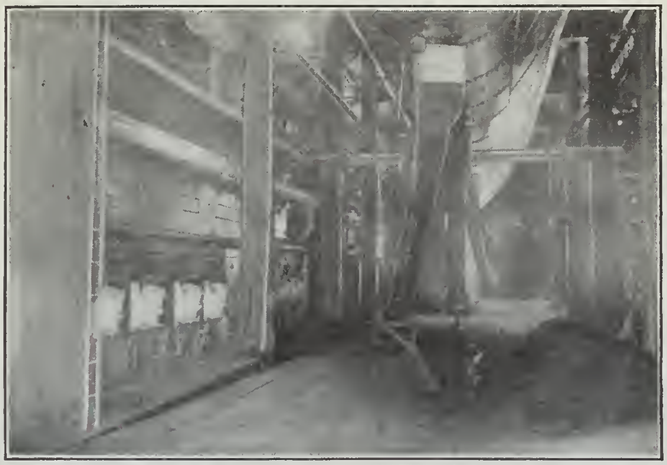


FIG. 2. BATTERY OF BOILERS AND COAL BUNKERS

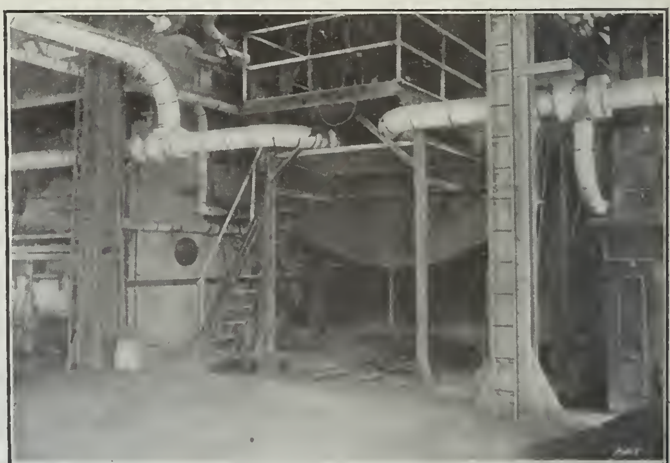


FIG. 3. INDUCED-DRAFT UNIT

bar. If a section at any part of the bar needs replacing, the repair is made by knocking out the defective section and adding another at the end of the rod. For burning culm, taken directly from the dump without any sorting or cleaning, a grate with ⅛-inch air spaces is used. This fuel is hand fired, the depth on the grates being about 12 inches. It has been found desirable to use a small amount of superheated steam from jets placed under the grates to prevent clinkering.

Each boiler is equipped with a Copes automatic feed regulator. The feed water is discharged from the pump into a 4-inch pipe, thence into a 5-inch main. A venturi meter having a combined register indicator and graphic recorder is connected to the main. From this main, a 2½-inch pipe leads to each boiler. All the drips from the steam header and separators are trapped to a 5000-horsepower Cochrane heater. These drips may be bypassed to the river in case the heater is out of service.

Feed Pumps

There are two 4-inch, two-stage centrifugal boiler-feed pumps, each direct connected to a De Laval turbine rated at 55 horsepower. One of these pumps is shown in Fig. 4. Each pump is capable of delivering 250 gallons of water per minute against a head of 382 feet. The speed variation is not more than 4 per cent. from no load to full load. On each pump there is a Ford regulating valve placed between the throttle and turbine. At present the feed water is delivered to the boiler at 165 degrees Fahrenheit.

The steam leads from each boiler are 7 inches in diameter and connect with 12- and 14-inch headers located in front of each battery. From these headers 10- and 12-inch lines run to two Cochrane receiver separators, and from one of these separators a 12-inch main supplies steam to the engine. The other separator is blanked for a future unit. A 4-inch auxiliary steam line connected to the boilers furnishes steam direct to the dry-air pump, the four fan engines, and the two turbine-driven boiler-feed pumps. The valves in all the large mains have bypass connections.

Mechanical-draft Apparatus

The mechanical-draft apparatus consists of two forced-draft fans and two induced-draft fans capable of handling a total of 4000 horsepower when burning anthracite culm.

Each forced-draft fan is 9 feet in diameter and is direct connected to a 9x8-inch vertical engine. Fig. 4 shows one of these fans located near one of the boiler-feed pumps and the Cochrane heater. The capacities of these fans, based on the burning of coal at the rate of six pounds per horsepower-hour, are as follows:

Cubic feet of air per minute	31,410	41,200	54,300	62,700
Revolutions per minute	106	149	183	211
Brake horsepower	6.5	12.5	23.3	35.1
Dynamic pressure	0.43 inch	0.85 inch	1.3 inches	1.7 inches

The speed of the fans is controlled by the draft requirements through the medium of a Spencer damper regulator connected to a Fitts chronometer-regulating valve located near the throttle in the steam-supply pipe.

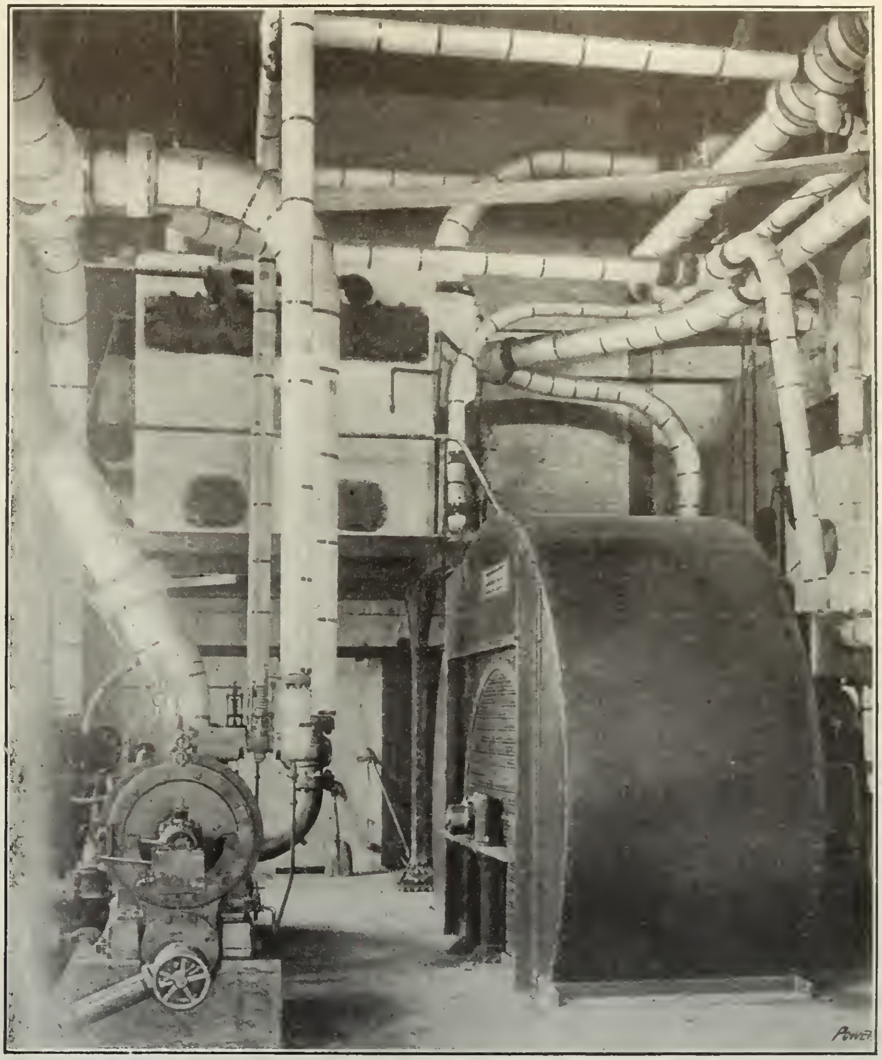


Fig. 4. Heater, Feed Pump and Forced-draft Unit

The induced-draft fans each have wheels 11 feet in diameter and are of the following capacities, based upon a frictional resistance of 62½ per cent. of the free discharge, with the temperature of the escaping gases 550 degrees Fahrenheit. These fans are located on an elevated structure—see Fig. 3—between the boiler settings and are driven by vertical slide-valve engines, the speed of which is automatically controlled by a special regulator to meet the draft requirements:

The capacities of these blowers are as follows:

Cubic feet of air per minute	83,376	94,560	105,600
Revolutions per minute	163	186	208
Brake horsepower	19.7	33.6	41.8
Dynamic pressure	0.75 inch	1 inch	1.25 inches

Engine and Piping

At present there is but one engine, but provision is made for the installation of a similar one at some future time. This is a 2400-horsepower 30 and 62 by 54-inch Rice & Sargent cross-compound condensing engine running at 100 revolutions per minute and direct connected to the generator. The low-pressure cutoff is so designed that it can be either automatically controlled by the action of the governor on the high-pressure cylinder, or it may be changed to a fixed cutoff.

The main steam pipe to the engine is 12 inches in diameter and the exhaust pipe from the low-pressure cylinder to the condenser is 30 inches in diameter; in the latter there is a gate valve operated by an electric motor. From this pipe a 20-inch pipe runs outside the building and up to the roof where it is topped with a

20-inch Sterling free-exhaust head. In the 20-inch line is placed an automatic free-exhaust valve, so that the exhaust may be bypassed around the condenser in case of derangement of the latter. The low-pressure drips are all trapped into the river.

All the high-pressure piping for steam and water is made of extra-strong, lap-welded soft steel. Corrugated-copper gaskets are used in all joints of pipes, 2½ inches and over.

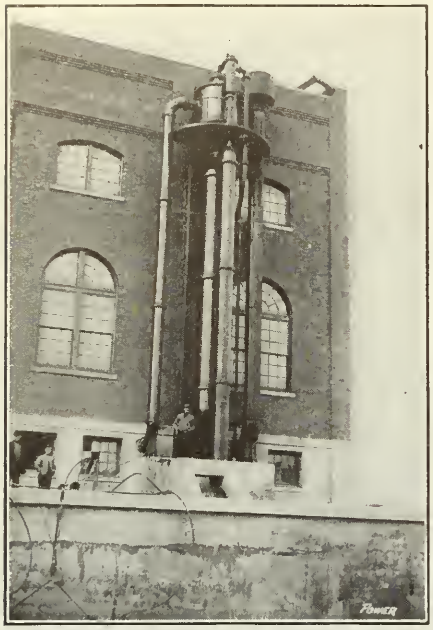


FIG. 5. BAROMETRIC CONDENSER

OILING SYSTEM

The engine is supplied with oil by a continuous-flow gravity system; the oil being forced from the filters, located in the basement, up into elevated tanks by air pressure obtained from the dry-air pump.

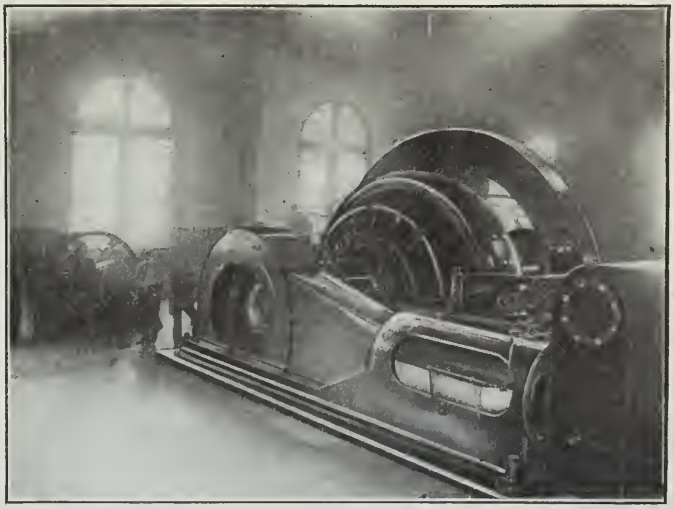


FIG. 6. ENGINE AND GENERATOR

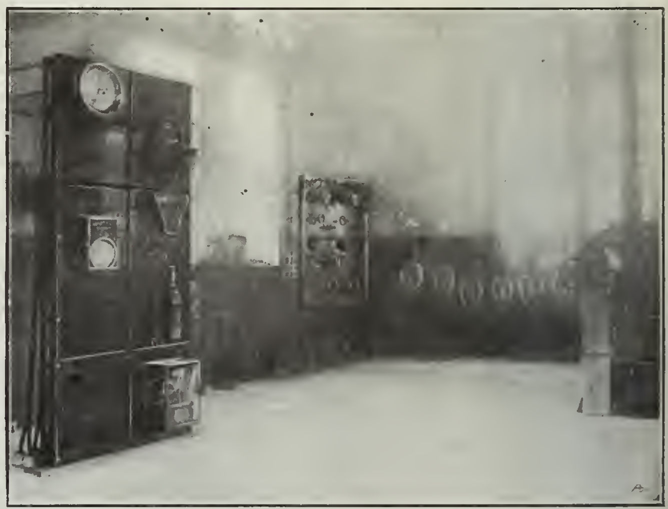


FIG. 7. MAIN SWITCHBOARD AND CONTROL PANEL

THE GENERATOR

The generator is of the General Electric direct-current, compound-wound type, of 1600 kilowatts capacity and delivers current at 575 volts when running at 100 revolutions per minute. The efficiency at full load is 95.7 per cent.; at three-quarters load, 94.5 per cent.; at half load, 92.5 per cent., and at 50 per cent. overload, 95 per cent. One hundred per cent. overload may be carried momentarily without injurious sparking. Fig. 6 shows the generator and part of the engine.

The switchboard, as shown in Fig. 7, is of simple construction, consisting of two panels, fitted with a clock, ammeter, voltmeter, circuit-breaker, wattmeter, and main switch. The control switchboard for the motors operating the auxiliaries is attached to the wall at the corner of the room.

BAROMETRIC CONDENSER

The condenser is of the Helander type and is shown in Fig. 5. The inlet for the exhaust steam is 30 inches in diameter, and the cooling-water inlet is 12 inches. The diameter of the tail pipe is 18 inches and there is a 6-inch connection for the air-pump suction. The capacity is 48,000 pounds of steam per hour, with the cooling water at 80 degrees Fahrenheit, using 3600 gallons of cooling water per minute and maintaining a 27-inch vacuum. The condenser is made of special acid-proof metal, and is guaranteed to give at least three years' service when using the acid water from the Lackawanna river.

DRY-AIR PUMP

The dry-air pump is of the crank and flywheel type, made by the Mesta Machine company, and has a capacity of 550 cubic feet per minute. The steam is supplied at boiler pressure, and the exhaust is led to the Cochrane heater.

CIRCULATING PUMPS

There are two 12-inch double-suction centrifugal, volute, circulating pumps, made by the Jeanesville Iron Works. These are of the horizontal-shaft type with extended bedplates, direct-connected through flexible couplings to 75-horsepower electric motors. They have a capacity of 4000 gallons per minute, with a head ranging from 60 feet at starting to 40 feet while running. The total head referred to includes suction, discharge and friction lifts, based on measurements from the center line of the pumps.

There is a strainer at the bottom of each suction pipe as well as a nonreturn valve made of acid-resisting bronze. The 12-inch discharge pipes from the pumps connect into a main 16-inch discharge.

OPERATION

The operating force consists of one chief engineer in charge of the two stations, two watch engineers, two oilers, a cleaner, three firemen, three firemen's helpers, an ash man and a man to look after the blowers, making a total of thirteen men, exclusive of the chief. The station is operated only from 7 a.m. till midnight, the old plant carrying the load during the early hours of the morning.

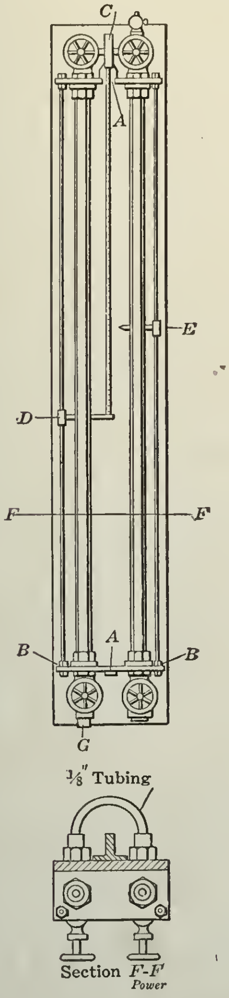
The induced- and forced-draft systems are used simultaneously. Four boilers are in use all the time, leaving one cut out for cleaning and overhauling. Culm is used entirely for fuel, 35 to 40 tons being consumed per 24 hours, and the fires are cleaned but once a day. The ashes are removed by hand at the furnace fronts and are pushed away in steel dump cars.

At present, the culm is hauled in horse carts from a nearby railroad siding and is dumped on the ground under the trestle; from here it is picked up by a Sprague one-ton clam-shell lift, and is conveyed into the boiler room and dumped into a bunker located over the aisle between the boilers and extending the entire length of the boiler room. The culm is dumped onto the boiler-room floor through swinging chutes in which are installed Havard coal meters. Eventually the culm will be brought direct to the trestle by cars from the railroad siding. The culm is of the poorest quality, unwashed and dirty, just as taken from the dump at the mine, and costs but little more than that required for the hauling and handling. A freshet in the river sometimes deposits considerable culm on the banks near the boiler room; this is carried into the fire room and fed to the furnaces at a cost of the laborer's wages only.

The results of a 10-hour efficiency test, conducted at about the time the plant was placed in service, showed 2.73 pounds of culm to be consumed per horsepower-hour and a steam consumption of 15.33 pounds of water per horsepower-hour (including both engine and auxiliaries). The boiler efficiency ranged from 55 to 61 per cent.

Mercury Gage from Stock Fittings

A mercury gage which can be put together from stock fittings, and which possesses several advantages, was recently assembled by William A. Warman, mechanical engineer of the Keeler Mechanical Engraving Company, of New York.



Brass being out of the question, two sets of malleable-iron ammonia gage-glass fittings were procured and mounted upon a plate of 1/4-inch iron 48 inches in length and 6 inches wide, with a stiffening rib of 1 1/2-inch T-iron riveted to its back as shown in section *F F* of the outline drawing.

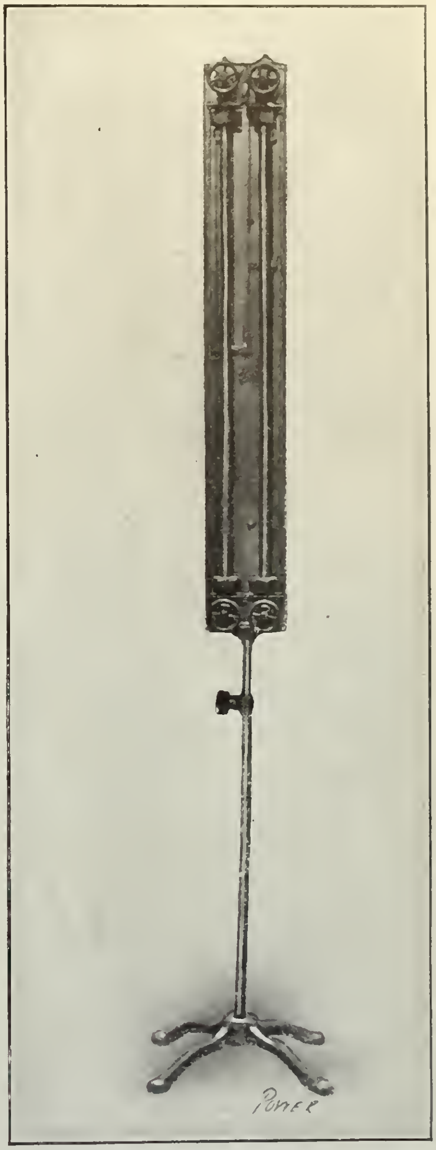
The plates for supporting the guard rods are fastened together at *A A*, and threaded rods with nuts at *B B* are used in place of the ordinary guard rods to resist any tendency to separate under pressure.

The smallest fittings procurable were for 3/4-inch glasses, and, in order to avoid the use of the large quantity of mercury which it would have taken to fill these, blued-steel rods 5/16 inch in diameter were run centrally through the glass tubes, leaving only the annular spaces between the rods and the glass to be filled by the mercury. The contrast between the dark rod and the bright mercury makes the gage easy to read.

Upon the top mounting at *C* is fastened the spring case of a pocket steel tape which is returned to the case by a spring when a button upon the side is pressed. This button is permanently held down so that the tension of the spring is always upon the tape. The free end of the tape is attached to a slide *D* upon the left-hand guide rod, which slide carries an indicator the point of which is in line with the zero of the tape and touches the left-hand gage glass. Another slide *E* upon the right-hand rod carries an arm which passes behind the gears and has an index reaching to the tape.

The slide *D* is moved so that its pointer is at the level of the mercury in the left-hand tube, and the slide *E* is adjusted so that its index is at the level of the mercury in the right-hand tube. The top index will then point directly to the difference in the mercury levels upon the tape.

A pet cock is provided at the top for filling the tubes and a plug at *G* for drawing off the mercury. The gage is mounted upon an adjustable iron stand as shown in the reproduced photograph.



Nothing but stock fixtures and material were used and the cost exclusive of labor did not exceed $14.

It is related that when the Massachusetts license law went into effect there was among the applicants for a fireman's license an old man who for years had been employed in one of the suburbs of Boston cooking garbage which was used for feeding hogs. Replying to the varied questions of the examiner he gave a description of the plant, method of feeding by city water pressure, etc. On being asked what steam pressure was carried he said with a rich brogue, "Tin pounds, Sorr." "Now," said the examiner, "you carry ten pounds pressure to do your work. Suppose that while you were busy and not noticing what was going on the steam pressure should run up to 50 pounds, what would you do?"

With eyes wide with astonishment at the display on the part of the examiner of such abysmal depths of ignorance of the art of swill boiling, and trembling with indignation at the thought that such carelessness could be even mentioned in connection with him, he hastily and hotly said:

"Oh, Sorr! The likes of ye shud know that it wud never do at all, it wud blow the schwill all to hell."

Air Supply and Boiler Efficiency

By A. B. McIntyre

Method of figuring furnace temperature due to the complete combustion of anthracite coal under ideal conditions and when the air supply is equal to that theoretically required. Loss due to incomplete combustion consequent to an insufficient air supply. Loss due to excess air.

While this article will treat of the conditions affecting economy in burning coal in the boiler furnace, little or no attention will be paid to the design or arrangement of the furnace itself. The location and general arrangement of all baffles and heating surfaces have a direct and important bearing on the efficiency obtained, but this is a means to secure the desired result and requires a different treatment than the conditions conducive to economic combustion.

Only the better grades of anthracite will be considered here, as a complete discussion of all grades of coal would make this article of too great length. Then too, a similar discussion covering the bituminous and semi-bituminous coals is more complex and difficult to comprehend, so that a rational method of treatment would seem to be a discussion of anthracite first.

For the greatest efficiency in the operation of the steam boiler, the flue gases must be maintained at the highest possible temperatures; provided, of course, that these temperatures do not reach values unduly destructive to the surfaces exposed to the heated gases. A moment's reflection will show why this is true. With intelligent firing the flue gases will be discharged to the stack at a fairly constant temperature. All heat contained in these gases is lost as far as the boiler is concerned. The temperature of these gases should be reduced by the heating surfaces to at least 600 degrees Fahrenheit, which is high, 500 degrees or less being considered good operation where no economizers are used. The gases will be reduced to approximately this temperature irrespective of the furnace temperature; hence, the higher the furnace temperature, the greater the drop in temperature, and therefore the greater amount of heat absorbed by the boiler.

In order to obtain the fullest advantage of the heat value of the fuel it is evident that complete combustion must take place. This demands a sufficient supply of oxygen to unite with the entire combustible content. A further supply of oxygen is useless as far as combustion is concerned, and, in that it is discharged at stack temperature, represents a direct loss of heat.

TABLE 1.* PROXIMATE ANALYSIS OF ANTHRACITE COALS.

Designation.	Moisture.	Fixed Carbon.	Volatile Matter.	Ash.	Sulphur.	B.t.u. per Pound Coal.
1 Northern coalfields	3.42	83.27	4.38	8.20	0.73	13,160
2 East middle	3.71	86.40	3.08	6.22	0.58	13,420
3 West	3.16	81.59	3.72	10.65	0.50	12,840
4 Southern	3.09	83.81	4.28	8.18	0.64	13,220

Before taking up the general discussion the constituents of several of the better grades of anthracite will be examined. Table 1* shows the results of a proximate analysis of several such coals. When first fired, except for a slight temporary chilling of the gases, the moisture has little effect upon the burning qualities but represents a loss inasmuch as the heat required for its evaporation and superheating to stack temperature is not available to the boiler. For the coals under consideration this loss is quite small, as will best be seen by example.

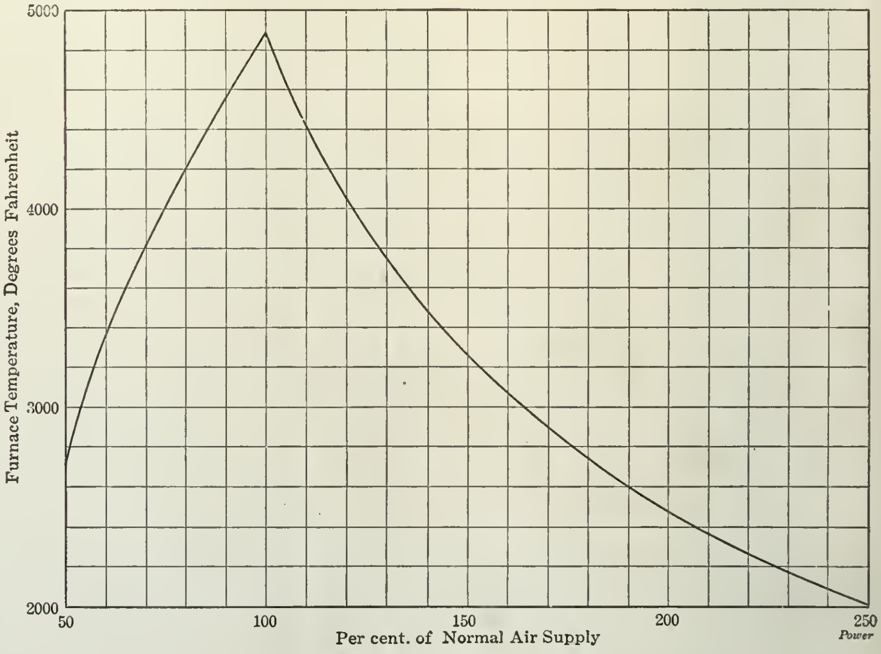


Fig. 1. Relation between Air Supply and Furnace Temperature

Take sample No. 2, containing 3.71 per cent. moisture. Assume the air temperature to be 50 degrees and the stack gas temperature, 500 degrees. The B.t.u. required to raise the water to 212 degrees are,

$$0.0371\ (212 - 50) = 6.01$$

The B.t.u. required for the latent heat of evaporation are,

$$0.0371 \times 970.4 = 36$$

The B.t.u. required for superheating to 500 degrees are,

$$0.0371 \times 0.48\dagger\ (500 - 212) = 5.13$$

The total is 47.14 B.t.u. per pound or about 0.34 per cent., which is well within the limits of other errors involved in this class of work.

In a similar way the ash represents a loss to the ashpit. The sulphur content is very small and its comparative heating value negligible. Its presence is very undesirable, however, as it produces rapid deterioration of the iron work and also mixes with the ash to form a fusible slag which chokes up the air spaces in the grates and is difficult to remove. The percentage of volatile matter in anthracite is very low. In fact, it is the percentage of volatile matter that determines whether a coal belongs to the anthracite or bituminous class. The lower limit in anthracite is generally taken at 7½ per cent. Unlike that of the carbon content, the heat value of a given quantity of volatile matter is not constant and may vary, roughly, anywhere from one-third below to one-third above that of carbon. The burning of the small amount of volatile matter present in anthracite does not present any difficulties, so its discussion will be left for a similar article on bituminous coal. Practically all of the heat value of a coal depends upon its carbon and volatile-matter content. As noted above, the percentage of volatile matter contained in anthracite is very low, and for the purposes of this article will be considered to have a heat value equivalent to carbon in the calculations of the furnace-gas temperatures. The error thus introduced will not be much over 1 per cent. in the majority of cases.

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*From Kent's "Steam Boiler Economy."

†Specific heat of steam.

In its combustion carbon changes to an invisible gas. If a limited supply of oxygen is present, carbon monoxide (CO) will be formed; if a surplus exists, the carbon will all pass off as carbon dioxide (CO_2). The burning of one pound of carbon to CO liberates 4450 B.t.u.; while if complete combustion takes place, that is, if the carbon changes to CO_2, 14,500 heat units will be set free. Part of this heat is lost by radiation and the remainder is accounted for by the increased temperature of the gases involved in combustion. The amount lost by radiation is, or should be, comparatively small and will be neglected in the calculations. The following give the chemical reactions for the combustion of carbon, and also the combining weights:

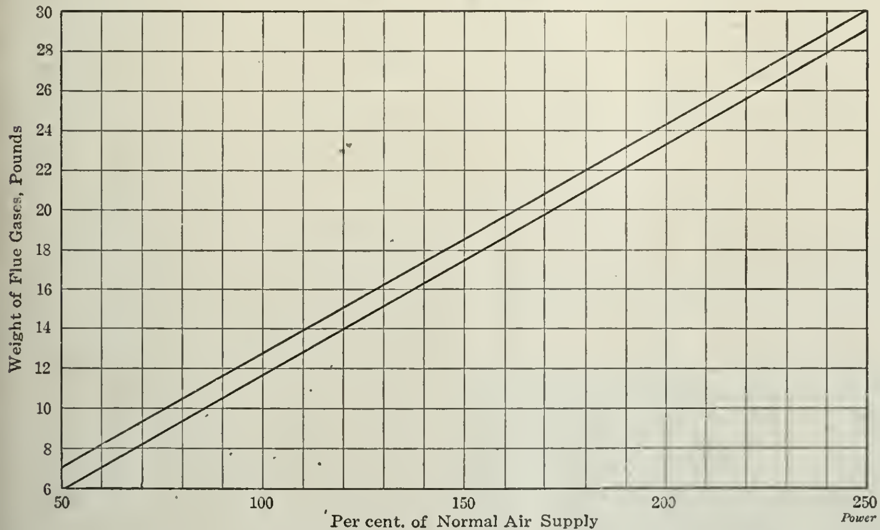


FIG. 2. RELATION BETWEEN AIR SUPPLY AND WEIGHT OF FLUE GASES

Element	C.	+	O	=	CO
Relative weight	12	+	16	=	28
Proportion	1	+	$1\frac{1}{3}$	=	$2\frac{1}{3}$
Element	CO	+	O	=	CO_2
Relative weight	28	+	16	=	44
Proportion	1	+	0.571	=	1.571
Element	C	+	2 O	=	CO_2
Relative weight	12	+	32	=	44
Proportion	1	+	$2\frac{2}{3}$	=	$3\frac{2}{3}$

By taking carbon as unit weight it is found that one pound of carbon unites with 1.33 pounds of oxygen to form 2.33 pounds of CO, and one pound of carbon unites with 2.67 pounds of oxygen to form 3.67 pounds of CO_2.

The maximum temperature obtained by burning one pound of carbon in pure oxygen will now be calculated. The temperature above 0 degrees Fahrenheit of the supporter of combustion will in all cases be neglected. The average temperature will be probably not far from 50 degrees and the effect of this upon the resulting temperatures may well be neglected as it will introduce but very small error. Table 2 gives the specific heat at constant pressure of the several gases involved in the calculations.

TABLE 2. SPECIFIC HEAT OF GASES AT CONSTANT PRESSURE.

Substance.	Specific Heat.
Air	0.238
Carbon monoxide (CO)	0.248
Carbon dioxide (CO_2)	0.217
Oxygen (O)	0.218
Nitrogen (N)	0.244
Chimney gas (approx.)	0.240
Steam (Regnault)	0.480

These specific heats are not constant quantities, but vary more or less with the temperature. In the case of air it is quite likely that the specific heat reaches a value as high as 0.29 for the temperatures obtained in the boiler furnace.

The burning of one pound of carbon to CO requires 1.33 pounds of oxygen, produces 2.33 pounds of CO, and evolves 4450 B.t.u. The maximum temperature reached will be

$$\frac{4450}{2.33 \times 0.248} = 7700 \textit{ degrees Fahrenheit}$$

Had this same pound of carbon been burned to CO_2 the temperature would have been

$$\frac{14{,}500}{3.66 \times 0.217} = 18{,}260 \textit{ degrees Fahrenheit}$$

As noted, these temperatures result from the use of pure oxygen and have been calculated to obtain figures for comparison with those obtained when air is used as a supporter of combustion.

In boiler practice air is used as the supporter of combustion, and the temperatures reached when carbon is burned in air will now be calculated. As far as practical combustion is concerned, air is composed of 23 parts of oxygen and 77 parts of nitrogen by weight.‡ Each pound of air contains, then, 0.23 pound of oxygen and 0.77 pound of nitrogen. The nitrogen takes no part in combustion but serves only to dilute the flue gases, thus lowering the resultant temperature. If a cheap supply of undiluted oxygen could be obtained, and if the heating surfaces could be designed to withstand the greatly increased temperatures, the operating efficiency of the steam boiler could be materially increased by using pure oxygen instead of air for combustion. Every pound of oxygen in the air carries with it,

$$0.77 \div 0.23 = 3.348$$

pounds of nitrogen.

The 1.33 pounds and 2.67 pounds of oxygen required to burn the carbon to CO and CO_2 will carry with them,

$$1.33 \times 3.348 = 4.46$$

and

$$2.67 \times 3.348 = 8.94$$

pounds of nitrogen respectively. The maximum temperatures will be, then, for CO,

$$\frac{4450}{(2.33 \times 0.248) + (4.46 \times 0.244)} =$$

$$2670 \textit{ degrees Fahrenheit}$$

and for CO_2,

$$\frac{14{,}500}{(3.67 \times 0.217) + (8.94 \times 0.244)} =$$

$$4880 \textit{ degrees Fahrenheit}$$

By referring to the temperatures obtained when pure oxygen is used, it is seen that these temperatures are about 35 and 27 per cent., respectively, of their former values. The temperature, 4880 degrees, is the maximum temperature attainable in the furnace, and, of course, is never realized in practice due to incomplete combustion, radiation, excess air and other reasons.

If only the exact amount of air required for combustion is admitted, it is necessary that every particle of oxygen be brought into intimate contact with every particle of the fuel. This is manifestly impossible in the practical boiler furnace, and for this reason a surplus of air must be admitted in order to secure fairly complete combustion. This surplus air must all be heated to the furnace temperature, thus reducing the temperatures given above. The quantity admitted varies considerably with the type of firebox and the care and watchfulness of the fireman. Good practice requires about 50 per cent. or a little less; but plants are in operation where the excess air runs to 100 per cent. and in many cases probably much higher.

The reduction in temperature caused by this excess of air can easily be computed. Take, for example, the case of 50 per cent. excess air.

Amount of heat liberated	14,500 B.t.u.
Pounds of CO_2 formed	3.67
Pounds of nitrogen	8.94
Pounds of excess air	5.8

The maximum temperature will be

---

‡Air also contains small quantities of water vapor, carbon dioxide, dust particles and ammonia, but these have comparatively no effect upon combustion.

$$\frac{14{,}500}{(5.8 \times 0.238) + (8.94 \times 0.244) + (3.67 \times 0.217)} = 3330 \text{ degrees}$$

This, though nearly one-third less than the temperature attained with normal air supply, is the maximum toward which good operation tends. This is still further reduced by the causes mentioned above, and it is quite probable that 2600 degrees is the highest average temperature reached by the flue gases.

In trying to avoid the evils of excess air, care must be taken not to make the greater error of not admitting sufficient. The effect upon the maximum temperature of a 40 per cent. deficient or a 60 per cent. normal air supply will now be considered. There will be but

$$11.6 \times 0.60 = 6.96$$

pounds of air present. Of this,

$$6.96 \times 0.23 = 1.6$$

pounds are oxygen, and

$$6.96 \times 0.77 = 5.36$$

pounds are nitrogen. In burning to CO a pound of carbon requires 1.33 pounds of oxygen and will produce 2.33 pounds of CO. There is left, then, but

$$1.6 - 1.33 = 0.27$$

pound of oxygen for further union with the combustible. Part of the CO will change to CO_2. Each pound of the CO requires for its further combustion 0.571 pound of oxygen. There is but 0.27 pound of oxygen remaining, and, therefore, but

$$0.27 \div 0.571 = 0.473$$

pound of the CO can be burned to CO_2, leaving

$$2.33 - 0.473 = 1.857$$

pounds of CO still unconsumed. The 2.33 pounds of CO burning to CO_2 evolves

$$14{,}500 - 4450 = 10{,}050 \textit{ B.t.u.},$$

or 4310 per pound. The amount of heat produced by this partial combustion is, by one pound of carbon to CO, 4450 and by 0.473 pound of CO to CO_2, 2030, or a total of 6480 B.t.u. Thus, considerably over half of the total heat is lost in the unconsumed 1.857 pounds of CO by incomplete combustion. The weights of the constituents of the flue gas are:

$$\begin{aligned} N &= 5.360 \text{ pounds} \\ CO &= 1.857 \text{ pounds} \\ CO_2 &= 0.742 \text{ pound} \\ \hline \text{Total, } &7.959 \text{ pounds} \end{aligned}$$

The specific heat of this mixture is:

$$\begin{aligned} 5.360 \times 0.244 &= 1.31 \\ 1.857 \times 0.248 &= 0.461 \\ 0.742 \times 0.217 &= 0.161 \\ \hline 7.959 \qquad &\quad 1.932 \end{aligned}$$

$$1.932 \div 7.959 = 0.242$$

the specific heat required. The maximum temperature will be

$$\frac{6480}{7.959 + 0.242} = 3360$$

degrees Fahrenheit. Thus a lower temperature obtains with a 40 per cent. deficient air supply than with a 50 per cent. excess.

It is quite evident from the foregoing that there is a great loss in the available heat when the air supply is deficient. This is manifest not only by the loss of the combustible gas, carbon monoxide, but also by the greatly reduced temperature of the products of combustion and the diminished boiler efficiency resulting therefrom. To show more clearly what effect a varying air supply has upon the temperature of the flue gas, the curve in Fig. 1 has been plotted. This shows at a glance the maximum temperature of the flue gas as the air supply is varied from 50 per cent. of normal to 250 per cent. of normal, where the term normal is intended to mean that amount of air carrying just sufficient oxygen to produce complete combustion. The points on the curve have been calculated in the manner just described, except that the specific heat of the flue gas has been taken as 0.24 for all cases instead of calculating it independently for the several conditions where the composition varies.

Upon examining this curve it is seen that for equal percentage values of excess or deficient air supply there are no great differences in the flue-gas temperatures over a considerable range from normal. This, at first thought, would seem to indicate that, for a reasonable variation from normal, just as efficient operation could be obtained with deficient air supply as with an excess, since with deficient air the loss to the stack will be less than with excess. But with excess air the heat absorbed by the boiler will be greater, due to the increased weight of the gases. To obtain a definite conclusion, the total weight of the flue gases with reference to the air supply has been plotted in Fig. 2. From this the actual weight of the flue gases can be obtained for any desired percentage of air supply. A concrete case will now be taken and the actual quantity of heat passing to the boiler will be calculated. No account will be taken of the loss due to radiation and ash, not that it can always be considered negligible, but because this loss is practically constant throughout the working range, and, therefore, will not affect the comparison. The temperature of the gases discharged to the stack is assumed to be 500 degrees. From Figs. 1 and 2, the temperature and weight of the flue gases with 120 per cent. air supply are found to be 4050 degrees and 14.92 pounds respectively. The boiler will absorb, then,

$$14.92 \times 0.24\ (4050 - 500) = 12{,}710 \textit{ B.t.u.}$$

The same percentage of deficiency, or 80 per cent. air supply, will now be considered. From Figs. 1 and 2 the temperature and weight are found to be 4260 degrees and 10.28 pounds respectively. The heat passing to the boiler will be

$$10.28 \times 0.24\ (4270 - 500) = 9280 \textit{ B.t.u.}$$

Thus it is seen that, although the temperature of the flue gas is actually higher with 20 per cent. deficient air supply, the increased weight with excess air more than compensates for this in spite of the greater loss to the stack. A sufficient number of these values have been calculated to plot the curve shown in Fig. 3. The general shape of this curve does not differ from that in Fig. 1. Fig. 3 shows conclusively what effect the varying weight of the gases has upon the efficiency.

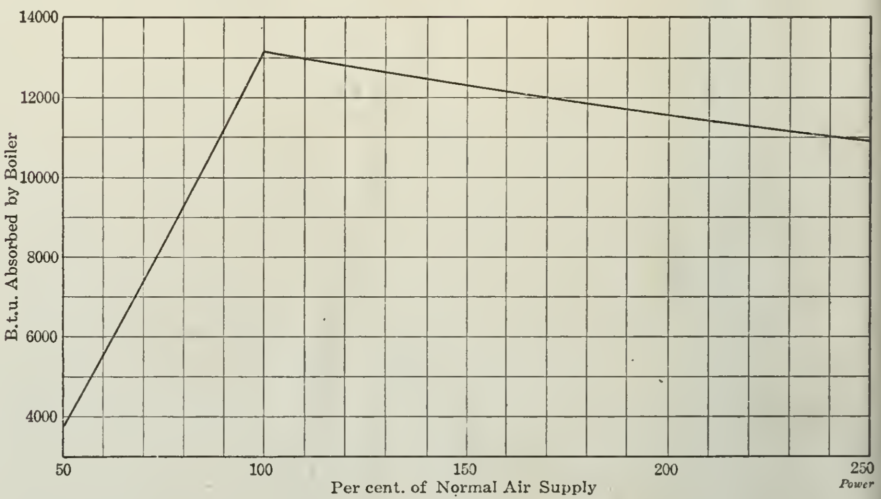


Fig. 3. Relation between Air Supply and Heat Absorbed by Boiler

As a rough check upon the work, the thermal efficiency of the boiler will be obtained from the information contained in Fig. 3. Take, for example, a boiler operating under good average conditions; that is, with 50 per cent. excess or 150 per cent. normal air supply. From the curve it is found that the boiler will absorb 12,280 B.t.u. per pound of carbon burned. This gives a thermal efficiency of

$$\frac{12{,}280}{14{,}500} = 84.69 \textit{ per cent.}$$

This is in close accordance with results actually obtained in practice when the assumptions made and the losses neglected in these calculations are considered.

A few deductions may now be made from what has preceded. The air supply must be closely watched to see that a sufficient but yet not excessive amount is admitted. This is most readily determined by some sort of automatic recording apparatus that determines the percentage of carbon dioxide in the stack gases. Several of these devices are now on the market and their use seems to show a decided improvement in the economy of the steam-generating end of the power plant. No smoke will be formed in any case, so that it will be impossible to judge the relative economies of combustion by watching its products.

When fired, anthracite kindles rather slowly and does not immediately give off large volumes of gas as is the case with the softer coals. Thus, very little chilling effect is produced and no increase in the air supply is required. With regular, even firing no change in the air supply is necessary except so far as it depends upon the call for steam.

The fact that the carbon dioxide is formed relatively close to the fuel allows the heating surfaces of the boiler to be situated comparatively near the fuel bed, thus somewhat reducing the cost of the installation. This also obviates the necessity of using external furnaces, heat-radiating surfaces and other gas-consuming devices, which must generally be used in the case of the volatile coals.

The principal argument against the use of anthracite is its high first cost, although this is partially offset by the greater ease with which it can be handled, and, with average firing, the better realization of its available heat.

Novel Arrangement of Valve Gear

A twin engine made up of two types of cylinder and valve gear. One governor controls both cylinders, the two being operated at high pressure.

At the Butler street power plant of the Atlanta Railway and Power Company, Atlanta, Ga., there is a peculiar arrangement of the two steam cylinders and valve gear of a double engine. The illustration shows the details. The unit consists of a 19x48-inch Greene-Wheelock engine cylinder on one side and a Ranken-Fritsch Corliss engine cylinder on the opposite side of what is now a Wheelock-Corliss double engine.

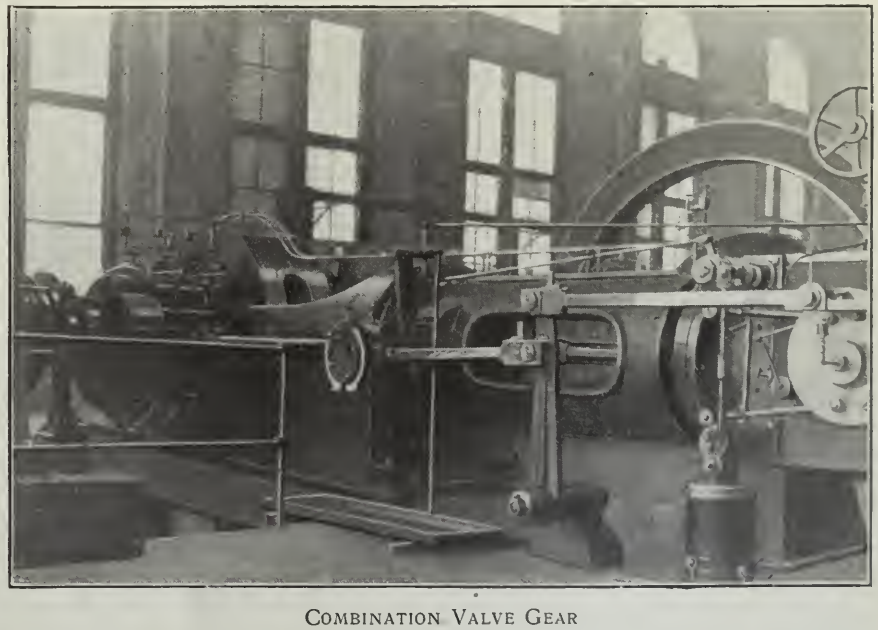
The original Wheelock engine had 19 and 44 by 48-inch cylinders, and the original Corliss unit was a tandem engine. When the plant was turned over to be used mainly as a heating plant, the exhaust steam being used to heat many of the business blocks and hotels of the city, it was decided to change the Corliss engine into a Greene-Wheelock unit by removing the Corliss cylinders and substituting the low-pressure cylinder of the original cross-compound Wheelock engine. The low-pressure cylinder of the Corliss engine was discarded, and the high-pressure cylinder fitted to the frame of the low-pressure side of the Wheelock engine.

The present arrangement allows the engine cylinders to act as reducing valves between the boilers and the heating main, and the power obtained in passing the steam through the engines is a substantial gain over what would be obtained with live steam used for heating purposes passed through a reducing valve, direct from the boilers. The combination cylinders now drive a 525-kilowatt generator which is used for railway service in connection with another of the same capacity. The two units are connected in series, the current going to the switchboard at about 550 volts. The unit has given better satisfaction than would naturally be expected from such a combination.

The particular point of interest lies in the peculiar arrangement of the valve gear of the Corliss side of the combination engine. This valve gear is driven by the original gearing of the Wheelock engine, and in order to do this, the original cam shaft was cut off at a length that would permit of securing a suitable bearing on the frame of the engine next to the section containing the guide. A casting was made to fit the engine frame, and of such design as to admit of arranging a suitable bearing for the end of the original cam shaft, also for a short eccentric shaft set at right angles to the cam shaft and driven by a set of bevel gears. On the end of this short shaft an eccentric was secured, which was fitted with eccentric straps. An eccentric rod was connected with the rocker arm, pivoted at the bottom end, as shown, its supporting base being bolted to the bedplate of the engine. From this rocker arm the valve gear of the Corliss side is operated in the usual manner.

The object of using the gear work of the old Wheelock engine was to eliminate the offset in either the reach rod or the eccentric rod, which would have been necessary had a split eccentric been put on the crank shaft in place of the original gear.

The same governor, which is on the Wheelock side of the engine, governs both types of valve gear. An extension link drops down from the governor and connects with a rod running under the flooring between the two cylinders; to this rod a vertical rod extends up to the Corliss governor cam rod, as shown, midway of the eccentric rod, and operates the cutoff cams in the usual manner. Both cylinders are operated at high pressure.



Combination Valve Gear

Selecting and Using Pyrometers

By C. Conam

The thermocouple and resistance types of pyrometer are considered. The latter are extremely accurate but are handicapped by a limited range and high first cost. For general work the thermocouple has proved to be more satisfactory.

In considering the installation of pyrometers for the control of furnace temperatures, in a works where a number of instruments are to be used, one of the chief points to be studied is the type of instrument. For it goes almost without saying that it is necessary to use, as far as possible, one type of instrument so that interchangeability is permitted. There are many other points needing close attention when this decision has been made, but at first this one stands well to the fore, and must be settled.

The two kinds of pyrometers that will attract attention, and which fulfil many of the requirements satisfactorily, are the thermocouple and resistance types. Much may be said for—and against—both these types of pyrometers, but in the end the thermocouple proves to be for general work the most suitable and satisfactory instrument.

The resistance pyrometer is very accurate in its indications, but the range of temperatures through which it acts is, as a rule, very much more limited than that of the thermocouple. This is very important, and must weigh in the balance seriously against the former type of instrument. Then again the high first cost of the resistance pyrometer, compared with the thermocouple, is another serious point. This is particularly noticeable when recording instruments are required, and in this case the controlling mechanism is very much more complicated than in the thermocouple, and requires the attention of a skilled man for regulation and adjustment.

Errors in Thermocouple Instruments

There are many points about thermocouples which require careful attention, and if neglected may lead to serious errors and losses. The first point that comes to my mind is that the internal resistance of couples and leads may vary with couples made from the same standard wires. This variation may be due to differences in the lengths of the couples and the leads, and the result is that all the couples will not indicate the same temperatures when their hot junctions are side by side in a furnace. This error does not exist if the internal resistance of the indicating millivoltmeter is very high, but as the indicators generally used by engineers for pyrometrical work have a resistance of about 100 ohms only a variation of, say, two ohms in the couples will be an appreciable percentage of the total resistance of the circuit, and will consequently show an error of that amount in the temperature indicated. It is quite possible to get an error of from 30 to 40 degrees Fahrenheit in this way.

There are two methods of overcoming this difficulty. The first is to increase the resistance of the indicator to such an amount that any normal variation, due to difference in the length of couples and leads, is made entirely negligible; and the second is to bring all the leads to one standard resistance by inserting coils of fine wire, and all the couples to a standard resistance by the same method. With the couples, the coil may be conveniently placed in the cold junction box.

This second method is open to the objection that the cold junction coils are liable to be raised considerably above their normal temperature by reason of their close proximity to furnace walls, etc., and so have their resistance increased. This increase, however, even if the cold junction box is raised by 80 degrees Fahrenheit—and with proper care it would never reach a point that much above the normal—is so very slight that the error is not appreciable and can be safely neglected.

Protecting Covers

Thermocouples adjusted as above to the same standard resistance, side by side in a furnace, with cold junctions at one temperature, and everything equal but the type of sheathing or protecting cover, will not even then show the same temperatures. As all other faults have been eliminated, the cover is the only possible cause of the error, and it is a fact that different kinds of covers create errors sometimes amounting to 20 or 30 degrees Fahrenheit.

It is absolutely necessary that all sheathing for thermocouples should be as far as possible of the same nature. For ordinary furnace work all that is necessary is an iron tube, solid-welded at one end, and this kind of cover finds the most general use. In special cases such as for use in the lead bath, or with high-temperature metallic salts baths, this form of sheathing affords no protection against the penetrating action of vapors and gases, which rapidly destroy the couple wires.

In cases of this description special kinds of sheathing are necessary, and one that may be safely used consists of an outer tube of steam piping, solid welded, containing a smaller solid-welded tube heavily plated on the outside, which, in its turn, contains the couple. This sort of cover can be used for a very long time with the simple reversal of the outer sheath when necessary.

Porcelain Quartz and Fireclay Tubes

Many other forms of protecting tubes are used, including porcelain quartz and fireclay. Porcelain does not find much favor as it is liable to crack at high temperatures. Quartz tubes are used to a moderate extent, and are valuable because they are comparatively rigid even when hot, and may be subjected to violent changes of temperature without cracking; such as being plunged into molten metal when cold. They are, however, only impermeable to gases at temperatures not exceeding 1800 degrees Fahrenheit, and even at that temperature should be used intermittently as their impermeability breaks down with constant use.

These tubes rapidly rise to the temperature of the furnace, and consequently indicate changes of temperature very readily. Marquardt tubes will stand a very high temperature without losing their impermeability to gases, but they are of little practical value as they will not stand rapid changes of temperature; nor must they be put into a position where there is likelihood of their being knocked or affected by vibration, as they are very fragile, and liable to crack. They, moreover, have to be placed generally in a vertical position, as they readily bend at high heats, and if placed horizontally may break of their own weight. They can be used under favorable conditions up to 2800 degrees Fahrenheit. In all cases where these protecting tubes are used it is usual to give them the extra protection of a sheath of wrought-iron tubing.

Why Protection Is Necessary

It is necessary to protect the cold junction of a couple as much as possible from the action of the hot furnace gases and from the radiation of the furnace. It is obvious that the position of the couple will have a good deal to do with the former action. It is better to insert the couple through a small aperture in the side of the furnace, than through one in the roof. In all cases a fairly good protection, and one that will allow the cold junction to be kept at a fairly low temperature, consists of a sheet of asbestos millboard with the couple passing through the middle. This, if the millboard is large enough, deflects the hot gases, and arrests the radiant heat effectively. In cases where the cold-junction temperature must be kept very uniform the cold-junction box has a water jacket through which water continually flows from the mains. This is quite unnecessary in works practice.

The insulation from one another of the couple wires is a simple matter, as porcelain tubes, double drilled, about 3 inches long, and ¼ inch in diameter, can be purchased from most wholesale chemists. Small balls, double bored, are also used for this purpose, especially when the couple sheath is of a special shape, say, right-angular. Small-diameter quartz tubes are also made for this purpose. It is very important to see that, in building up a thermocouple, the wires are not twisted and touching between each insulating tube, for, if they do touch, a false junction is thereby made and the indicator readings will be wrong.

The Testing of Cooling Towers

By Wilfred Yorke

> *From some practical tests of large natural-draft towers the author shows how the necessary data are obtained and calculated, and also points out how more efficient service could have been obtained.*

The writer had occasion recently to test some large natural-draft cooling towers to ascertain if the towers were cooling the specified quantity of water per hour, and also to discover the amount of water lost through evaporation. Though the underlying principles are quite well known, the methods by which the results were arrived at are quite novel, and may prove interesting to many engineers.

The towers were installed for cooling the circulating water from a plant containing one 4000-kilowatt reciprocating set, and two 6000-kilowatt turbine-driven sets, all fitted with surface condensers. As the amount of circulating water utilized by these three units was about 15 million pounds per hour, the impracticability of measuring this quantity mechanically for a comparatively short test period, will be apparent.

TABLE 1. OBSERVATIONS ON CONDENSERS.

Generator.	Load, Kilowatts.	Vacuum, Inches.	Circulating Water, Degrees Fahrenheit.		Exhaust Temperature, Degrees F.	Wet Air-pump Suction, Degrees F.
			Inlet.	Outlet.		
Engine	3,500	25	85	113	133	90
Turbine A	4,000	27	86	99	114	106
Turbine B	5,600	27.3	86	100	112	98

Engine, steam consumption = 18.5 pounds per kilowatt-hour.
Turbine A, steam consumption = 18.5 pounds per kilowatt-hour.
Turbine B, steam consumption = 14.5 pounds per kilowatt-hour.

By working on a heat basis, however, a very close approximation can be obtained, sufficiently accurate for the purpose of most commercial tests, and it was along this line that the inquiries in question were conducted.

Estimate of Total Amount of Cooling Water

On each of the three condensers the following observations were made:

The inlet and outlet temperatures of the cooling water.

The temperature of the exhaust steam as it entered the condensers.

The temperature of the condensed steam, i.e., the wet air-pump suction.

The vacuum in inches of mercury at the engine exhaust pipe.

Simultaneous readings on the switchboard were taken of the load each generator was carrying, and then knowing the steam consumption per kilowatt-hour of each generating set from previous steam trials it became an easy matter to figure out the quantity of cooling water passing through each condenser per hour. An example will make this clear. See Table 1 for these observations.

In the case of the reciprocating engine the quantity of heat given up to the cooling water by one pound of steam is the difference between the amount it contained on entering the condenser as exhaust steam, and the amount taken away by the condensed steam.

Neglecting conduction, etc., if,

H = Total heat given up by one pound of steam,
T' = Temperature in degrees Fahrenheit of exhaust steam.
T'' = Temperature in degrees Fahrenheit of condensed steam.

Then,

$$H = 970.4 - 0.7\ (T' - 212) + (T' - T'') = 1068.7 \text{ B.t.u.}$$

The factor $0.7\ (T' - 212)$ being derived from the fact that while the latent heat of steam at 212 degrees is 970.4, to evaporate it at one degree higher would require 0.7 B.t.u. less.

Now, by dividing this quantity by the rise in temperature of the cooling water the number of pounds of cooling water required for the condensation of one pound of steam can be ascertained,

$$1068.7 \div (113 - 85) = 38.$$

Then the total weight of cooling water per hour required to condense the steam from the engine equals

$$38 \times 3500 \times 18.5 = 2{,}450{,}000.$$

In a similar manner the quantity of cooling water used on turbine A will be found to equal 5,300,000 pounds, and on turbine B, 6,300,000 pounds per hour.

Engine = 2,450,000 pounds per hour.
Turbine A = 5,300,000 pounds per hour.
Turbine B = 6,300,000 pounds per hour.

Total = 14,050,000 pounds per hour.

Percentage Lost by Evaporation

While the foregoing set of observations was being taken on the condensers, other readings were simultaneously taken on the towers, the results being shown in Table 2. These were:

Dry- and wet-bulb thermometers of an hygrometer.

Temperature of the air inside the towers close to the distributing troughs.

Temperature of the cooling water as it entered the towers.

Temperature of the water in the catch pond below the towers.

TABLE 2. OBSERVATIONS ON COOLING TOWERS.

External Air, Degrees Fahrenheit.		Relative Humidity, Per Cent.	Internal Air, Degrees Fahrenheit.	Circulating Water, Degrees Fahrenheit.	
Dry Bulb.	Wet Bulb.			Inlet at Top.	Temperature of Catch Pond.
55	51.5	90	95	104	86

If we imagine 1 cubic foot of air passing through the towers it will enter at a temperature of 55 degrees Fahrenheit with a humidity of 90 per cent., and will emerge at the top in a saturated condition at a temperature of 95 degrees. One cubic foot of dry air at 55 degrees Fahrenheit can absorb 4.85 grains (Table 3, column 5) of aqueous vapor;

therefore a humidity of 90 per cent. will represent 4.37 grains present in 1 cubic foot of external air.

In traveling up the towers the original cubic foot of air will expand 0.002 for each degree rise in temperature, and will therefore have a volume of 1.08 cubic feet when it emerges as saturated air at a temperature of 95 degrees. From Table 3, column 5, it may be seen that 1 cubic foot of saturated air at 95 degrees Fahrenheit contains 17.2 grains of equeous vapor; then 1.08 cubic feet will take away 18.6 grains. But 4.37 grains are already present in the air at its entrance; hence,

$$18.6 - 4.37 = 14.23$$

grains are evaporated by each cubic foot of air in passing through the towers.

Further, 1 cubic foot of air will take up heat from the cooling water in three ways:

Heat required to warm up 1 cubic foot of dry air from 55 degrees to 95 degrees Fahrenheit.

Heat required to warm up aqueous vapor already present (4.37 grains) from 55 to 95 degrees Fahrenheit.

Heat required to cvaporate 14.23 grains of water at an inlet temperature of 104 degrees into aqueous vapor at 95 degrees Fahrenheit.

These three heat values are found as follows:

One cubic foot of dry air at 55 degrees weighs 0.077 pound. (Table 3, column 9. Therefore, the heat required to warm 1 cubic foot of dry air under the existing conditions is

$$0.077 \times \mathit{sp.\ heat} \times \mathit{rise\ in\ temperature} = 0.077 \times 0.238 \times 40 = 0.73\ B.t.u.$$

One pound (7000 grains) of aqueous vapor requires

$$40 \times 0.48\ B.t.u.$$

Therefore, 4.37 grains would require

$$(4.37 \div 7000) \times 0.48 \times 40 = 0.012\ B.t.u.$$

which is the heat required to warm the vapor present from 55 degrees to 95 degrees.

The heat required to evaporate one pound (7000 grains) of water under these conditions is

$$970.4 - 0.7\,(T_2 - 212) - (T_1 - T_2) = 970.4 - 0.7\,(95 - 212) - (104 - 95) = 1043\ B.t.u.$$

where T_1 is the temperature of the water and T_2 the temperature of the aqueous vapor.

Therefore, 14.23 grains require

$$(14.23 \div 7000) \times 1043 = 2.11\ B.t.u.$$

which is the heat required to evaporate 14.23 grains of water at 104 degrees into aqueous vapor at 95 degrees.

TABLE 3. PROPERTIES OF AIR AT 29.92 INCHES OF MERCURY.

Temperature Degrees F.	Pressure of Aqueous Vapor, Inches of Mercury.	One Cubic Foot of Saturated Air.					Volume of Dry Air at Different Temperatures, Vol. @ 32° = 1.	Weight of Dry Air at Different Temperatures, Pounds.
		Pressure of the Air in Mixture, Lb.	Weight in Pounds.	Wt. of Aqueous Vapor.		Total Weight of Mixture, Pounds.		
				Grains.	Pounds.			
32	0.18	29.74	0.08020	2.15	0.00031	0.08051	1.000	0.0807
35	0.21	29.71	0.07970	2.40	0.00034	0.08004	1.006	0.0802
40	0.25	29.67	0.07879	2.86	0.00041	0.07920	1.016	0.0795
45	0.30	29.62	0.07785	3.4	0.00049	0.07834	1.026	0.0787
50	0.36	29.56	0.07694	4.04	0.00058	0 07752	1.037	0.0778
55	0.42	29.50	0.07619	4.85	0.00069	0.07688	1.047	0.0772
60	0.53	29.39	0.07506	5.78	0.00083	0.07589	1.057	0.0764
65	0.64	29.28	0.07410	6.8	0.00097	0.07507	1.067	0.0757
70	0.75	29.17	0.07311	8.0	0.00114	0.07425	1.078	0.0749
75	0.87	29.05	0 07208	9.35	0.00134	0.07342	1.088	0.0742
80	1.03	28.89	0.07105	10.99	0.00157	0.07262	1.098	0.0735
85	1.20	28.72	0.06996	12.7	0.00182	0.07178	1.108	0.0728
90	1.35	28.57	0.06896	14.78	0.00212	0.07108	1.118	0.0722
95	1.65	28.27	0.06763	17.2	0.00246	0.07009	1.128	0.0716
100	1.93	27.99	0.06641	19.8	0.00283	0.06924	1.139	0.0708

Then the heat absorbed by one cubic foot of air is made up as follows:

Heat to warm air.......... = 0.73 B.t.u. = 25.5 per cent.
Heat to warm vapor....... = 0.012 B.t.u. = 0.5 per cent.
Heat to evaporate cooling water....... = 2.11 B.t.u. = 74.0 per cent.
Total....... = 2.852 B.t.u. = 100 per cent.

That is, 74 per cent. of the total amount of heat dissipated in the towers evaporates part of the cooling water, and it is now but a step to calculate how much water is thus lost.

We have seen that 14,050,000 pounds of water pass over the towers per hour, and this weight of water in cooling from 104 degrees to 86 degrees will liberate,

$$14{,}050{,}000 \times (104 - 86) = 253{,}000{,}000\ B.t.u.$$

74 per cent. of this quantity, or 187,000,000 B.t.u., is responsible for evaporating the cooling water.

But we have already shown that 1043 B.t.u. is the heat necessary to evaporate one pound of water at 104 degrees into aqueous vapor at 95 degrees Fahrenheit; therefore,

$$187{,}000{,}000 \div 1043 = 180{,}000$$

pounds of water evaporated per hour, which equals 1.28 per cent. of the total weight of cooling water.

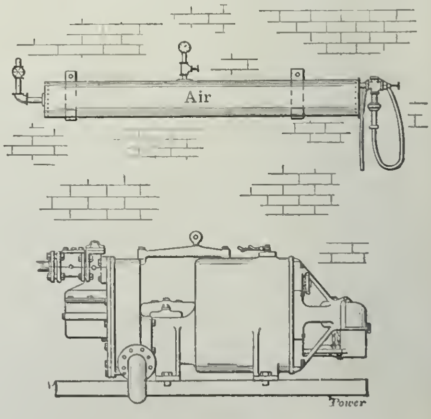
From this test it was deduced that of the total heat dissipated in these towers approximately 75 per cent. was responsible for evaporating part of the cooling water to the extent of 1.28 per cent. It can also be figured that the water so lost through evaporation is also equal to about 85 per cent. of the total steam consumption of the three generating sets, which is a very fair average. A cooling tower in which the evaporation loss outweighs the weight of steam condensed would be regarded as inefficient.

A study of Table 3 will reveal that the main factor in the evaporation loss is the exit temperature of the air, which is, of course, dependent upon the temperature of the cooling water as it enters the towers.

Compressed Air Blowing Outfit

The accompanying illustration shows the method used by an engineer in Richmond, Va., for obtaining an air supply for blowing out dust from the generator end of a small Curtis turbine set.

In his plant was a small belt-driven air compressor, which furnished compressed air for removing shavings from the wood-working machinery in the factory to the boiler furnaces. In this case a small 30-gallon water tank was obtained which was capable of withstanding an air pressure of 70 pounds to the square inch. This air tank was secured to the wall of the engine room directly over the turbine unit, by means of two hangers made of ⅛-inch strap iron, curved so as to form a cradle for the tank, as shown. At one end of the tank was fitted a nipple to which was screwed the air pipe running to the compressor, this pipe being fitted with a valve to permit shutting off the tank from the air-pump line. On the other end of the tank, at the top, was fitted a long nipple to which an angle valve was screwed. To this valve was screwed another long nipple to which a hose coupling was attached, the hose being long enough to reach to all parts of the turbine as well as to the switchboard. The tank was inexpensive, neat in appearance, being painted with aluminum, and efficient in operation.



ARRANGEMENT OF AIR TANK

Stream Flow at Single Cross Section

By Franklin Van Winkle

It is often difficult to obtain appropriate conditions for measuring stream flow by the float method. The advantages of estimating the discharge based on average velocity at a single well-defined cross-section have been recognized and instruments have been perfected to measure the velocity at different points in the given section. Current pressure gages, such as the pressure plate, hydrometric pendulum and pitot tube, also velocity meters and float wheels are used. All of these instruments are given attention.

The difficulty which is frequently experienced of obtaining appropriate conditions for measuring stream flow by the float method, and an appreciation of advantages of estimating discharge based on average velocity at a single, well defined cross-section, has led to employment of various devices for this purpose. In the use of these instruments the mean velocity of the cross-section is obtained by application of the instrument to different points in the given section. All instruments of this kind require to be rated, i.e., their error must be obtained for different depths and velocities, either while holding them stationary in currents of water of known velocity, or by moving them at known velocities through still water, and in computing the mean velocity the errors of instruments so found must be duly allowed for. Apparatus of this kind may be divided into two general classes:

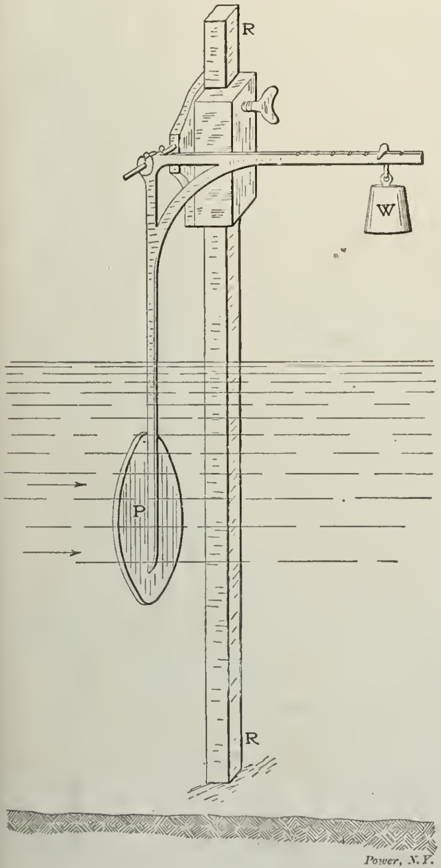


FIG. 1. PRESSURE PLATE

(1) Instruments in which velocity of current is inferred from pressure which the current exerts at their points of application, and which may, therefore, be designated as current-pressure gages.

(2) Instruments which receive and register the movement of currents and which are known under the names of wheel meters, velocimeters, but more generally under the designation "current meters."

Those of the first class, coming under the designation of current-pressure measuring instruments, are intended to measure head or pressure due to velocity for interpretation of velocity of current in feet per second from the formula

$$v = \sqrt{2gh}.$$

The forms of current-pressure instruments most commonly referred to are the pressure plate, the hydrometric pendulum and the pitot tube. The pressure plate as an instrument for determination of stream velocity is said to have been invented by Gaunthy in 1779. It consists of a metallic disk *P*, Fig. 1, which is set to oppose the pressure exerted by the current, velocity being computed from the weight *W* necessary for holding the disk *P* vertical. The instrument was swung from the side of a rod *R R*, as shown in the figure, or suspended by suitable frame so as to take readings at different depths. Various modifications of the pressure plate have been employed in important gagings. The hydrodynamometer of M. Perrodil is one of these, consisting, as shown in Fig. 2, of a pressure plate *P* acting around a vertical axis which is supported by the vertical rod *M N* and framework *A B C D*. The vertical axis consists of a torsion bar *U* suspended from the upper end by a collar *K* and guided in bearings *J* and fastened to the frame *A B C D* at the bottom. The pressure plate *P* being connected to the lower end of the torsion bar by the arm *R*, rotates the whole framework *A B C D* when released by a set screw *S*.

In the upper part *E F* is a horizontal divided circle carried by the frame, and *H* is a pointer hand fastened to the upper end of the torsion rod. Suppose the apparatus to be so arranged that the pressure plate *P* is at such level of the stream as it may be desired to determine the velocity. By loosening the set screw *S* the skeleton framework *A B C D* is permitted to adjust itself in a vertical plane parallel to the direction of the current by the pressure plate *P* acting like a vane or rudder, carrying the divided circle *F F* with it so the pointer hand indicates zero on the scale. If then the instrument is turned by pressing the pointer sideways until the plane of the frame *A B C D* and of the pressure plate and of the zero of the graduated circle is at right angles to the stream, the torsion rod will be twisted through an angle which measures the normal impulse of the stream on the pressure plate *P*. The set screw *S* is then tightened and the average position of the pointer is read for obtaining average velocity. Careful experiments have been made with this instrument, using various sizes of pressure plates and of torsion rods, and the results have been very satisfactory, especially in measurement of currents which were too slow to be measured by current meters or pitot tubes. Rating of the instrument has been performed by fixing it to a radius bar or turn table sweeping over a circular course in still water a distance of about 75 feet in circumference.

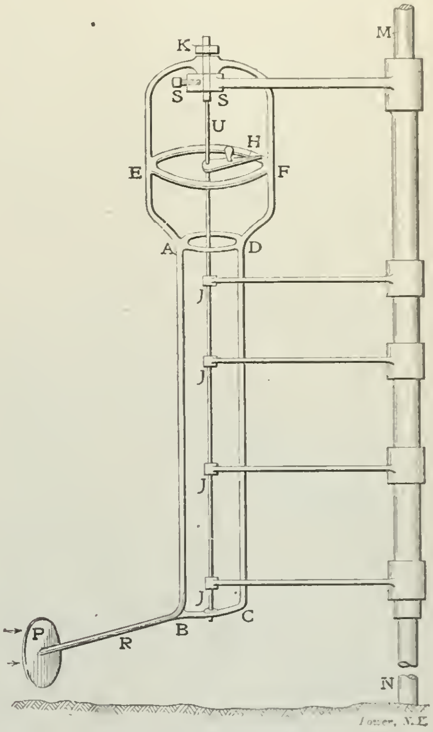


FIG. 2. M. PERRODIL'S HYDRODYNAMOMETER

Castelli's quadrant or hydrometric pendulum is another instrument of the first class mentioned, and appears to have been devised by Castelli for measuring velocities of streams in the early part of the seventeenth century (about 1628). It consists of a ball suspended by a thread from the center of a graduated arc; the instrument when used being placed in the current with the arc vertical and parallel with the direction of the flow, horizontal pressure and velocity are determined from the weight of the ball and the angle formed between the suspending string and a vertical line on the arc. This apparatus has the merit of simplicity, but in the computation of velocity, pressure of the current on the suspending cord and lifting or "boiling" action which is always present in running water, renders employment of this apparatus of very little practical value. The method is suggestive, however, that some use might be made of it for approximate determinations of velocity, as in times of freshet, for determining the velocity of a stream by suspending a plummet from a bridge. But in such cases the velocities could only be estimated, empirically, after determining the deflections of suspending cords which are obtained with the same size, shape, weight, depths of submergence and lengths of suspending cord when used in currents of known velocities.

THE PITOT TUBE

The pitot tube is the simplest, and in its improved form is undoubtedly the most reliable instrument for determination of velocity of a current from indication of its pressure. It was invented or said to have been used for this purpose by Pitot in 1730. It consists simply of a vertical glass tube with a right-angle bend placed so that the mouth of one leg of the tube will be presented at right angles to the direction of flow, the impulse of the current of water being balanced by a column of water raised in the other leg of the tube above the general level of the stream. Pitot, for his use, enlarged the mouth of the tube to a funnel or bell shape, as shown in Fig. 3. This, however, causes the liquid to rise a hight h, which is about 1½ times the true hight or head due to the velocity. This form of entrance is additionally objectionable because it interferes with the current, and the velocity in front of the mouth is not the same as the velocity of the unobstructed stream.

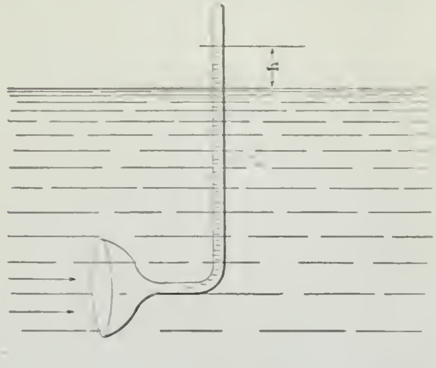


FIG. 3. PITOT TUBE WITH BELL MOUTH

Fig. 4 shows the improved form of orifice employed by Darcy and Bazin, in which the tube is drawn out very small with the great benefits of interfering little if any with the natural velocity of the stream; and also that the reduced size of opening tends to check oscillations of the column of water in the tube, instead of encouraging them as is the case of a tube provided with a bell-shaped mouthpiece In the drawn-out form shown in Fig. 4, Darcy found that when it was placed as shown at P_1, the hight h was almost exactly

$$h = \frac{v^2}{2g};$$

when placed as shown at P_2, having the plane of the orifice parallel to the direction of flow, the water rose practically level with the surface of the stream; and when turned with the mouth down stream like P_3, the water sinks to a depth h', which is nearly the same amount as the rise in case it is headed up stream like P_1. When however, a tube is turned down stream like P_3 the partial vacuum created tends to release air bubbles, accompanied by a surging of the hight h', so that small dependence can be placed on combined readings like R; i.e., for determination of velocity it is not proper to assume that the velocity due to the head can be taken as

$$\frac{h + h'}{2}.$$

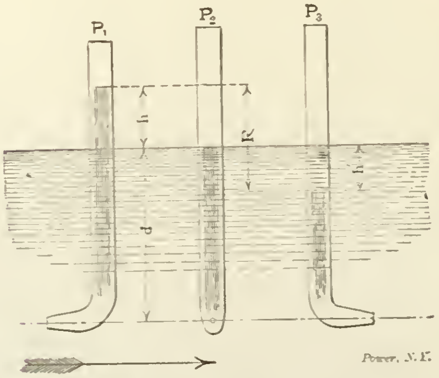


FIG. 4. IMPROVED FORM OF TUBE EMPLOYED BY DARCY AND BAZIN

An objection to employing a simple tube like Fig. 4 is the difficulty of reading the hight h direct from the surface of the stream. This is overcome in Darcy's improved form of pitot tube, by means of which readings can be made above the surface of the stream, or the instrument may be entirely removed from the water for that purpose.

Fig. 5 illustrates the leading features of the Darcy instrument. Two pitot tubes, *HE* and *JG*, made of copper, have openings at right angles to each other. In making velocity measurements the instrument is so held that the opening in the end of the tube *HE* is presented against the current while that of *JG* is downward. The space between the tubes is filled with a solid piece of wood or metal for strengthening the tubes and holding them in place. The upper ends of the tubes are made of glass mounted on a wooden support *W W*, which in turn is supported by the clamp *Q* and the guide bracket *K* which surround the standard *B B*. Each glass tube is provided with a vernier *C* for reading the hight of the columns on a vertical scale *S*. By means of the handle *A* and hook *L* the instrument can be raised or lowered to any desired depth and held at the desired elevation by adjustment of the clamp *Q*, and when thus supported the main body of the instrument acting as a rudder, swings itself around the standard *B B* to a position parallel with the cur-

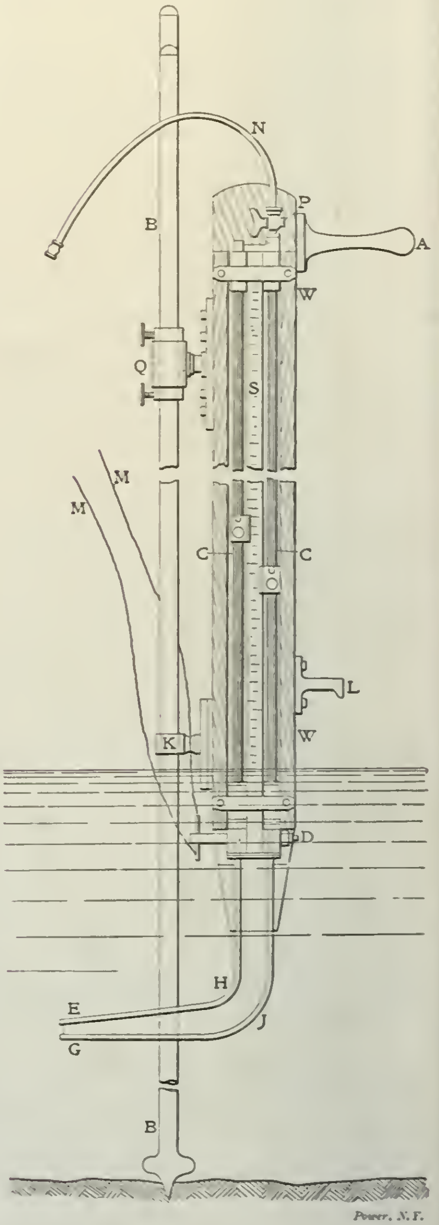


FIG. 5. DARCY'S IMPROVED FORM OF PITOT TUBE

rent and presents the opening of the tube *E H* up stream.

Connection of the copper tubes to the glass tubes is made through a two-way cock *D* which can be operated by cords *M M*. The glass tubes are connected at their upper ends by a brass fitting which is provided with a stop cock *P*, the outlet of which is provided with a piece of flexible tubing *N* with a mouthpiece. Having adjusted the instrument to the desired depth, with the cocks *D* and *P* open, water then rises in the tube *E H* to greater or less extent above the surface of the stream, while it rises in the tube *J G* to the same level as that of the stream. If then a little air be sucked out of tube *N* and the cock *P* closed, water will rise in both glass tubes an amount equal to their respective differences from atmospheric pressure and will stand with the same relative difference between their levels as they first had in the lower part of the instrument before any air was exhausted. The cock *D* is then closed, preserving the relative hight of the columns, and the difference is easily read off with the instrument in place, or by removing it from the stream.

In using this instrument for obtaining the mean velocity of a stream, a number of readings have to be taken to obtain average velocity at any point, as the velocity of a stream is continually changing at every point of its cross-section; and for determining the mean velocity of a section, the mean of averages of different points of the cross-section must be taken. But the mean velocity of a stream is quickly found by one accustomed to using the instrument, once the cross-section of the stream is established. The principal drawbacks to its employment are, that it is not suitable for use in streams where there is floating grass or weeds, nor for measuring very slow velocities. A difficulty which should be guarded against is the liability of obtaining too small readings of the column which is connected with the tube *G J*, due to dirt gathering in the opening at *G* and so changing the form of its orifice as to create a sucking action, as in the tube P_3, Fig. 4. This defect can be guarded against by occasional examination of the instrument before exhausting air by the tube *N*. Using the instrument in clear water rarely gives much trouble, but in all work of importance the instrument should be rated in water of known velocity to obtain its mean variation of readings from the formula,

$$h = \frac{v^2}{2g}.$$

FLOAT WHEELS

Wheels driven by currents of natural streams of water were among the first if not the earliest forms of water motors. In their simpler forms they consist of wheels mounted on horizontal axes and provided, as shown in Fig. 6, with radial floats or vanes *F F*, rotation of the wheel resulting from vanes on the lower side of the wheel dipping down into the water and partaking of the motion of the moving current.

When little or no resistance is offered to rotation of a wheel of this kind, the mean velocity of submerged portions of the floats is approximately the same as the mean velocity of the cross-section of as much of the current as may be intercepted by the floats, provided the radial dimension of the submerged portion of the float is only a small proportion of its distance from the axis of the wheel.

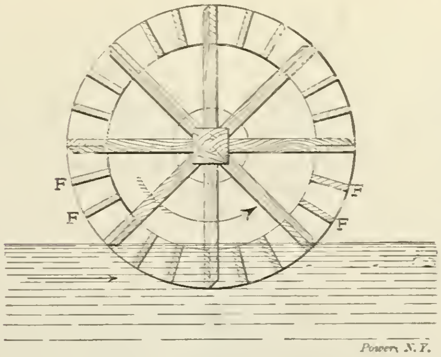


Fig. 6. Float Wheel

Light portable float wheels, having connected clockworks for recording their revolutions on dials, have been used for measuring velocities of open streams, but are impracticable for measuring more than surface velocities, and when so used have to be "rated" for ascertaining their error at different velocities, and are not efficient instruments for obtaining the mean velocity of a stream. Special applications have been made of float wheels, however, in which the wheels filled the whole breadth of channel through which the flow was directed, and, with the floats extending to the bottom of the channel, the spaces between the floats acted as measures of volume. Water wheels used in this manner for metering water have been called "gage wheels."

A very elaborate gage wheel of this kind was employed in 1841 by three skillful engineers, Messrs. Baldwin, Whistler and Storrow, to determine the quantities of water drawn from the canals of the Proprietors of Locks and Canals on Merrimac river by the several manufacturing companies of Lowell, Mass., and is worthy of mention. In this instance measurement was made of the flow which took place through a canal which was 29 feet wide and about 8 feet deep, in the manner generally illustrated in Fig. 7. For the purpose of adapting the channel of the canal to this method of measurement, the canal was broadened out by excavating a basin *A A A A* to a width of about 80 feet and the bottom *B B* was raised so as to leave a depth of water of about 4½ feet. Across this basin they placed seven paddle wheels each 16 feet in diameter by 10 feet long with shafts of the wheels coupled together, the shafts being supported in bearings which rested on solid but narrow piers *P*.

The wheels were made with great care and were accurately fitted so as to run within about ¼ inch of the apron or floor *F* and the piers at the sides of the wheels.

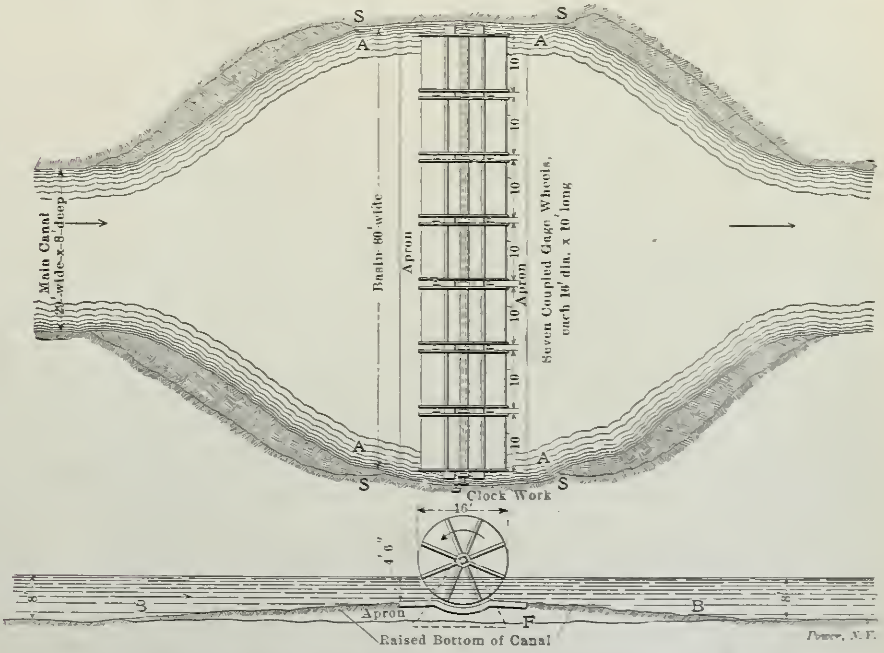


Fig. 7. Elaborate Gage Wheel Used to Measure Flow of Canal

Sheet piling *S S S S* was driven across the heads of the apron, requiring the whole flow of the canal to pass between the piers and drive the wheels. The apron *F* was formed of timbers cut to the sweep of the wheels, being of sufficient length in direction of the current for one float to enter the curved portion of the apron on the up-stream side before the preceding float had left the apron on the down-stream side. The body of water between two floats and the apron, which they called a "bucket full," was thus cut off from the rest and passed over the apron by itself, and as the wheels revolved the whole flow thus passed in successive buckets full.

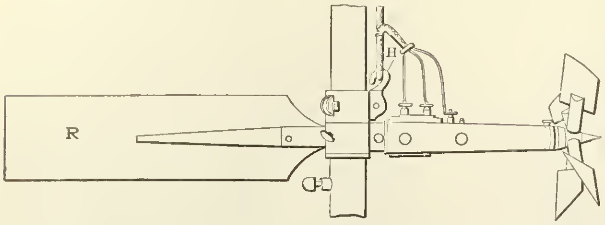


Fig. 8. Current Meter of Medium Size

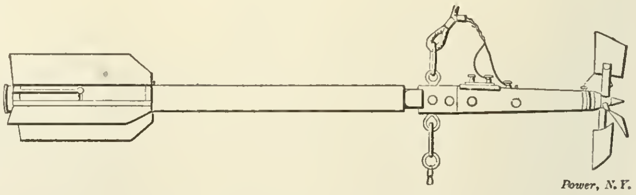


Fig. 9. Same Meter Suspended

The floats were equally spaced, and knowing all the dimensions of the wheels, the quantity contained in one bucket full was readily obtained by computation. A clock work connected to one end of the shaft showed on a dial the number of rotations and fractions of rotations made per minute, so that knowing the number of buckets full which were delivered and discharged, the quantity of water flowing over the apron in a given time was readily determined. The principal sources of error in use of such an apparatus would be from leakage through and around the apron construction and by water passing through the clearance spaces which were left for the wheel to turn in; but as the engineers in their report do not appear to have taken any such errors into account they must have been satisfied that errors of measurement from these causes were small enough to be neglected.

The quantities of water which they reported to have thus measured ranged from 405 to 595 cubic feet per second, and the mean velocities of currents discharged through different branch canals were found to vary from 81 to 85 per cent. of their surface velocities, which is of interest in showing the variation between surface and mean velocity of current, the surface velocities in these cases ranging from about 2 feet to about 3½ feet per second.

Current Meters

For determining the mean velocities of open streams at given natural cross-sections, current meters are undoubtedly the most convenient and accurate instruments that can be employed. The earlier forms of these instruments were defective in many details of design which have been remedied in the later forms, so that under most circumstances measurements of open streams can be made with the improved forms of current meter with as great accuracy as could be desired for most practical purposes, and when conditions are equally favorable, with as great accuracy as ordinarily obtainable by weir measurements.

The first of these instruments consisted of revolving fans or helicoidal blades mounted like a propeller on a horizontal shaft on which there was a screw, the latter being meshed with a worm wheel connected to wheel work for registering the number of revolutions made by the propeller shaft. The instrument was set at different depths by clamping it to a rod, the lower end of which was held firmly on the bottom of the stream, and the submerged registering mechanism was thrown in or out of gear by pulling a cord or wire. This arrangement required the whole apparatus to be lifted out of the water in order to obtain readings and exposed the registering mechanism to grit and dirt carried by the water, resulting in uncertainty of operation of the instrument. This was later modified by gearing the main worm wheel to a light shaft extended above the surface of the water and there connected to the registering apparatus, but this arrangement was found awkward for use at variable depths, and the shaft connections gave rise to so much frictional resistance as to render the use of the instrument very unreliable. These difficulties are overcome in the later forms of propeller wheel instruments, in which the wheel work connected to the propeller shaft is reduced to the simplest form necessary for making and breaking electrical connection with a buzzer, an electrical recorder or a registering apparatus conveniently located above water for indicating the revolutions of the propeller, and very excellent instruments of this kind are now produced by various manufacturers.

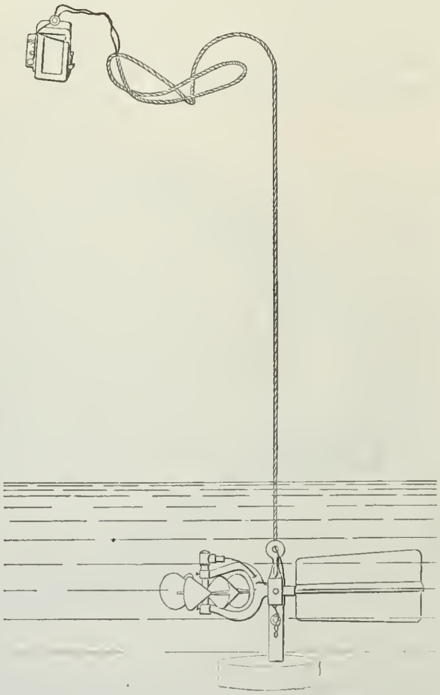


Fig. 10. Price Current Meter Provided with Weight and Rudder

Fig. 8 illustrates a medium-sized current meter made by the Keuffel & Esser Company. The propeller axis of this instrument is mounted on ball and agate bearings and electrical contacts are arranged for a single and for every twenty revolutions of the propeller, the axis, worm gear and contacts being protected by inclosure in a metallic torpedo-shaped case. The instrument is provided with a metallic rudder *R* which is about 4½x12 inches. The main framework fits loosely around a pole 1 inch in diameter, to which it can be clamped for holding the instrument at various depths; or by means of a hook attached to the framework, the instrument may be raised or lowered by sliding it up or down on the pole by means of a suspending rope passed over a pulley secured to the upper end of the pole. For using this current meter at greater depths than would be practicable to use a supporting pole in this manner, the torpedo-shaped body of the instrument can be unscrewed and attached to a large metal rudder, as shown in Fig. 9, by which it can be suspended by a line and anchored at any desired depth, thus forming a submerged floating-current meter. Two forms of Price current meters, as manufactured by W. & L. E. Gurley, are illustrated in Figs. 10 and 11. The main difference in their construction and operation from ordinary forms of current meters is that motion is imparted to a short vertical shaft by five conical buckets arranged on the ends of arms which rotate in a horizontal plane, like the cups of a vertical-shaft wind gage. An advantage of this form of propulsion is that the instrument can be employed for measurement of velocity of currents very close to the bottom and also that positive motion can be obtained with slower velocity of currents than with instruments fitted with helicoidal blades for rotation of horizontal axes. Another advantage claimed for instruments of this type is that they revolve at slower speed for a given velocity of current and their revolutions are therefore more easily counted. Instruments of this kind are, however, more liable to derangement from débris floating in the water and require to be more carefully handled than the common form of current meters, but for measurements of velocities in shallow streams or where the currents have very slow velocities, the cup form of current meter appears to have decided advantage over the other. In the Price meters here illustrated, the shaft bearings are in air chambers which afford protection from the water and any gritty matter it may contain. The upper end of the shaft extends above its bearing and is cut down a short distance, forming an eccentric. A light spring is so arranged as to come in contact with the eccentric to successively make and break contact at each revolution of the wheel. Fig. 10 shows a Price electric current meter provided with weight and rudder. In use the instrument is suspended by a cable containing the wires for electrical connection to a registering apparatus, but employment of an electric sounder is preferable, the latter being supplied with instruments of this kind in the form of a buzzer which is attached to a small leather case containing a bisulphide of mercury battery cell. Each revolution of the wheel is indicated by the buzzer, and the observer counts off the revolutions in a given time.

Fig. 11 shows a very simple form of Price acoustic current meter, made by the same firm. It is very compact, light and portable and is especially well adapted for measurement of water flowing in irrigation ditches, raceways or streams where there is moderate depth of water. The cut shows the appearance of the meter and sections of brass tubing by which it is held while in use. Revolutions of the wheel are indicated by a hammer striking against a diaphragm one blow for every ten revolutions. The sound of the recording stroke is transmitted through the tubing by which the meter is suspended, and is conveyed to the ear of the operator by means of a rubber tube *T* and receiver *R* The operator fixes the receiver in position for hearing by passing a rubber band around his head, thus leaving both hands free for manipulation of the meter.

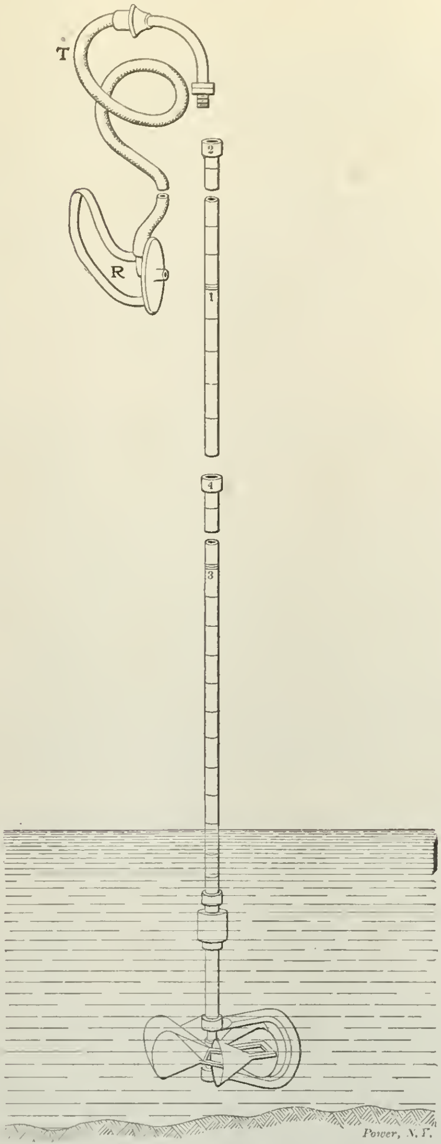


Fig. 11. Price Acoustic Current Meter

The disadvantages attending the use of any form of current meter are mainly that they must be used with care, cannot be used in waters containing floating weeds and grass and that they require to be frequently rated. Their advantages are that they can be employed with less labor and expense than by employment of any other method of obtaining accurate measurements of open streams and can be used in many situations where other methods of measurement would be impracticable. Current meters are usually provided with apparatus for determination only of the number of revolutions of the wheel, and manufacturers generally supply rating tables with their instruments which give values for lineal feet of water passing the instrument per revolution, correct within 1 per cent.; but for best results, indications of an instrument should be checked in some manner before and after an important gaging, especially if it may have been subjected to derangement from rough handling. This can be done by comparison with surface floats, by moving the instrument a known distance through still water, or from its indications of mean velocity of a cross-section whose mean velocity is known.

In obtaining average velocity the current meter is submerged to the desired point of cross-section of the stream for a known time, and the number of lineal feet of current passing the instrument is determined for the given time, and this, divided by the time in seconds, gives the velocity in lineal feet per second.

In using current meters it is best to select a well defined cross-section of stream and one through which the flow is most nearly uniform in velocity in all points of the section. The mean velocity may be obtained either by taking the mean of averages of velocities of equally spaced points in this section; or by taking the mean velocity of the averages of velocities taken along equally spaced vertical lines; or by taking a single reading from movement of the instrument uniformly over the section. The first and second methods are most appropriate for taking measurements with an instrument which is supported by a pole resting on the bottom of the stream, but for employment of instruments capable of being suspended, the last mentioned method has advantage in greater convenience and saving of time.

Wooden Pipes for Conveying Water

In reporting the recent establishment of a factory in Sydney, Australia, for making wood pipes for conveying water, Vice-Consul-General Henry D. Baker gives the following details: In speaking at the opening of the factory the Minister of Works of New South Wales said that the cast-iron and steel pipe factories could not at present fully supply the requirements of the State. So large was the government demand for steel pipes that the manufacturers found it most difficult to fill even the State's orders. These wood pipes were not going to supersede iron or steel pipes, but they were going to meet a demand that could not readily be met by the heavier systems of piping. A most important fact to be considered was that of cost. Here the figures given by him were in favor of wood pipe. For example, a 3-inch wood pipe to withstand a 100-foot head pressure would cost 21⅕ cents a foot, in cast iron 45½ cents, and in steel 38⅖ cents. In the larger diameter the difference was still more apparent. Taking the same pressure, a 12-inch wood pipe costs 72½ cents per foot, against $1.55 for cast iron. Another matter bearing intimately on the cost was that of weight. A 3-inch wood pipe weighs about 4 pounds per foot, a cast-iron pipe 12 pounds per foot; a 12-inch wood pipe weighs 15 pounds per foot, the cast iron 77½ pounds. This entailed a tremendous saving in freight.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Installing Induction Motors

By R. H. Fenkhausen

Belt Drive

Motors should always be installed so that a long horizontal drive may be secured, and the driving side of the belt should be the bottom one. This allows the top or slack side of the belt to cling to a large part of the pulley circumference, with a consequent reduction in the belt tension required, which results in longer life of bearings and a saving in current due to reduced friction loss.

Pulleys should be crowned and well balanced, and for all except small sizes spoked pulleys should be used. A pulley with a web connecting the hub and the rim interferes with the ventilation of the motor and should be avoided in the larger sizes. The only advantage of the webbed pulley is that it can be easily balanced due to the possibility of machining it all over. Paper pulleys give excellent results for motor drive and are supplied by most makers with the motors; the belt tension required for a given load is less than with a steel pulley owing to the better adhesion of the belt to the pulley.

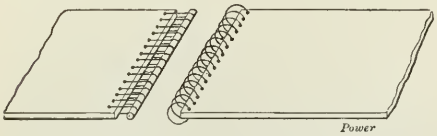


Fig. 1 Wire Lacing in Formation

TABLE 1. INDUCTION MOTOR SPEEDS.

Poles.	25 Cycles.		30 Cycles.		40 Cycles.		50 Cycles.		60 Cycles.		Poles.
	Synchronous.	Full Load.	Synchronous.	Full Load.	Synchronous.	Full Load	Synchronous.	Full Load.	Synchronous.	Full Load.	
2	1500	1390	1800	1700	2400	2230	3000	2830	3600	3350	2
4	750	705	900	840	1200	1120	1500	1390	1800	1710	4
6	500	470	600	575	800	750	1000	950	1200	1120	6
8	375	355	450	435	600	575	750	710	900	850	8
10	300	283	360	350	480	455	600	580	720	630	10
12	250	235	300	290	400	375	500	470	600	580	12
14	214	200	257	248	343	325	428	400	514	495	14
16	187	178	225	218	300	290	375	355	450	435	16
18	167	160	200	190	267	258	333	320	400	390	18
20	150	142	180	170	240	232	300	287	360	350	20
24	125	118	150	140	200	190	250	240	300	290	24
30	100	95	120	112	160	150	200	190	240	230	30
36	83⅓		100		133		167		200		36
42	71½		87½		114		143		175		42
48	62½		75		100		125		150		48

Note—Nameplate speeds of General Electric motors are synchronous. Nameplate speeds of Westinghouse are full load.

Double belts will, of course, carry heavier loads than single belts of equal widths, but where the motor pulley is small the greater flexibility of the single belt makes its use advisable, because it can conform more closely to the curve of the pulley rim, and greater arc of contact is therefore secured. A wider belt must be used to transmit the required power in this case. To obtain the proper size of belt for any given load, multiply the horsepower by 33,000 and divide by the belt speed in feet per minute. This result should be further divided by 50 for single belts and by 80 for double belts to give the width of belt required for the horsepower. This rule applies to leather belts only, with the grain (smooth) side next to the pulley. The driving power of leather belts run with the flesh (rough) side to the pulley is about 30 per cent. less.

For use out of doors, or in locations exposed to water or fumes that would destroy a leather belt, it is often necessary to use rubber or canvas belts. A rubber belt will carry almost as much load as a leather belt run with the flesh side to the pulley, but canvas belts can be relied on only for half this load.

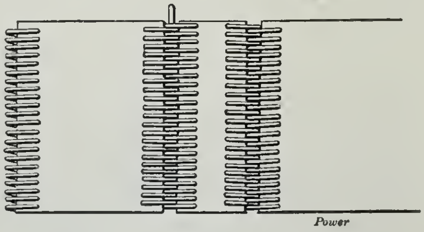


Fig. 2. Wire Lacing Complete

Pulleys should be as large as possible, as not only is the arc of contact greater, but a smaller belt can be used for a given load with large pulleys than with small ones, owing to the greater speed of the belt. The size of pulley selected, however, should be such that the peripheral speed will never exceed 4500 feet per minute, as at belt speeds higher than this there is a decided tendency to slip on the pulley, due to centrifugal force, and the friction and heat caused by this slip will rapidly destroy the belt. The standard speeds for which induction motors may be obtained are given in Table 1. In measuring crowned-face pulleys the size should not be determined either at the peak of the crown or at the edge of the pulley, but at a place midway between. Endless belts are, of course, the best for motor use, but in locations where there is little room to move the motor to take up the stretch of the belt, their use is a needless expense, as sooner or later it will be necessary to cut out a piece.

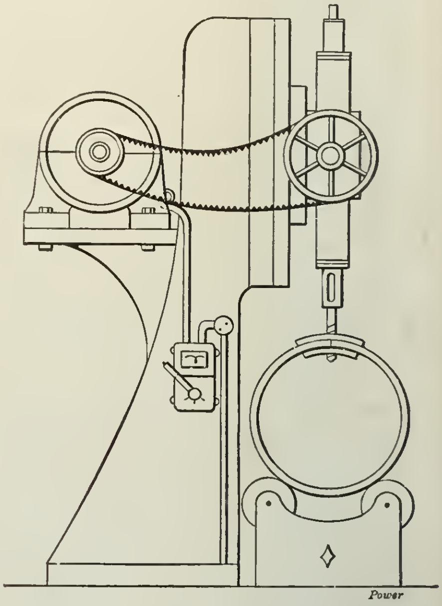


Fig. 3. Chain-driven Drill

Of all known methods of joining belts the rawhide lacing is the worst, especially for motor belts, because the lacing makes a large lump which strikes the pulley with a hammer blow every time the joint comes around, and severe wear on the bearings results. Wire lacing gives good results, but the best lacing known to the writer is that shown in Figs. 1 and 2. A helical spring is inserted in each end of the belt by means of a special tool. The springs are then flattened out, and a small rawhide rod inserted in the loops, forming a hinged joint which is more flexible than the belt itself, and perfectly flush. Another advantage is that the joint may be opened at any time in a few seconds, when it is necessary to remove the belt, and replaced without relacing. When belts are in inaccessible places, it is possible to insert the lacing while the belt is on the floor, and it is only necessary to push the small rawhide rod into place after the belt is passed over the shafts.

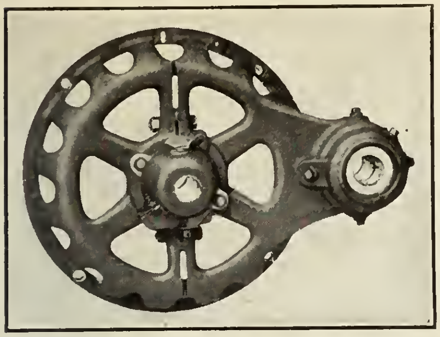


Fig. 4. Journal Bracket with Countershaft Bearing

Chain Drive

For short vertical drives where neither belts nor gearing can be successfully applied, as when a motor located on the floor above drives a line shaft directly under it, the modern "silent" chain drive will give the best satisfaction. There are several types of chain now on the market, the best known being the Renold and the Morse. Both of these makes of chain work with special sprockets accurately cut to mesh with the links of the chain, and when properly installed and not overloaded, they afford one of the most satisfactory methods of connecting a motor to its load. The chain runs more quietly than a gear drive, even when the latter is equipped with rawhide pinion, and is less troublesome and more efficient than a belt.

Fig. 3 shows a chain drive installed in the writer's plant. This is a case in which a small relative movement was necessary between the motor shaft and the driven shaft, and the application would have been impossible with belts or gearing.

Gear Drives

For gear drives the first essential is that both the bearings for a pair of gears must be in the same casting or else bolted to the same rigid foundation. A wood foundation must never be depended upon to maintain the correct center to center distance for a pair of gears, because shrinkage of the wood or loosening of the foundation bolts in their holes is almost certain to occur, and prevent quiet and efficient power transmission. Even when a steel foundation is used, either fitted bolts or dowel pins must be employed to insure the maintenance of the original centers. The best method and the one that should be employed whenever possible, is to use back-geared motors. Almost all manufacturers will supply interchangeable heads for their standard motors, somewhat similar to the one shown in Fig. 4, containing extra bearings for the countershaft, and the bearing centers are so calculated that a wide range of speed reduction (or increase) is possible.

The distance from center to center of a pair of gears should always be calculated. The following simple rules will be found useful for the solution of gearing problems on the diametrical pitch system, which is the one employed for nearly all cut gears:

Gearing Rules

The diameter pitch of a gear is the number of teeth to each inch of its pitch diameter.

The circular pitch of a gear is the center to center distance of the teeth on the pitch circle.

Divide 3.1416 *by the circular pitch to get the diametrical pitch.*

Divide 3.1416 *by the diametrical pitch to get the circular pitch.*

To obtain the diametrical pitch of any gear, add 2 *to the number of teeth and divide by the outside diameter.*

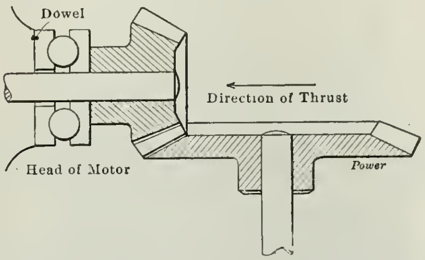


Fig. 5. Thrust Bearing for Bevel-gear Drive

To obtain the proper clearance between the top of the tooth on one gear and the bottom of the tooth on the other gear, divide 0.157 *by the diametrical pitch, or divide the thickness of the tooth on the pitch circle by* 10.

To obtain the working depth of the teeth divide two by the diametrical pitch.

To obtain the center to center distance for any pair of gears, divide the sum of the number of teeth in the two gears by twice the diametrical pitch.

To obtain the pitch diameter of any gear, divide the number of teeth by the diametrical pitch.

The practice indulged in by some "mechanics" of shoving gears into mesh until they look "about right" is barbarous, and should never be allowed. The proper centers can only be determined by calculation, which is quite simple, however.

Rawhide pinions should be used for all motor pinions, but except in special cases they should not be shrouded, as the presence of shrouds on the pinion will often necessitate displacement of the entire motor frame in case the removal of either gear becomes necessary.

When bevel, miter, spiral or worm gearing is to be installed in conjunction with a motor drive, due allowance must be made for the end thrust on the motor shaft and some form of thrust bearing similar to that shown in Fig. 5 installed.

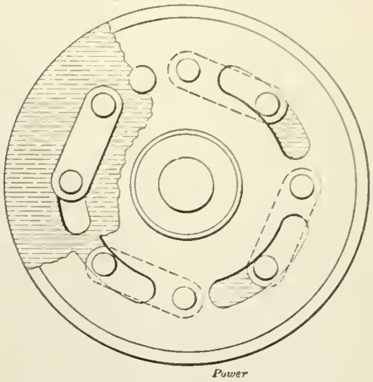


Fig. 6. Flexible Coupling

Ordinary motor bearings are not designed for end thrust and if a special thrust bearing is not used, the shoulder on the shaft will soon wear its way into the soft lining of the bearing, and alter the gear centers considerably, besides risking serious injury to the motor.

Vertical motors also require thrust bearings, but as these are invariably furnished by the manufacturer of the motor it is only necessary to call attention to the fact that such bearings are designed for a limited load, and before any additional load, such as a centrifugal pump, rotor, etc., is imposed upon them, the manufacturer's device should be secured.

Direct Connection

The ideal method of connecting a motor to its load is, of course, by direct coupling, but owing to the high speeds of motors as compared with the average industrial load, its use is rather restricted. A slow-speed motor is, of course, heavier and consequently more expensive than a high-speed motor of the same horsepower, but whenever the speed of the driven shaft is high enough to allow direct connection to a slow-speed motor this method should be employed. The saving in belts, shafts, pulleys, gears, and hangers will almost, if not quite, offset

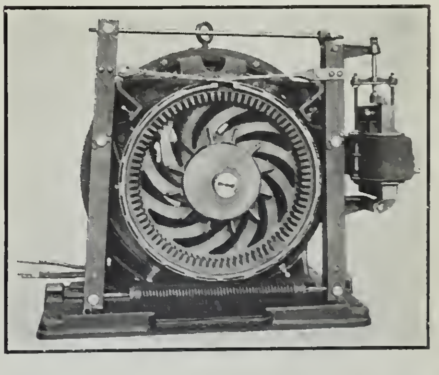


Fig. 7. Brake with Magnetic Release

the difference in price, and the saving in transmission losses and the shafting and belt maintenance will be quite an item. For induction motors the number of speeds possible with direct connection is, of course, limited, there being only one possible speed for each number of poles, as given in Table 1.

Owing to the difficulty of procuring accurate mechanical alinement with three or more bearings, the use of some form of flexible coupling, such as that shown in Fig. 6, between the motor and its load is desirable. This not only takes up any shock and vibration due to unsteadiness of the load but serves to protect the motor from negligence on the part of the mechanical department, in establishments large enough to have separate electrical and mechanical repair forces. Induction motors necessarily have small air gaps and the bearings must therefore be kept in good condition. If the motor shaft is rigidly connected to the load shaft, any undue wear of the bearings of the latter shaft will impose additional load on the motor bearings, but by the use of a flexible coupling each set of bearings can assume its proper portion of the load.

General Suggestions

For reversing service the use of a brake is imperative, in order that the motor may be stopped quickly. Otherwise, the motor connections might be reversed while the rotor is still running under its momentum, which would inflict a severe shock on the motor. A form of brake for use with induction motors is shown in Fig. 7. The magnets are made to respond to alternating current and are connected across one phase of the motor winding. They hold the brake shoes off the drum while the motor is in circuit, but as soon as the current supply to the motor is shut off, the brake shoes are released by the magnet and applied by a powerful spring. Heat radiating fins are cast on the brake drum, as shown.

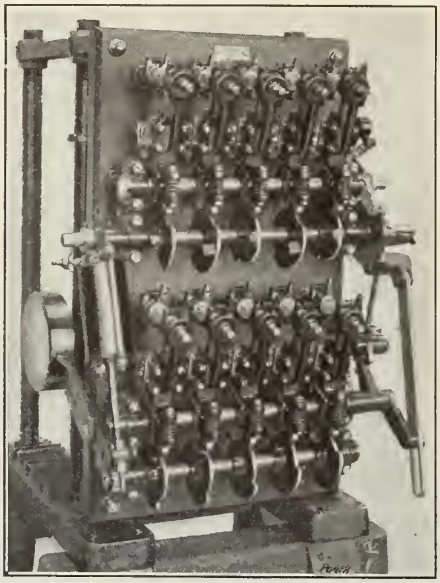
When keying pulleys, gears, etc., to motor shafts, only gib-head keys should be employed; else great difficulty will be experienced whenever the key has to be removed. When the pulley is loose on the shaft, a feather and set screws are preferable to a key, because a badly fitted key in a loose pulley only aggravates a bad fault, whereas a feather, fitting sidewise only, may be made to improve matters by careful location of the set screws.

Overheating of gas engines is caused by insufficient cylinder lubrication or by lack of sufficient jacket water. The pump is usually responsible for this latter deficiency, due to its being worn or to the packing being loose or worn out. A loose union on the supply pipe will allow the pump to draw in air with the water, and air locks have been known to form in water jackets and prevent the water from circulating freely.

A New Motor Controller

The accompanying engraving shows the general appearance of a new controller recently brought out by the Westinghouse Electric and Manufacturing Company for use in conjunction with alternating-current elevator motors. The elevator operator has complete control over the starting and stopping of the motor, but the acceleration is performed automatically at a rate that can be adjusted over a wide range at the controller but cannot be altered by any action of the operator in the car. The electric contacts are of the quick-break, butt type and are protected by arc shields; there are no sliding contacts.

The controller consists of a slate panel on which are mounted two rows of switches and their operating mechanism.



Induction-motor Controller

The switches in the upper row serve to connect the motor primary with the line and those in the lower row cut out, by steps, the resistor in series with the motor secondary. All of the switches are alike, and therefore interchangeable. Five primary switches are used for a three-phase controller and six for a two-phase controller. The switches are opened and closed by cams. In closing, each cam acts on its own switch through a buffer spring which serves as a cushion and also compensates for wear; in opening, a lug on the cam engages a projection on the switch arm and forces the switch open positively. The resistor for the motor secondary is mounted on the rear of the panel.

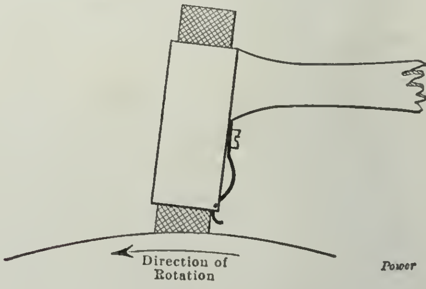
The movement of a hand rope or lever in the elevator car operates a sprocket attached to the primary cam shaft; turning the cam shaft from the off position closes the primary switches and starts the motor, moving the elevator car up or down according to the way the shaft is turned. The movement of the primary cam shaft simultaneously releases the secondary cam shaft and allows a weight, attached to an arm geared to that shaft, to fall; this rotates the secondary cam shaft and closes the secondary switches in proper order. The fall of the weight is retarded by an air dashpot, the piston speed of which can be so adjusted by means of a valve that proper acceleration can be given to the motor.

On turning the primary cam shaft back to the off position, the primary switches are opened and the link at the end of the shaft pulls the secondary shaft to the "off" position, opening the secondary switches and raising the weight, ready for the next operation. Opening the primary switches completely disconnects the motor from the line and stops it.

LETTERS

Copper Plating and Brush Heating

Referring to P. L. Werner's letter in the issue of August 23, may I suggest that the copper plating on dynamo and motor brushes is not intended so much to reduce the resistance of contact between the brush and a box-type holder as to reduce the longitudinal resistance of the brush between the face and the point where the pigtail is attached. However, there is no doubt that unequal efficiency of contact between the several brushes and their holders will overload those brushes which have the most efficient contact; but a better plan than scraping off the copper plating is to attach a flat strip of spring bronze to the rear side of each holder so that the end of the strip will press against the rear side of the brush just above the commutator, as indicated in the accompanying sketch. It is necessary, of course, to bend the end of the strip into a curved form and to file a slot in the wall of the box to give the curved end access to the brush. If the construction of the holder and its shank is such that the spring cannot be attached to the rear side of the box, I should put it on the front side and turn the holders around so that the commutator surface runs toward the holder shanks instead of away from them.



Mr. Malcolm's Arrangement

Geo. W. Malcolm.

Brookyln, N. Y.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Portable Gas Power Plant

The accompanying engravings illustrate a portable suction gas producer and engine recently designed by Capel & Co., of London, to do the same work and fulfil the same conditions as the portable steam engine. The advantages claimed for the portable suction engine over the steam engine are as follows: (a) The initial cost is about half that of the steam engine; (b) the weight of a gas engine is about two-thirds of that of a steam engine; (c) the cost of running the gas engine is from one-tenth to one-twentieth of that of a steam engine; (d) the gas engine can be started in 10 to 15 minutes from cold, and (e) the attention required by it is much less than that required by a steam engine.

The halftone illustration is reproduced from a photograph of an 18-brake-horsepower set. The engine itself has a shallow bedplate bolted to the girder framework. The cam shaft is supported in three bearings of the ring-oiling type. The cylinder is supported by the bedplate right to the back, and the liner and piston are longer than it is the usual practice to make them. The liner is held up at the back by a broad flange, and is left free to expand at the front. There is no water joint at the end of the liner and the makers allege that they have eliminated one of the most frequent causes of trouble in gas engines. The governor controls the speed by throttling the mixture. Ignition is effected by means of a magneto, with variable timing gear.

The cylinder is lubricated by a force pump; a separate lubricator is provided for the gudgeon pin, and the crank is lubricated by a centrifugal ring. The lubricators can be refilled without stopping the engine. The water-cooling spaces in both cylinder and breech end are large, and can be inspected at any time by removing the back plate—a simple operation, taking a minute or so.

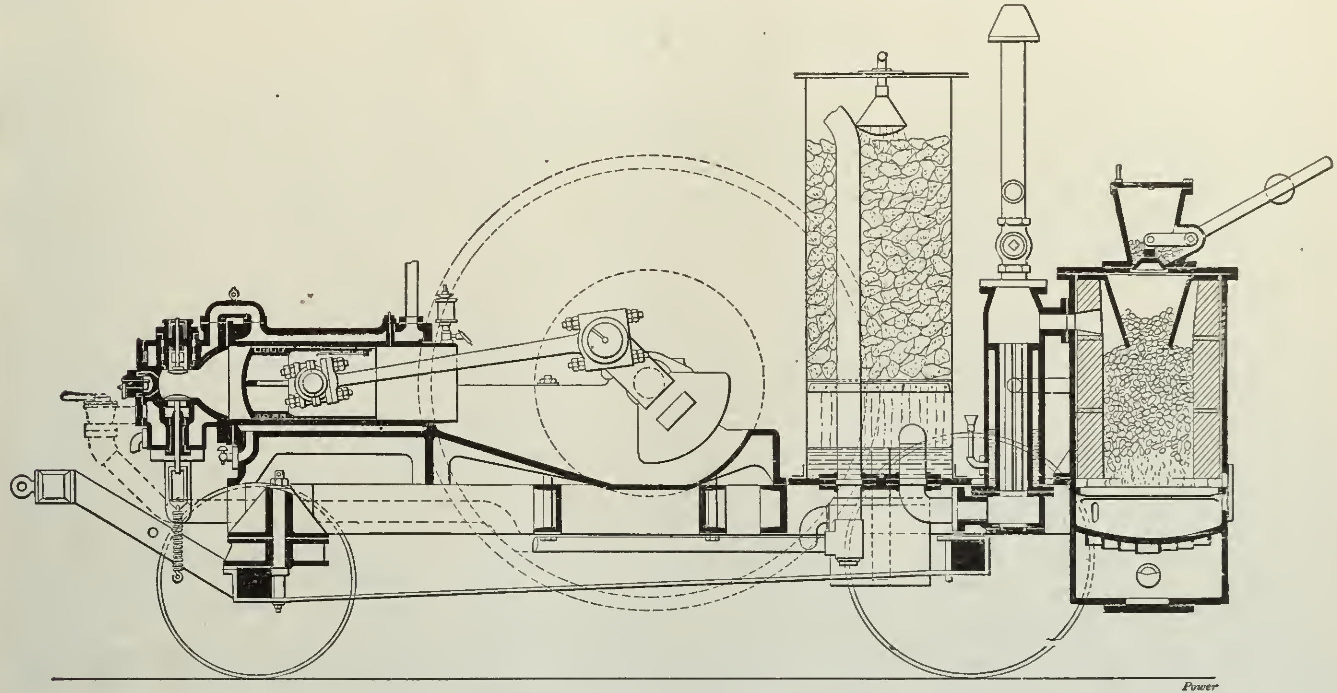


FIG. 2. SECTIONAL ELEVATION OF PORTABLE SUCTION GAS-POWER PLANT

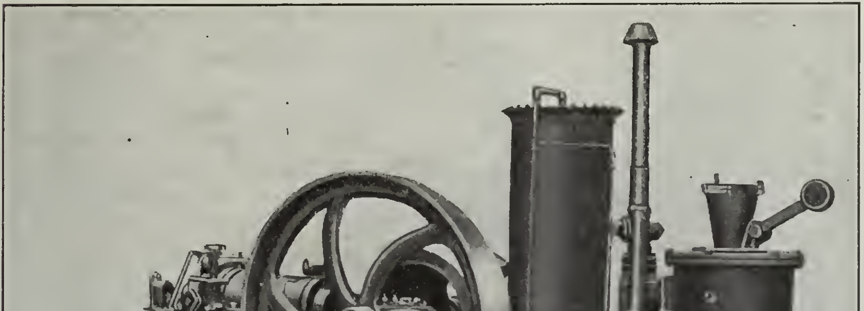


FIG. 1. PORTABLE SUCTION GAS-POWER PLANT

The gas producer has been specially

designed for this portable plant, and the grate area has been made large so as to burn coke, or even charcoal, as well as anthracite coal. The generator is arranged on the regenerative principle, the exhaust from the engine being led into an expansion chamber underneath the ashpit, the gases being directed by a curved pipe against radiating vanes on the under side of the ashpit. A shallow layer of water is kept automatically in the ashpit immediately under the fire grate, and the combined heat of the engine exhaust below and of the fire above evaporates this water to the full extent that it can be taken up by the fire. This arrangement, the builders state, has proved of great advantage in their plants. With it the consumption of fuel is considerably reduced, the gas is cleaner and more reliable, and the grates last longer by reason of the large quantities of steam available. The scrubber is of the ordinary type, except that the inlet gas pipe from the generator, the outlet gas pipe from the top of the scrubber, and the overflow water pipe all enter underneath. The sump box is fixed immediately underneath the scrubber between the girder frames.

The framework is supported on axles of 4-inch square iron. The road wheels are double spoked, 8 inches wide, and the whole plant is said to be particularly rigid, the engine running under full power without vibration, almost as if on a concrete foundation. On the pulley side of the engine there is a clear run back and front, so that a belt drive can be applied in either direction, as the case may require.

The cooling of the engine and scrubbing of the gas are carried out in a simple manner. A small plunger pump, driven by an eccentric from the cam shaft, pumps water from a small rectangular tank alongside the engine through the cylinder jacket and scrubber and over a wire-mesh cooler. A comparatively small amount of water does, we understand, all the cooling and cleaning necessary. The following are some of the leading particulars of this suction gas-power plant:

Cylinder bore......	9 in.
Piston stroke......	15 in.
Flywheel..........	4 ft. 8 in. diam. by 6½ in. face
Pulley............	2 ft. 4 in. diam. by 10 in. face
Road wheels:	
Front pair.......	2 ft. diam. by 8 in. face
Back pair.......	2 ft. 6 in. diam. by 8 in. face
Overall length, exclusive of shafts..	12 ft.
Overall width at axles............	5 ft. 6 in.
Overall width at draw bar........	8 ft. 6 in.
Hight from ground to top of scrubber.	6 ft. 3 in.
Hight from ground to center of engine	3 ft.
Total weight when ready for running.	about 4 tons

We are indebted to *The Engineer*, of London, for the illustrations and the foregoing particulars.

Edward T. Adams Leaves the Allis-Chalmers Company

The announcement that E. T. Adams, for five years the chief engineer and manager of the Allis-Chalmers Company's gas- and mill-engine department, has resigned that position naturally calls to mind some of the notable achievements which marked his connection with the company. Amongst these is the design of the pumping engines at the Thirty-ninth street station in Chicago, which engines showed a duty of about 150,000,000 foot-pounds per 1000 pounds of dry steam; this was about 50 per cent. higher than the best previous record.

Another notable piece of work was the originating of the present Allis-Chalmers gas-engine design. When Mr. Adams took hold of the department, the Allis-Chalmers Company had made a deal with the German builders of the Nürnberg engine to use their designs and had built a 300-horsepower engine which was in operation in the shops. The complexity of the valve gear and some radical divergencies from American practice in steam-engine construction proved insurmountable obstacles to the marketing of the German type of engine, so Mr. Adams worked out the present designs, from the foundation up. The only Nürnberg feature retained was the location of the inlet valve at the top and the exhaust valve at the bottom of the cylinder, and this was not exclusively a Nürnberg feature but had been used for years by other European builders.

Under Mr. Adams' supervision the Allis-Chalmers Company has built nearly 50 engines of 4000 horsepower each; these include the engines in the Gary plant of the United States Steel Corporation. A remarkable feature of the huge contract for the Gary engines was that when it was made the builders had not in operation a single engine of this type, so that the designer had no tangible evidence as to the probable success or failure of his designs.

It is not generally known that Mr. Adams has worked his way to the top from the grade of journeyman. In his young days he carried the tin dinner pail, and it was never full enough to endanger the handle. He worked in the "shop" for the money to pay for his technical education, and after completing it he was employed as an instructor at Leland Stanford Junior University and later at Cornell. The taste for mechanics was stronger than the pedagogic instinct, however, and he soon abandoned teaching to take a position as personal assistant to the late Edwin Reynolds. While in this position he worked into pumping-engine design and construction, making a specialty of low-lift work.

When the E. P. Allis Company combined with Fraser & Chalmers, the Gates Iron Works and the Dixon concern to form the Allis-Chalmers Company, Mr. Adams had worked up to the management of the Allis pumping-engine department, which he resigned to take charge of the gas-engine department of the Westinghouse Machine Company. After holding this position three years, during which period he made an excellent record in designing engines for blast furnace and producer gases, he went back to Milwaukee to take the position which he has just resigned.

Mr. Adams recently spent three months in Europe, studying the internal-combustion engine industry there, and found that the latest German designs show great contrast with those which have hitherto been considered representative practice in that country. They are even simpler than the typical American designs.

CORRESPONDENCE

Was It Premature Ignition?

In a letter in the June 14 number, headed "What broke the cylinders off?" Clarence Stirling tells of an accident to a three-cylinder gas engine in which two of the cylinders broke away from the frame. The account he gave of the matter would lead one to suspect premature ignition as the cause of the mishap.

Suppose a three-cylinder gas engine to be running under a heavy load and for some reason one cylinder stops firing; the resulting overload would probably pull the engine up to such an extent that the timing of the ignition as set for normal speed would be much too early, causing explosions before completing the compression stroke, which might break off the two active cylinders. If some part of the ignition mechanism should break, serious back-firing might cause the cylinders to let go.

Frank E. Booth.

Toronto, Canada.

[It is difficult to conceive how back-firing could break the cylinders loose from the frame, because the force of the explosion, when back-firing, is relieved by the inlet port being open. Perhaps Mr. Booth meant preignition instead of back-firing.—Editor.]

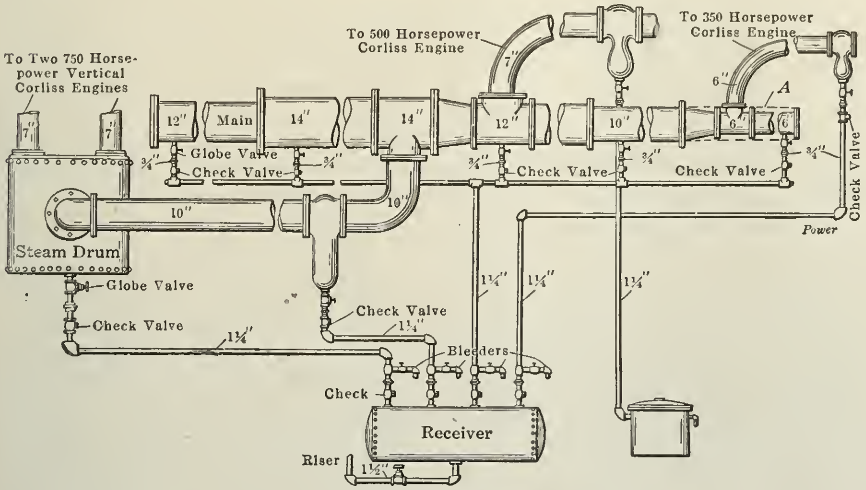
Broken threads on a spark plug, broken gaskets, a cracked exhaust pipe, loose pet cocks and loose igniters cause a hissing sound, while squeaks are usually due to lack of lubrication. Knocking may be due to something loose, but it is often caused by the ignition point being advanced too much or by pre-ignition, due to carbon deposits. If due to looseness of some parts, the flywheel may be loose on the shaft; the main bearings or a connecting-rod box may be worn. Lack of lubrication or too rich a mixture will also produce thumping.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Trouble with a Gravity Loop

The accompanying sketch shows the main steam piping to our engines, together with the Holly gravity-loop system of handling the water of condensation. A 10-inch pipe leads from the 14-inch main, through a separator to a steam drum. From this drum two 7-inch pipes pass up through the engine-room floor to two 750-horsepower vertical Corliss compound-condensing engines. Another 7-inch branch connecting with the main header supplies steam to a 500-horsepower horizontal Corliss compound-condensing engine, the separator of this line is connected to a bucket trap. A third branch, 6 inches in diameter, supplies a 300-horsepower Corliss simple noncondensing engine.



STEAM LOOP

Three of these engines would not carry an overload without taking water over into the cylinders, the simple engine once knocking its head out and smashing the piston and crank disk. The two vertical engines did not meet with any serious mishap, but when water came into the cylinders they pounded badly. The 500-horsepower engine, however, did not cause any trouble.

After the small engine was wrecked the conclusion was reached that the fault was in the steam main and the 6-inch pipe was replaced by a 10-inch pipe as indicated by the dotted lines at *A*. This did not improve matters any and the next move was to examine the check valves on the drip pipes to the gravity-loop system.

It was found that every one of the check valves was out of order. When these were repaired no more trouble was experienced from water being carried over. Moreover, it is now possible to carry the peak load with two engines where formerly three were required.

THOMAS SHEEHAN.

Pittsfield, Mass.

Auxiliary Economy in an Ice Plant

In the modern ice plant it is as necessary to obtain efficiency from the auxiliaries as it is from the refrigerating machine itself. In a plant with which I am connected an atmospheric type of fore-cooler was used. The cooling water contained much scale-forming matter and as the water was sprayed over the coils from a tank clamped at the top of the cooler, considerable water evaporated and left a deposit of scale; consequently the best the cooler could do was to reduce the temperature of the distilled water from 212 degrees to 100 degrees, whereas the temperature of the cooling water was 70 degrees. Hence, the temperature of the distilled water leaving the cooler was 30 degrees higher than the temperature of the cooling water, which meant that each pound of water contained 30 B.t.u. more than it should if the cooler had been doing its work properly. As 100,000 pounds of distilled water passed through the cooler every 24 hours, it meant that 3,000,000 heat units were delivered to the ice tank each day in excess of that which the tank was designed to take care of. As one ton of ice from and at 32 degrees Fahrenheit is equal to 284,000 heat units, it was evident that we were losing nearly seven tons of ice daily.

To obviate this loss we replaced the atmospheric type of cooler with a double-pipe type of cooler in which there was less chance for evaporation and consequent deposit of scale. After making the change we were able to obtain an absolute interchange of temperature between the distilled water and the cooling water. This resulted in an increased output of 6½ tons of ice per day. When it is understood that the plant did not have an abundant supply of cooling water, and that the ice retailed at $10 per ton the increased efficiency can be readily appreciated, as the increased receipts netted the company $65 per day.

PAUL MCINTIRE.

El Reno, Okla.

Removing Links from a Flywheel

It was desired to remove the oval links from the rim of a flywheel and as they had been put on hot the difficulty of the operation became evident. At the start it was realized that it would be necessary to heat the links without heating the wheel, for if both were heated the links would become tighter. The operation was accomplished in the following manner:

Two large coal-oil blow torches, operated by compressed air, were used to heat the link and the space outside the link, also the portion inside the link was packed with sheet asbestos. As soon as the flame was removed, cold water was sprayed onto the side of the wheel opposite to the link and a chunk of ice was applied to the inside and outside of the rim and another piece on that part of the wheel inside of the link. Then with a bar in each eyebolt the links were removed at the rate of one every 50 minutes.

The coefficients of expansion of steel and cast iron are different, the cast iron expanding more than steel. Hence, had both been heated the link would not have come off. The coefficient of expansion of cast iron is 0.00000617 and of soft steel 0.00000599. If the link, which was 24 inches long, could have been heated 1500 degrees more than the cast iron of the wheel, the clearance of the link over the wheel would have been 0.215 inch provided the link and wheel sections were the same length when cold. However, all links for holding sections of flywheels together, when first made, are machined to a length somewhat shorter than the section of the wheel, so that when heated and put on they hold the wheel sections firmly together.

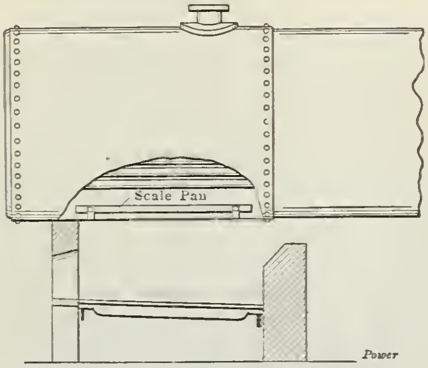
FREDERICK L. RAY.

Louisville, Ky.

How to Prevent Bagging

A device for preventing the bagging of fire sheets, which was formerly in common use in the boilers of steamboats plying the Mississippi river, but which is practically unknown to the stationary engineer, is what is called a scale pan. This device is made of galvanized or thin sheet iron, about 2 inches deep and 5 or 6 feet long, and as wide as the entrance through the manhole will permit. Small feet are riveted to the bottom or sides to hold the bottom of the pan about 2 inches above the shell.

These pans are placed inside the boiler just over the fire sheet as shown in the illustration; and their function is rather remarkable to those not familiar with their use. In the operation of a boiler of the horizontal-flue or tubular type, all the loose scale is carried by the circulation in more or less definite paths, a fixed direction of which is a vertical movement directly above the fire sheets. With the scale pan located on the fire sheet as shown, there is an eddy formed in the circulation immediately above the pan; and loose scale that gets into this, settles down into the pan and remains there.



SCALE PAN IN BOILER

As a proof that the scale settles down from above and is not swept in over the sides, the pan, when removed, is found to be packed with scale resting on edge as evenly as if placed there by hand. No loose scale will be found on the fire sheets when such a device is used, unless, the emptying of the pan is neglected until it is completely filled. Those who are troubled with loose scale in their boilers, will find this device an excellent preventative for bagged or burned sheets. It, of course does not in any way tend to prevent the formation of scale; it merely cares for the scale that has been formed and has become detached from the heating surface. Such scale, however, is often a source of serious trouble.

J. E. TERMAN.

New Haven, Conn.

The Expansion Valve

The purpose of the expansion valve is not generally understood by engineers operating ice and refrigerating machinery; in fact, there seems to be an erroneous opinion concerning it. One reason for this may be in the term "expansion valve," which to me appears to be a misnomer.

Many seem to think that the valve is for the purpose of regulating the rate of expansion of the ammonia liquid; that this valve is to be opened only enough to allow the liquid ammonia to instantly flash into a vapor and that great care must be taken to prevent any ammonia liquid from getting past it.

Ammonia was chosen as a refrigerating agent for a number of reasons, one of which is that ammonia gas under moderate pressures will rise in temperature to a point at which water at ordinary temperatures can be used to remove the heat from it; also because a certain quantity of ammonia gas will be condensed for every heat unit that is removed by the water. The fact that it will condense at moderate pressures makes it particularly valuable. If, like air, it condensed only when subjected to excessive pressure, it would not be of commercial value.

It might be stated in this connection, that it is possible to refrigerate with air or other gases without condensing them. It is necessary only to compress a given volume, at a low pressure, into a much smaller volume, thereby raising the pressure and temperature; then, by removing the heat the gas will expand to its original volume and take up heat from the surrounding vessel. The great objection to using air and some other gases, which are much cheaper than ammonia, is because of their high vapor tension; that is, they give up their heat slowly and, even after giving it up they remain in a gaseous state, occupying practically the same space as before and prevent other hot gases from coming in contact with the heat-removing surfaces.

A greater number of heat units must be removed from a pound of ammonia gas to change it to a liquid than from any other gas whose boiling point is sufficiently low at low pressures to warrant its use for refrigerating purposes. Also, ammonia at low pressures will boil at a temperature below that of the surrounding atmosphere.

Still another reason for ammonia being the choice of many for this purpose is because of its high latent heat. At a gage pressure of 152 pounds, 502 heat units must be removed from a pound of ammonia gas to liquefy it at a temperature of 85 degrees Fahrenheit. To reevaporate this pound of liquid it is necessary to apply 502 B.t.u. This is done in refrigerating systems by opening the expansion valve and allowing the ammonia liquid to pass to a lower pressure (say, 15 pounds gage), in the cooling coils. Under this pressure its boiling temperature is zero degrees Fahrenheit as against a temperature of 85 degrees at 152 pounds. For the purpose of illustration consider the specific heat of ammonia liquid as one, which is nearly correct. In passing through the expansion valve a pound of ammonia liquid will fall 85 degrees in temperature, and 85 B.t.u. will be given up by the ammonia to reduce its own temperature. Due to this change of temperature,

$$85 \div 502 = 0.169$$

pound of ammonia liquid will be evaporated into gas. Beyond the control of the expansion valve

$$(502 - 85) \div 502 = 0.832$$

pound of ammonia will remain in the liquid state to be boiled finally by the heat to be absorbed from the cold-storage room or ice tank, which must give up 417 B.t.u.

J. J. NASH.

New Haven, Conn.

What Causes the Scale?

We use steam-heated cast-iron forms to bend wood into various shapes. The wood is first dipped into hot water (never soaked) and then placed between the forms. After using for a short time the smooth surfaces of the forms take on a black scale, similar in appearance to rubber. This stains the wood and necessitates frequent scraping and cleaning. Kiln-dried basswood is used and contains no pitch.

Are there any POWER readers who have had a similar experience, or who can account for the formation of the scale?

E. F. LYNCH.

Charlestown, N. H.

Philosophical View of Trouble

Not long ago I visited a large turbine plant in which the working steam pressure was 200 pounds with 150 degrees superheat. I asked the chief if he had ever had any real troubles with leaky valves due to superheated steam. "Not a bit," he replied. Just then the steam fitter came in and said, "Steam is blowing through that joint at such a rate that we cannot put in a new gasket; that gate valve leaks like a sieve."

The chief touched a button and when another man, who was the operating engineer, appeared, he said to him: "Start the auxiliaries on No. 2 turbine and when they are going all right open the throttle and see if the condenser and air pump will take care of the steam that is leaking past the line stop valve."

After the men had left the room he turned to me and said, "That is the worst case of trouble I have had from the use of superheated steam. Such an experience might annoy some men, but I do not think of calling a thing trouble that can be remedied by giving a few orders. Such things do not occur often enough to increase the operating expenses of the plant and I am not going to get wrinkles and headaches over them. It is no more trouble to start a 3000-kilowatt turbine to take care of a little steam than it would be to put the turbine into regular service. By the time the job is completed, it will be nearly dark enough to call for throwing in another unit and it will already be running. Trouble, well, no, not much."

H. A. WARNER.

New York City.

A Broken Valve Stem

Our engine is of the slide-valve type with a throttling governor and runs at 150 revolutions per minute. One morning upon starting, the governor seemed to stick and the engine speeded up more than usual. But as the governor soon had the engine under control, I concluded that the trouble had been due to the stuffing box on the governor stem.

Everything ran smoothly until about 8 p.m. when the valve stem broke. However, the engine stopped before I reached the throttle. We proceeded to remove the valve-chest cover and found a strip of brass about 3 inches long wedged tightly between the valve and the seat. The question arose, "Where did the piece of brass come from?"

About four years ago the piping between the boilers and the engine had been removed and larger piping installed. The new piping had been unloaded upon the ground among a pile of scrap and had then been put into place without being cleaned out. This undoubtedly accounts for the presence of the piece of brass which had worked along the pipe until it reached the valve chest.

L. H. SISSON.

Muskegon, Mich.

Fan Troubles

Some years ago I was sent to locate the trouble with a drier, such as is used to dry sand and broken stone. I started without blueprints, confident that the trouble was due to the people in charge not knowing how to operate the fan and that I did. The fan was of the disk type, 4 feet in diameter and was running at its rated speed in the right direction. The smoke was pouring out from almost every place except the top of the stack.

My first move was to open the cleaning door at the bottom of the fan housing, thinking that the fan was starved, but with no results. After due deliberation I had the 16-foot section of the stack unbolted at the elbow, but with still no results. While the stack was off, I climbed on top of the elbow to test the force of discharge with a newspaper and instead of the paper being blown away, it was drawn down the elbow and into the fan.

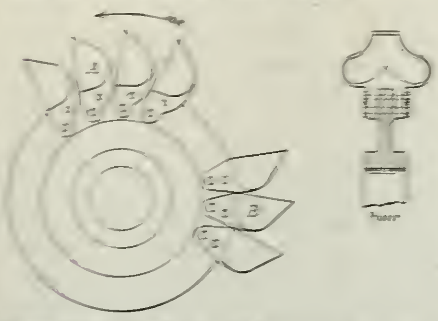
About this time I began to wish for some blueprints, but as they were 1000 miles away, they were out of reach. After taking numerous measurements and using the long-distance phone, I had a new elbow made at our shop. When it arrived, we found that it was 2 inches larger in diameter at the fan end than the one that had been sent with the fan. The elbow, stack and fan casing had 2-inch angle-iron flanges riveted to them and the flange at the fan end of the old elbow projected about 1 inch all around the fan flange. Then we noticed that there were two sets of holes in the fan flange; the first set had not fitted the elbow, so the erector had made a new set. It was this unused set of holes that caused all the trouble.

ALBERT [illegible].

Cleveland, Ohio.

What Causes the Bolts to Break?

A 3-foot water wheel working under a 320-foot effective head and running at 300 revolutions per minute has been giving considerable trouble by the breaking of the bolts which hold the buckets to the runner. The wheel is constructed as shown at A in the sketch, and it is the smaller set of bolts that break.



MANNER IN WHICH BUCKETS ARE ATTACHED

The centrifugal force causes a tendency of the bucket to swing around the larger bolt in the direction of the arrow, while the force of the water tends to swing the bucket in the opposite direction. My theory is that the bucket vibrates between these two forces and crystallizes the bolt by this continual jarring.

The wheel is built so close that it is impossible to tighten the bolts with an ordinary monkey or S-wrench; hence it was the practice to tighten the nuts with a hammer and chisel. Recently, however, socket wrenches have been used for this purpose, and no more bolts have broken. The wheel has now been run about two weeks, while formerly a bolt broke often.

I believe that the buckets should have been designed like those shown at B, so that the centrifugal force would have a tendency to swing the bucket in the same direction in which the force of the water acts, so that there would be no vibration between the two forces.

C. E. [illegible].

[illegible]

Rope Transmission Problem

In a certain mill a rope drive was installed under the following conditions: The engine pulley was 14 feet in diameter and grooved for 12 ropes. As only about one-half of the machinery was installed at first, a sheave 7 feet in diameter was placed on the driven shaft, this sheave containing seven grooves. When the remaining machinery was later installed, another sheave was ordered and placed on the driven shaft beside the first one, both were keyed to the shaft, making them the same as one sheave. The same amount of rope was added to fill the entire 12 grooves, the drive being operated on the continuous wind system.

After the change had been made it was discovered that the new sheave was ½ inch smaller in diameter than the other one. The engineer of the plant would like some opinions as to whether this difference of diameter between the two sections of the driven sheave will seriously interfere with the efficiency of transmission. What effect will it have on the life of the rope, and which section of the driven sheave will transmit the greater amount of power?

E. [illegible].

Blackwell, Okla.

A Problem in Economy

We have a cross-compound, two-stage air compressor, compressing to 100 pounds and rated at 1000 cubic feet of free air per minute. As only about 350 cubic feet per minute are used I would like to know whether it would be cheaper to use the air for forcing the fires of our boilers instead of steam which is now employed. In this way the machine could be worked at a higher efficiency. We use the exhaust steam for heating the feed water and in winter for heating the building, and it seems to me that the use of the air would be a cheaper proposition.

The steam blowers used at present are of ¾-inch pipe and the boiler pressure is 150 pounds. The feed water enters the boiler at 200 degrees. The boilers are of the firebox type, each having 1800 square feet of heating surface and 27 square feet of grate surface, and a natural stack draft of ½ inch.

We also have two 100-kilowatt turbine-driven alternators which carry only an average load of 45 kilowatts. Would it make an appreciable saving to install a motor-driven air compressor of about 300 cubic feet capacity to bring the machines up to nearly full load?

H. M. LEWIS.

Lincoln, Neb.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Measuring Clearance

I noticed in the June 14 issue an inquiry on getting the clearance of an engine or compressor. I offer my method as it is quick and accurate.

Set the engine on its crank-end center, loosen the piston-rod nut and screw the rod into the crosshead until the piston strikes. Then, set the nut up with the fingers and roll engine over onto its head-end center and back the rod out of the crosshead until it hits the cylinder head. Caliper the distance between the nut and the crosshead. This gives exactly the clearance.

P. S. Black.

Terrell, Texas.

The Advantage of Compression

In the article, "Is Compression Economical?" in Power for June 28, the subject seems to be dismissed by the editorial comment, inserted on the first page of the article, with the faint praise that compression is merely an aid in "helping the engine quietly over the center." As "compression" has been such a faithful servant to me, I am not willing to have it "damned with faint praise."

I agree with the article referred to in the conclusion that the work done by a well designed and constructed engine with a given amount of steam is not appreciately changed with or without compression and that the only apparent benefit is the quiet and smooth running of the engine. But the consequent low cost of upkeep of the brasses may be called economy. There are benefits gained outside of the engine room which are derived from the engine passing its center quietly.

In order to make this plain it will be necessary for me to be reminiscent. I was called upon by a business man of ample means from a Mississippi river town in Iowa in the middle fifties. He had traveled by stage some 60 miles to consult with me about the probable cost of the machinery necessary for a flouring mill, the output of which would be 500 barrels in 24 hours. After the preliminaries had been discussed, the question was propounded, "Can you build a steam mill which will give as good results as a water mill?" By this my client wanted to know whether it was possible to obtain as steady motion from an engine as from a waterwheel. That an engine was capable of doing this was denied by nearly all head millers of that period. My answer was that I could and that the miller on the grinding floor with no other knowledge than that obtained from the operation of the machinery could not tell whether it was driven by water or steam. The result of the conference was that I was commissioned to go ahead with the plans and erection of a steam-driven mill. My employer instructed me not to spare any expense necessary to make the mill first class in every particular and capable of making 500 barrels of flour in 24 hours without using more than five bushels of clean wheat per barrel.

In the middle fifties railroads west of Chicago were conspicuous by their absence and as the mill was on the Mississippi river, St. Louis was the most natural place to go to find what I wanted. A Corliss engine was considered but the high price, difficulty and uncertain transportation facilities and long time wanted for delivering caused its elimination. The engine which was selected was 24 inches in diameter and had a 72-inch stroke. The engine was of the side-lever, balanced-poppet valve style. The admission valves were actuated by pointed cams made to cut off admission at one-quarter stroke and arranged so that 5 per cent. of variation either way could be made. The exhaust valves were operated by separate cams, arranged for adjustment for late or early closure. The engine outfit was of typical Mississippi river steamer style. The engineer selected to take charge was an old "river" engineer. He nearly had a fit when he learned that this engine was to make 50 revolutions per minute.

The 12 runs of 5-foot stone in line on a husk frame at the front of the mill were driven by cone gears 60 inches in diameter matching pinions on the stone spindles 30 inches in diameter. Both cone gear and pinion had an 8-inch face. The husk shaft was driven by a belt, 40 inches in width, from the engine shaft. The driving pulley on the engine shaft was 24 feet in diameter. The receiving pulley on the husk shaft was 12 feet in diameter. The flywheel on the engine weighed about 15 tons. I found that the ratio of the speeds of the engine and the mill stones was not large enough to secure smooth operation. The distance between the centers of the engine shaft and the husk shaft was 75 feet and a tightener pulley, 60 inches in diameter, rode on the belt 20 feet from the husk shaft. The tightener would "bob" up and down in synchronism it seemed with the exhaust of the engine and the growl of the cone gears played an accompaniment to the dancing of the tightener.

I had an indicator even at that early date. I had never seen one but I had seen a picture of one in the *Scientific American* and as I realized its utility I made one which answered my purpose for some years. When I attached the indicator to the engine and took some diagrams my "river" engineer was nearly paralyzed. The diagrams were as nice square-headed cards as one could wish for. I manipulated the exhaust cam until I stopped the dancing of the tightener and the growl of the cone gears and after squaring up the admission valve I took some more diagrams and found that I was using compression for about 15 per cent. of the stroke. Everything was running silently and smoothly. The old engineer swore by all that he held sacred that they were using only *about* three-quarters of as much coal as they had been when they started.

This is one of the reasons why I do not like to have compression lightly referred to as something we may use or not as it suits us. I would advise young engineers to make a close study of the question, not so much with the mathematical formula but with just good common sense; some theory and a "heap" of practice.

Of course, the old engineer was away off about the saving in coal but all was going so smoothly and nicely that we were willing to let him have his think.

W. A. Clarke.

San Diego, Cal.

Incompressibility of Water

In the July 12 issue, M. M. H. asks "if water at great depths is so dense that iron will not sink to the bottom," and the answer given was that "water is incompressible." This is not absolutely correct, but is approximately correct.

"Kent" states that the compressibility is from 0.000040 to 0.000051 for one atmosphere depending upon the salts held in solution and also upon the temperature. Figuring roughly, the weight of cast iron as 450 pounds per cubic foot and water at atmospheric pressure as 62.5 pounds per cubic foot, and assuming a cubic foot of water to increase ½ pound for every mile in depth, at a depth of 775 miles cast iron would stop sinking.

Perhaps some of the unfathomed parts of the ocean run this deep.

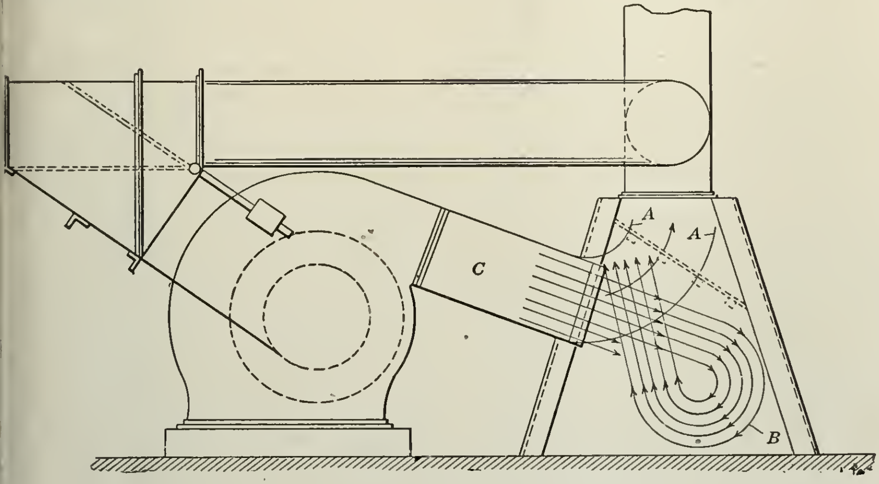
This "splitting hairs" over such a matter may be unnecessary, as perhaps M. M. H. wanted only an approximation.

OSMER EDGAR.

St. Louis, Mo.

Induced Draft Trouble

Referring to the letter of Mr. Faulthier, in the July 12 issue, I believe his trouble is caused by the manner in which the discharge pipe *C* in the accompanying sketch is set. The effect is shown by the arrows *B*. Any increase of power or speed will only magnify this effect, but the draft will not improve.



DIRECTION OF CURRENTS INDICATED BY ARROWS

The trouble may be corrected by an elbow being attached as at *A*. This is not the best, but it is the cheapest solution of the trouble.

W. H. KELLER.

Ansted, W. Va.

A right-angle bend is a bad thing in a chimney flue. An acute-angle bend is much worse. The gas is choked after leaving *C* by having its direction suddenly changed at an acute angle. The remedy is to make a curved flue between C and the stack so as to reduce the frictional losses.

WILLIAM KENT.

Montclair, N. J.

Stress in Rim Bolts

In reply to Patrick Mulham, who inquires in the July 19 issue about the stress in the rim bolts of a driving pulley, I give my opinion that the bolts are both in tension and in shear. They are in tension due to centrifugal force and in shear due to inertia.

CHARLES A. BLUE.

Roslindale, Mass.

Bearing Lubrication

In the issue of POWER for July 19, William Kavanagh has reiterated some of those old theories known to the older class of engineers. Some of the statements, to my mind, are misleading to the younger fellows who are working their way up the ladder. For instance, the use of the brass tube in the hollow crank pin was abandoned in all good practice years ago. It looks O. K. theoretically, but let us study the practical side a little. The tube has to be filled before the engine is started and the plug screwed in. How long is it to be supposed that the quantity of oil contained in this small tube (it must necessarily be a small tube in order not to weaken the pin) is going to last? There are no means of replenishing the tube without shutting down. Most all of the modern engines that are fitted with a hollow crank pin are also fitted with a centrifugal oiler or some arrangement to provide for a continuous supply of lubricant.

Mr. Kavanagh states that all of the soft-metal bearings in his plant were consigned to the scrap heap long ago. In my experience, covering a period of 20 years, I have found that a brass box will not work successfully with a machine-steel shaft or crank pin as the two metals will cut each other. If the pin is case-hardened, the brass will be cut. On the other hand, a soft-metal box such as of babbitt metal, will wear indefinitely. I have here in my engine room a 12x30-inch engine running at 120 revolutions per minute that has babbitt-lined bearings that have not been rebabbitted in 27 years; they are still in splendid condition. On the contrary, the crank pin of this same engine was fitted with brass boxes which have been renewed two or three times as also has the pin itself during that time. The last time that this operation was performed the engine was fitted with babbitt-lined boxes. Since then no trouble has been experienced either from heating or unsatisfactory running in any way.

If brass boxes are superior, why do the majority of the large, engine-building firms of the present time fit their engines with babbitt-lined bearings and crank-pin boxes and even babbitt-covered shoes on the crosshead?

CHARLES H. TAYLOR.

Bridgeport, Conn.

Boiler Setting

In reply to Mr. Shout's questions in the July 26 number concerning boiler-setting design, I believe that a distance of 60 inches between the grates and the shell of a 72-inch boiler is good practice. The bridgewall should be 44 inches high.

With this construction ample combustion space is assured.

GEORGE W. SHILLING.

Massillon, O.

Two Meters on the Same Line

In the August 2 issue, Mr. Blue asks for an explanation of the fact that two meters on the same line, one ahead and the other back of the pump, vary from 30 to 60 per cent. in their readings.

The variation of the meters may be due to three causes: First, both may not be calibrated so as to record the correct amount of water passing through them; second, if the meter between the pump and the tank does not register as much as the other meter it may be due to water being lost through the packing around the pump pistons; third, unless there is a fairly constant flow of water through the meters they will not register exactly alike. I recall a case some years ago where there was a meter on a line supplying a fountain. The owner, with a view toward economy, sometimes cut down the supply of water. When the flow was decreased to a certain point the meter would not register, although a small amount of water passed through it.

W. H. KELLER.

Charleston, W. Va.

Asbestos Packed Blowoff

In answer to R. G. Parker in the August 2 issue, I would like to say that I have had a good deal of experience with blowoff valves and have not found asbestos-packed cocks reliable. In the plant where I am now employed we have a jet condenser and cooling tower. As the water is used over and over again, it is only necessary to blow down the boilers once a day, but the operation of opening and closing the asbestos-packed cocks once a day caused them to leak badly within two or three weeks after being installed.

We purchased expensive valves with screw stems and brass disks and seats to replace the cocks. The seats had to be ground every time the boilers were cleaned, and in spite of this the valves would leak in a very short time. We are now using valves with removable disks made of composition. These valves give every satisfaction, and it is a short job to renew the disks.

I was once in charge of a power plant, the boilers of which were fitted with cast-iron plug cocks. These cocks were placed just inside the wall of the boiler room. One frosty morning when I went to open the blowoff cock the upper half came off in my hand, the lower half remaining in its seat imbedded in ice. I did not lose much time in replacing the top section of the valve and fastening it down with a beam until the steam pressure was reduced.

From my own experience, I would certainly recommend Mr. Parker to buy valves having a screw stem and some kind of a disk that can be renewed. It is a good idea to have two valves in service on each blowoff line.

Charles Fenwick.

Duck Lake, Sask., Can.

Settings of Return Tubular Boilers

In an editorial in the August 2 issue, under the above heading, the following statement was made: "A weak point in boiler practice is found in the boiler setting, particularly in small plants where return-tubular boilers are largely used." This attracted my attention and I set to wondering if the author realized to what extent his words were true. How many return-tubular boiler installations would we find in a week's travel that did not show external evidence of improperly built settings to say nothing of the fuel-wasting conditions which an internal inspection of the furnaces and arches would be likely to reveal.

The fact that the manner of setting a return-tubular boiler has much to do with its economy is so important that it should be every engineer's duty to see that his boilers are correctly set and the settings kept in proper condition.

The most common fault in "amateur" boiler setting is that of closing in the side walls to meet the boiler too far below the tube line and not leaving enough space between the sides of the boiler and the side wall. My practice has been to leave from 4 to 6 inches clearance whenever possible right up to the point of closing in; this point being at or a little above the tube line. Ample space around the shell affords an opportunity for the heat to circulate and is necessary for the most economical operation.

In a certain plant a boiler furnished steam to a Corliss engine, belted to a generator, which transmitted power to two motors, each having the same load. The engine and generator were newly installed but the boiler, though in good condition, was badly in need of resetting. Repeated attempts had been made to operate with full load but the boiler steadily refused to maintain the required pressure under forced firing and it became necessary to make temporary connections from one of the motors to the city lines. About this time a new engineer was employed and when an opportunity came to reset and overhaul the boiler, part of the side walls were found to have fallen in and were leaning against the sides of the boiler, cutting off much valuable heating surface. After this and other repairs had been made the boiler handled the full load without difficulty.

Another case of incompetent boiler setting cost a lumber company considerable delay and money. Two new tubular boilers of ample size were installed but failed to produce sufficient steam to handle the load, as a result part of the machinery had to be shut down several times a day to permit the fireman to catch up with the required steam. After several weeks of trouble and misdirected experiments, the space behind the bridgewall, which had been made several inches deeper owing to excavation for the foundation, was found to be entirely open. When this was properly filled, sloped and paved, no further trouble was experienced.

F. C. Holly.

Yazoo City, Miss.

Barometric Condensers

I have read with considerable interest "Notes on Barometric Condensers," by Warren H. Miller in the issue of August 2. Mr. Miller takes a "shot" at power-plant design by engineers who have not operated power plants and all condensers except barometric.

In regard to the design of power plants, my experience has led me to believe that the power house designed by one whose sole experience and education lies along operating lines is quite likely to be of a rather inefficient type, for the reason that the operating engineer, unless his experience has been of a very diversified nature, will unconsciously put into the plant that which his personal experience has proved to him will operate day in and day out, and require the least attention. The average operating engineer prefers not that which will give the best dividends on the investment but that which costs him the least work, and while I am a great believer in paying a man for what he knows how to do, I fail to see the necessity of wasting money in order to permit the operating engineer more thoroughly to digest the morning paper or color his meerschaum.

A man who devises a condenser that will suit all conditions will confer upon the engineering world a very great favor. Mr. Miller apparently takes into consideration no factors that enter into th determination of the type of condense to be used, such as water temperatur and available supply of water. Jus buy a barometric condenser, put an induction motor on the pump and watch it run This *sounds* good.

In the first place if the load is alway constant and the temperature of the wate varies but slightly, an induction motor driven pump would be good, for then the volume of water to be pumped would be constant at all times. If these idea conditions do not exist, and they seldom do, we must either arrange to pump the maximum quantity of water required when it is at its hottest temperature fo all loads and thus waste a great deal o power on light loads, or strike an average and have, say, a low vacuum with heavy loads, a fair vacuum with medium loads and a good vacuum at light loads or some other distribution.

It is very apparent, then, that some kind of a drive for the pump must be installed that will permit the quantity of the water being varied without consuming full power irrespective of load. My experience is that the best drive for a centrifugal pump, under the conditions described, is a correctly built steam turbine. A turbine is no more trouble to operate than an induction motor. The exhaust can be utilized to splendid advantage in a open feed-water heater, as it has no oil and if it is not needed to heat the feed water the turbine can be operated condensing. Such an outfit will permit of sufficient change in speed to give the quantity of water required under most conditions. Some plants have such a wide range of load that a reciprocating pump would be better even if it had to be repacked quite frequently and an oil separator had to be used on the exhaust, if an open feed-water heater was used.

Passing on, I am impressed with a mystery, it is as follows: Injection water temperature 70 degrees Fahrenheit, 300 gallons per minute discharge, temperature of the mixture, 118 degrees, vacuum, 28½ inches. I would like to inquire if this vacuum was measured with a mercury column, and if so what percentage of tin was in the mercury, or was it measured with a gage like that on one of the turbines at the St. Louis fair that registered 34 inches of vacuum all the time? A condenser that will so thoroughly intermingle the exhaust and the cooling water as to bring the temperature to within 5 degrees of the theoretical temperature of the vacuum, is doing splendidly. Most condenser builders today allow 10 per cent. for precondensation and figure to bring the mixture to within 10 degrees of the theoretical temperature of the vacuum. Therefore, if Mr. Miller's temperature of 118 degrees is correct, he probably had about 26 inches of vacuum, corresponding to 126 degrees temperature.

Mr. Miller enumerates the units in the station as two 300-kilowatt cross-compound marine engines, one 100-kilowatt simple engine and one 70-horsepower compressor. He does not give the loads that any of these carry. For the purpose of determining what is being done I think it is safe to assume that the heat in the exhaust is 1034 B.t.u. and to allow 10 per cent. for precondensation. The quantity of water required to condense one pound of steam under the given condition is

$930 \div (118 - 70) = 930 \div 48 = 19.4$ pounds.

I presume that Mr. Miller did not secure a steam consumption better than 15 pounds per indicated horsepower; therefore, to determine the load we have,

$$\frac{15 \times 19.4}{8.3 \times 60} = 300$$

gallons per minute or,

$$\frac{500 \times 300}{19.4 \times 15} = 530$$

horsepower.

Therefore, Mr. Miller's load of 1000 horsepower on a 14-inch barometric condenser, which with 70-degree injection water will normally take care of 20,000 pounds of steam at a 26-inch vacuum without an air pump, I judge to be considerably under its real capacity.

I think Mr. Miller might to advantage use a mercury column, check his figures carefully, or try an air pump, which he finds so useless. It would certainly appear that one of the best steps that Mr. Miller made was when he prepared the condenser for an air-pipe connection. As a matter of fact, when the water is hot the noncondensable vapors cannot successfully be carried away without an air pump. On the other hand, with very cold water, for instance, at one very prominent power station in Indiana they secure 28 inches of vacuum without an air pump but the injection water was at a temperature of from 40 to 45 degrees and the discharge water was from 75 to 80 degrees. Unless the conditions are very unusual it is, of course, a useless waste of money to consider a barometric condenser with a cooling tower on account of having to pump the water twice if for no other reason.

As compared with the surface condenser, if the cost of the 80 feet of 14-inch pipe, base, ells, erection, etc., were added to the cost of the barometric condenser and allowance made for the short exhaust pipe to the surface condenser the comparison of figures, I dare say, would not entirely put the surface condenser out of the running. Incidentally, I could mention a number of cylinder heads and engine frames that have been broken on account of barometric condensers flooding the engines.

I believe there are places for each of the four types of condenser now on the American market and that no hard and fast rule can be laid down as to what condenser to use. Study the conditions and use the condenser best suited to meet the requirements.

THOMAS B. WHITTED.

Charlotte, N. C.

Criticism of Piping Layout

D. L. Temple asks for a criticism of the piping layout shown in the August 2 issue and asks if it is a safe system.

In my opinion it is perfectly safe. The advisability of locating the back-pressure valve at *A* would be governed by the conditions. If all the exhaust steam from the engine is used in the heating system then the location as shown in the sketch will do; but if the exhaust is utilized only part of the time for heating and the engine exhausts into the atmosphere during the remainder of the time, then the exhaust pipe should form as short and as straight a path as possible to the atmosphere. If the location shown fulfils these conditions, it is the proper place for the back-pressure valve.

If the engine is shut down and live steam is used for heating, the 7-inch gate valve should be closed and the bleeder opened or a low-pressure trap installed to take care of the condensation.

Mr. Temple evidently went to the wrong school because he says that he has been taught that the back-pressure valve bears the same relation to the engine that the safety valve does to the boiler. A cylinder-relief valve performs the same duty as a safety valve, but a back-pressure valve is installed for the purpose of increasing the pressure to a desired point. A check valve at *B* is not necessary, the 7-inch gate being sufficient.

R. MCLAREN.

Berlin, Ont.

Installing Globe Valves

Considerable discussion has appeared in the columns of POWER upon the "Installation of Globe Valves." The article by James G. Sheridan in the August 2 issue sounds good to read, but might be the means of deceiving someone. Before discussing the proper way to install the valve, consider the design of a globe valve. The stem has from four to six threads that are engaged in the bonnet, and it is held to the disk by a counterbored nut that has four or five threads. Also, the stem has a small collar which serves to keep it from coming through the nut; this collar is not as strong as two of the threads on the stem.

As to placing the valve with the pressure above or below the seat, it makes little difference with the stress on the stem or the threads. The only difference is that, with the pressure under the seat, there is a continual load on the threads of the stem when the valve is closed; but the stem is designed to withstand this, while the nut and collar are not. By installing the valve with the pressure above the seat, all the load is thrown upon the counterbored nut and the collar when the valve is being opened. The only advantage that I can see in having the pressure above the seat is that in case the disk comes off the pressure will have a tendency to close the valve. With the pressure under the seat the valve can be packed when it is closed; this cannot be done with the pressure on top of the seat. Again, the disk is not so apt to come off with the pressure under the seat for there is a tendency to hold it in place. If a globe valve receives proper attention it will give little trouble. A stripped thread is caused not so much by the load on the disk as by the careless use of the monkey wrench.

E. F. TRACY.

Mechanicsville, N. Y.

Pump Problem

In the August 9 issue of POWER there appeared an inquiry by W. E. Thompson regarding the necessary steam pressures in the high- and low-pressure cylinders of a 10x16x8x18-inch boiler-feed pump making 35 strokes per minute against a boiler pressure of 185 pounds.

A rule which I have found to hold in practice is as follows: "Consider the total effective steam pressure (absolute admission pressure minus absolute exhaust pressure) as acting upon the high-pressure piston to balance the load on the water end; the low-pressure piston will give the speed." Applying the rule to this particular case,

$$\frac{1}{4} \pi \times 8^2 \times 185 = \frac{1}{4} \pi \times 10^2 \times \textit{steam pressure}$$

from which the effective steam pressure will be found to equal 118 pounds per square inch.

The inquiry does not state whether the pump is to run with free exhaust or condensing, so Mr. Thompson will have to settle that part of the problem for himself.

CHARLES W. LONG.

Wilkes-Barre, Penn.

New Edition of "Kent"

The communication of Donald S. Sammis, in the August 9 issue, compares the steam tables of Peabody and Marks & Davis with those in "Kent." It may be of interest to state that in the new edition of my "Mechanical Engineers' Pocketbook," which will be issued next month, the old steam tables have been replaced by new tables based upon those of Marks & Davis.

WILLIAM KENT.

Montclair, N. J.

Belgian Exposition at Brussels

By F. R. Low

A visit of short duration to the exposition revealed several features of special interest, such as the Humphrey pump exhibit, pumps made of vitrified clay, engines of special design and semi-portable units.

On the breaking up of the joint meeting of the British Institution and American Society of Mechanical Engineers at London many of the American visitors passed to Brussels, which is *en fête* for the exposition. Time did not permit the gathering of a complete description even of the power-plant features, but a quick review revealed several things of especial interest.

This exposition is not held to celebrate anything in particular, but is one of a series held at Brussels commencing with that of 1885 at which R. E. Mathot, known to our readers as an occasional contributer and as the author of several works upon internal-combustion engines, was in charge of the Machinery Hall. There was another in 1897 and the present is the third of the international order.

FIG. 1. LOOKING ACROSS THE NORTH END OF THE SUNKEN GARDENS

The main mechanical exhibit is located in the Machinery Hall, but Germany has preferred to keep her machinery as well as her other exhibits in her own section, and there are a number of water-purifying plants and other exhibits grouped about the neighborhood of the Machinery Hall. The latter building incloses the boiler room, which is set off from the main room by a glass partition, leaving the boilers and their operation open to view from the main hall without inconvenience from dust or heat. All within the boiler room proper is, however, spotlessly clean, and all who wish may enter.

Steam is furnished by ten De Nayer (Babcock & Wilcox type) boilers, each of 500 horsepower capacity and each set separately instead of in pairs or batteries as is usual with us. Underfeed stokers are used on all and the first boiler is fitted with an elaborate coal-weighing apparatus. The gases discharge through Green economizers into a reinforced concrete stack having a hight above the ground level of 230 feet, an internal diameter of 11.5 feet, and a weight of 551 tons. It appears to be built of short sections with external flutes for the reinforcement, and we hope to say more of it in a later article. The chimneys here are ornamented to a much greater extent than is seen in the United States.

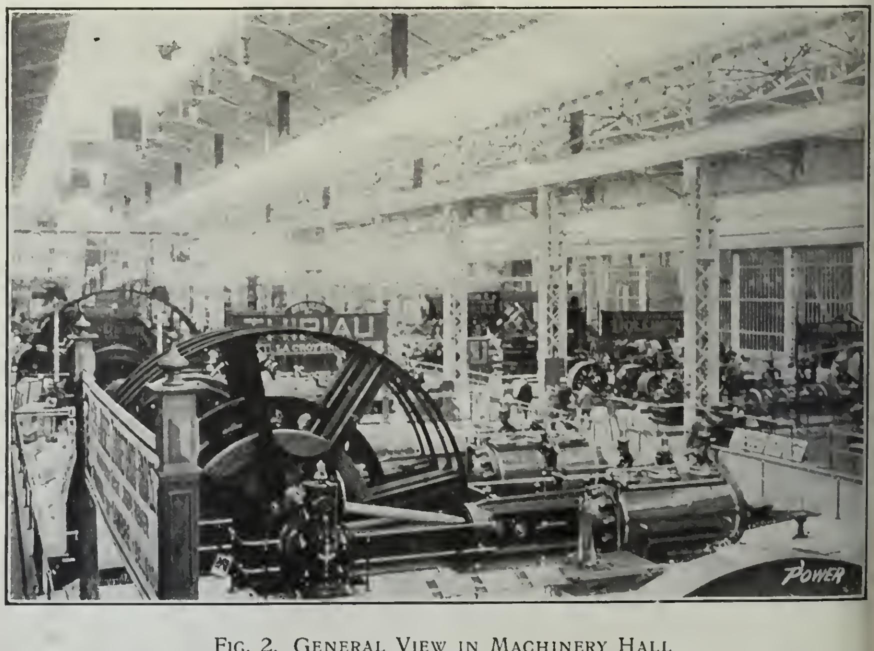


FIG. 2. GENERAL VIEW IN MACHINERY HALL

The fuel used is coal which is landed from the cars onto the ground, carried by men with wheelbarrows to a grating and dumped into the hopper of a bucket conveyer. Anything which will go through the grating the elevator will handle all right. The returning side brings out the ashes. The pressure carried is about 140 pounds.

THE HUMPHREY PUMP

One of the most attractive exhibits is a Humphrey pump in operation working against a head of a few feet simply for show purposes but capable of pumping 250,000 gallons per hour against a 39-foot head.

Pumps of this type can be made to handle a suction head of about 16 feet but it is necessary to modify the valve action so as to preserve the vacuum which would be impaired were the aspiration valve allowed to open as described later. POWER readers will remember that this is an internal-combustion pump in which the gas is exploded directly against the water. (See description in POWER of February 1, 1910, page 221). The head pumped against by direct action is limited by the mean effective pressure attainable to about the figures previously given, but may be increased to about 80 feet with an "intensifier," which is an air chamber arranged to take advantage of the forward surge.

The pump works upon the four-stroke cycle. The mixture of gas and air is exploded in the top of a vertical cylinder above a column of water which it drives out. The momentum acquired by the column carries it along allowing the gas to expand away below the atmospheric pressure, whereupon the water intake valves open and water passes into the pump. There is also drawn in through the scavenging valve a charge of air which mixes with and dilutes the expanded products of combustion. As the water, carried by its momentum above the normal level or pressure, surges back, the scavenging valve closes and the exhaust valve opens allowing the spent gases to escape. The rising column of water, however, shuts in a portion of the diluted products of the previous combustion, and compresses it until the momentum of the column is absorbed. The compressed gas sets the column into motion forward again, and into the vacuum which it leaves behind it rushes the charge of explosive mixture for the next stroke. Finally the backward surge compresses this charge and it is exploded at the commencement of a new cycle. The number of impulses per minute is governed by the mass in motion and in the present case is 9 or 10. With a greater mass the pump makes a longer stroke less frequently; with a smaller mass, *vice versa.* The aspiration stroke will be shorter than the explosion and exhaust strokes, and the compression stroke shorter still. The valves, which all open inward, are pushed open by the pressure of the atmosphere when the pressure in the cylinder becomes less than atmospheric. The movement of one valve is made, however, to lock the two others by a simple system of levers which constitute the only moving parts of the pump. The fuel consumption per water horsepower-hour is about 30 cubic feet of town gas having a heat content of 600 B.t.u., higher value. Especially in simplicity the pump offers great advantages for handling large volumes of water, as for city supplies, but Siemens Schuckert at Spandau are putting in one of considerable size to pump water to be used in turbines for power production.

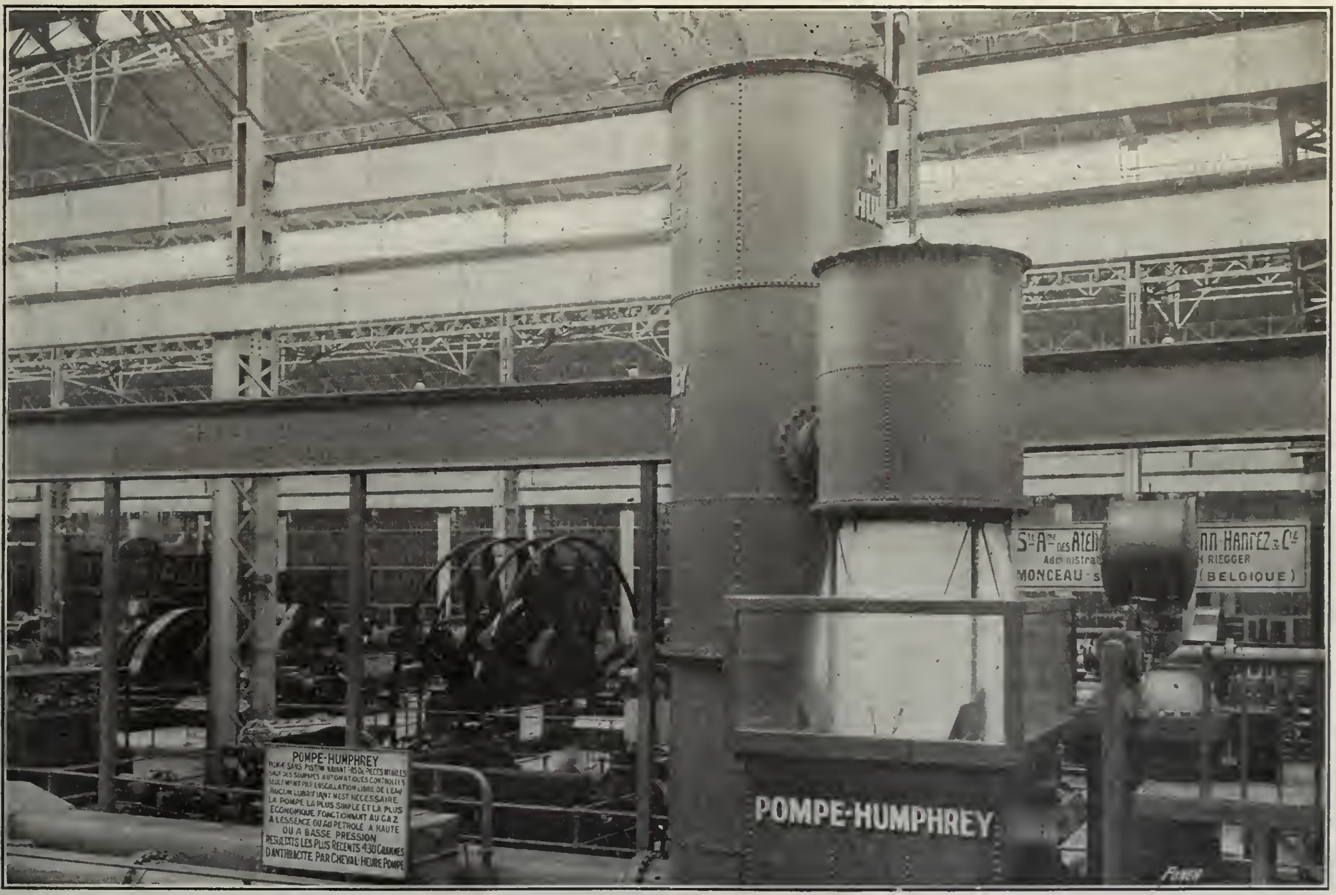

POMPE-HUMPHREY
POMPE SANS PISTON N'AYANT PAS DE PIECES MOBILES
SAUF DES SOUPAPES AUTOMATIQUES CONTROLEES
SEULEMENT PAR L'OSCILLATION LIBRE DE L'EAU
AUCUN LUBRIFIANT NEST NECESSAIRE
LA POMPE LA PLUS SIMPLE ET LA PLUS
ECONOMIQUE FONCTIONNANT AU GAZ
A LESSENCE OU AU PETROLE A HAUTE
OU A BASSE PRESSION
RESULTATS LES PLUS RECENTS 430 GRAMMES
DANTHRACITE PAR CHEVAL-HEURE POMPE
POMPE-HUMPHREY
S^te A^me des ATELI
Administrat
MONCEAU-s
NN-HANREZ & C^ie
H RIEGGER
(BELGIQUE)
Power

Fig. 3. Humphrey Pump Exhibit at Belgian Exposition

Professor Humphrey does not claim to be the first to build a pump upon this principle. There was a pump of the same type exhibited at the International Exposition at Antwerp in 1885 by Bakeljar, but in previous attempts, check valves have been used which quickly beat themselves to pieces. This is the first instance in which the kinetic energy of the water column has been absorbed entirely in useful offices while leaving the column perfectly free to vibrate between the resistances of the head and of the working fluid.

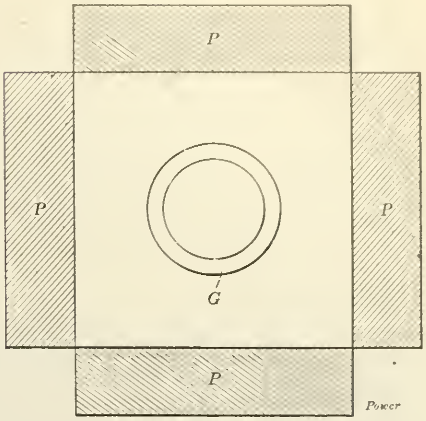


Fig. 4. Protecting Gage Glass

The absence of moving parts in contact with the fluid being pumped suggests the adaptability of the type for pumping corrosive materials or chemicals which would be troublesome in a piston pump and a number have been ordered by the Solvay Process Company at Syracuse, N. Y., perhaps for such purposes.

Pumps Made of Vitrified Clay

And this reminds me of a couple of centrifugal pumps and a piston pump made entirely of vitrified clay in one of the exhibits of the German section. The impellers are of clay also and the steel shaft which carries them is surrounded by a collar of the same material. The same company exhibits a worm for a still containing ten or a dozen coils some 30 inches in diameter made of clay tubing of perhaps 2 inches inside dimension, bent without a joint just as a copper pipe would be; plug valves or spigots, etc. They have promised to furnish photographs and further information.

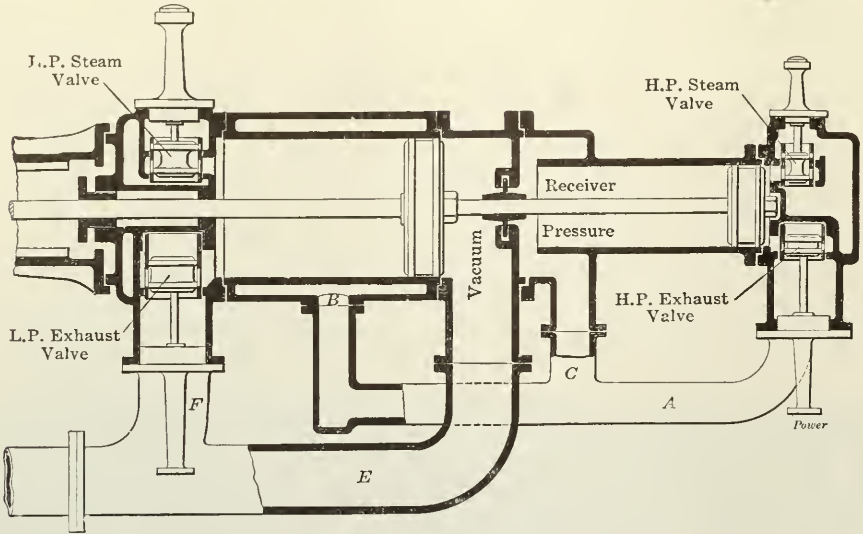


Fig. 5. Single-acting Tandem Compound

An Engine with One-half Per Cent. Clearance

One of the most interesting things in the steam-engine line is that exhibited by Ernest Mennig, successor to Walschaerts, of valve-gear fame. The exhibit consists of an engine some 18 inches in diameter with the metal cut away on the head end so as to expose the cylinder and giving a section through the head-end valves. The engine is running with steam of 140 pounds admitted to the crank end only, yet so tight is the piston that no steam escapes at the open end. All of the surfaces in the clearance are highly polished to reduce heat transfer. The clearance in this particular engine is 0.75 of 1 per cent., but upon larger cylinders it is reduced to ½ of 1 per cent. Longitudinal motion is given to the cam rods working the valves from a lay shaft lying parallel to that movement through eccentrics with universal joints in the rods. They produce an indicated horsepower with 11 pounds of dry saturated steam per hour equaling the usual performance with superheat; but if it is desired to use superheat, they can do considerably better.

Superheat

Mons. Mennig is not an enthusiast so far as superheating is concerned. He and others with whom I have talked upon the subject say that the absence of piping troubles from the use of superheat in Europe is due to the early recognition of the fact that superheating requires special attention to the piping. Weldless steel pipe is used, bent where possible to avoid the use of elbows and with branches welded in to avoid the use of tees. The joints are made with steel flanges grooved next to the pipe which is expanded in. Cast steel is used for valves.

Upon an English machine I noticed a water gage surrounded by plate glass at least a half inch in thickness arranged as in Fig. 4 to shield the operator in case of breakage of the gage glass *G*. The plate glass is indicated by *P*.

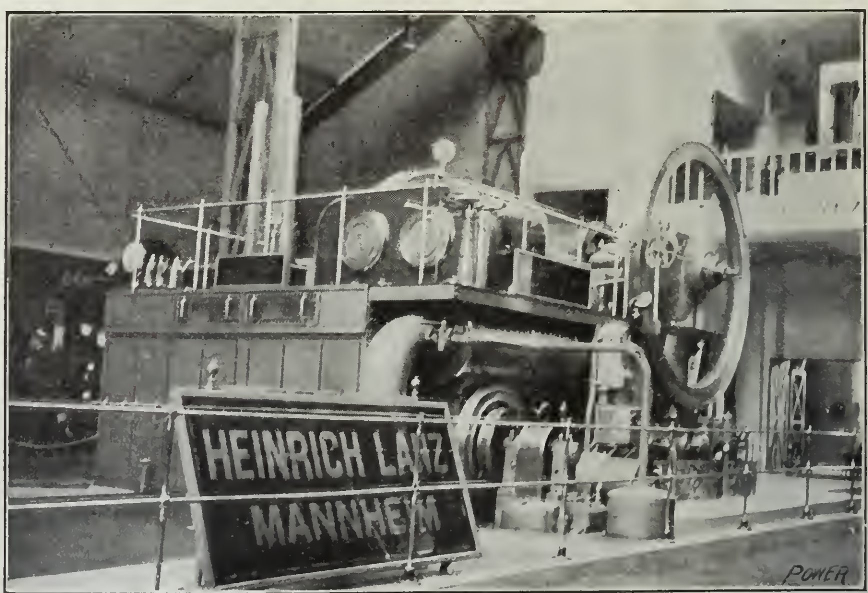


Fig. 6. Lanz Semi-portable Engine, Rated at 1000 Horsepower

A Single-acting Tandem Compound

Another novelty in the steam-engine line is the single-acting tandem compound exhibited by Van Den Kerchove, of Ghent, and shown in section in Fig. 5. The motive is to produce an engine less expensive than the ordinary compound without sacrifice of efficiency. The method by which it is ordinarily sought to do this is to simplify the valve gear and reduce the clearance of the simple engine, but this introduces trouble from excessive compression, and if considerable expansion is used the aggregate initial pressure upon the necessarily large piston is such as to require a heavy connecting rod, bearings, etc.

The method adopted by Van Den Kerchove is made apparent by the section. The steam passes from the high-pressure to the low-pressure cylinder through the pipe *A*, delivering at *B* into the jacket of the low-pressure cylinder which is in communication with the low-pressure steam chest. At *C* a branch from the high-pressure exhaust connects with the front or idle end of the high-pressure cylinder which thus serves as a receiver. The idle end of the low-pressure cylinder is connected by the pipe *E* with the condenser, although the low-pressure steam passes out through the pipe *F*. This makes an internal stuffing box necessary between the receiver and the vacuum chamber. I did not learn what precaution is taken to prevent heat transfer through the wall. For the same power a somewhat large cylinder is required than with the ordinary compound, but there are only two cylinder heads, four valves and a single cutoff. It is the number of pieces, the designer argues, and not their precise size which governs the complexity and price. There are but two stuffing boxes, and no chance for air leakage into the low-pressure cylinder. They claim to get the same low consumption with this as with their regular compound with either saturated or superheated steam. The 125-horsepower size runs at 140 revolutions per minute. The valve motion is extremely simple, using double-ported piston valves operated by eccentrics on a lay shaft working through a system of linkage which by taking advantage of centers allows the valve to dwell and to move quickly when motion is required with no disconnection.

THE SEMI-PORTABLE

Much has been said lately about the semi-portables which are used so much here and with which such surprising efficiencies are obtained. In America the only engines mounted upon boilers which we use for stationary purposes are rough-and-ready machines, put in to meet a temporary demand. Here one finds them finished like a telescope, jacketed with sheet iron or with iron panels, fitted with condenser, feed-water heater and air pump, and yet a crane hooked to the boiler would lift practically the whole outfit. Not only those for the exhibition but those for ordinary use are beautifully finished and one finds them installed in rooms with tiled floors and walls and every evidence of the intention of being put there to stay.

Why then the element of portability? It is not so much a matter of portability as of concentration. The setting of the engine upon or even into the boiler avoids piping and pipe losses. The water is heated by the outgoing gases and the pumps run from the shaft of the engine itself. The cylinders are jacketed by the boiler itself and the steam is superheated before it gets into them. There are all of the elements of high efficiency grouped together in a most simple and effective way, and the whole installation takes so little space. When will some American manufacturer take up the line?

There are a number of these about the exposition, but two in the German section are deserving of particular notice. One is the largest of the type which has ever been made, a 1000-horsepower unit by Heinrich Lanz, of Mannheim, of which a view is shown in Fig. 6. It was so clean that we thought that the steam was being furnished to it from boilers outside and that the coal in the hoppers, for it has mechanical furnaces, was part of the scenery. The overall dimensions, guessed at, are 25x27 feet in plan and 16 feet in hight above the floor.

Another semi-portable in the same room is interesting as having the central exhaust recently proposed by Professor Stumpf and described in our issue of June 21, page 1117.

The piston is as long as the stroke and uncovers, just before the stroke is completed, a central exhaust port as does that of a two-stroke cycle gas engine. It is still another method of simplification and works out nicely with a good vacuum, but since the contents of nearly the entire cylinder have to be compressed into the clearance on the return stroke the compression will be excessive if the engine is not exhausting into a good vacuum.

Cockerill, of Leraing, has on exhibition and running his first blowing engine built in 1853. It has a long vertical cylinder working with poppet valves operated by vertical rods like the side-wheel marine engine used in Eastern waters, but operated by cams on a revolving lay shaft instead of the reciprocating tappets.

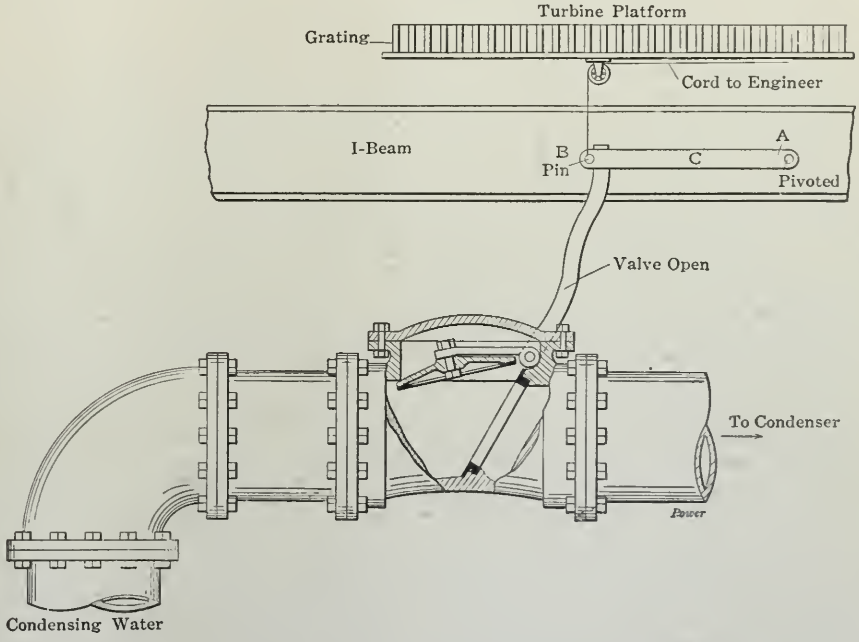
On August 14, soon after this article was written, practically the entire exposition was destroyed by fire. The conflagration is reported to have started in the Telegraph building, and the flames, driven by a high wind, spread rapidly in all directions.

An Emergency Stop Valve

A certain power plant, located in Virginia, has a novel arrangement of the supply pipe furnishing the condensing water to a jet condenser. In this instance the condensing water flowed to the condenser by gravity, but this flow was considerably increased, due to the action of the centrifugal discharge pump. Experience had shown that when the discharge pump stopped for any reason, such as the flying off of the engine-governor belt, etc., the condenser became flooded.

In order to overcome this trouble a Ludlow back-pressure valve was placed in the 20-inch condenser supply pipe to act in conjunction with a safety device arranged on the condenser proper. The illustration shows the position of this valve, also the lever attached to the valve stem for operating the valve disk. The supply pipe ran along under the grating placed around the turbine unit. A lever *A* was attached to an iron beam, and hinged as shown. A strong cord was attached to the outer end of the rod and ran over a pulley, secured to the turbine platform, to a convenient point where the engineer could pull it in case of emergency. A second cord extended to within reach of the engine-room floor in close proximity to the discharge pump.

The valve lever was so designed that it swung close to the iron beam when the valve was in an open position, and was held in place by means of a pin *B*, secured to the lever *C*. Thus the valve-stem lever cannot drop until the pin is moved sufficiently to allow it to pass over the end of the lever.



SECTIONAL VIEW OF STOP VALVE, ALSO LEVER ARRANGEMENT

If the centrifugal pump should fail, the engineer, by pulling the cord, can lift the lever and pin *C*. The weight of the valve disk will cause it to drop, and the water will then hold it to its seat. By this arrangement and quick work on the part of the engineer the supply of condensing water can be quickly and effectually cut off from the condenser.

Inquiries of General Interest

Prony Brake

When a prony brake has an arm of 6 feet in length, the engine runs at 200 revolutions per minute and the brake is balanced by a weight of 14 pounds, what is the horsepower of the engine?

D. B. D.

The power absorbed by a prony brake is found by multiplying together the numbers denoting the circumference of a circle whose diameter is twice the length of the lever, the number of revolutions made by the engine, and the pressure exerted at the end of the lever, and dividing this product by 33,000. In this case the power developed by the engine would be

$$\frac{37.7 \times 200 \times 14}{33{,}000} = 3.1957$$

Area of Safety Valve

A boiler generates 14,000 pounds of steam per hour at 85 pounds gage pressure. What should be the area of the safety valve?

J. K. L.

The Massachusetts formula for safety-valve areas is

$$A = \frac{W \times 70}{P} \times 11$$

in which

A = Area of safety in square inches per square foot of grate surface,

W = Pounds of water evaporated per second per square foot of grate surface,

P = Absolute pressure at which steam is generated.

If the total grate area be represented by g and the total valve area by a, then

$$\frac{a}{g} = \frac{W \times 70}{g \times P} \times 11$$

whence by cancelation

$$a = \frac{W \times 70}{P} \times 11$$

As 14,000 pounds of steam per hour is at the rate of 3.89 pounds per second, if numerical values are substituted in the equation, it will read

$$\frac{3.89 \times 70}{100} \times 11 = 29.95$$

or 30 square inches as the valve area required. This would be supplied by two valves of 4½ inches diameter.

Crown Bars

What are the crown bars and crown-sheet stays, and what duties do they perform?

J. D. B.

Crown bars are bars or girders extending across the firebox and having feet at the ends which rest upon the side sheets. A common form of crown bar is a two-piece bar bolted together so that the crown bolts can pass between, with the heads on the firebox side and having nuts and washers on the top of the bars. Crown-sheet stays perform the same duty as the crown bars but differ in that crown-sheet stays, when of the radial stay type, consist of long bolts like staybolts, passing through the outer sheet of the boiler and the crown sheet, riveted over at both ends. Another form of crown-sheet stay consists of "T" bars riveted to the crown sheet and connected to the outside sheet by means of sling stays.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Boiler Efficiency

In the report of a boiler test it was stated 11,500 heat units evaporated nine pounds of water from and at 212 degrees, giving an efficiency of 75½ per cent. How is the efficiency of a boiler found?

J. A. L.

Since it requires 970.4 heat units to evaporate one pound of water at 212 degrees into steam of the same temperature it will require

$$9 \times 970.4 = 8733.6$$

heat units to evaporate nine pounds. As 11,500 heat units were used in the case mentioned the per cent. of heat realized is found by dividing 8733.6 by 11,500 and multiplying by 100, which gives 76 as the efficiency.

Valve Events

Please explain where admission, cutoff, release and compression take place, and what is the meaning of these terms?

B. C. A.

Admission is when the valve admits steam, or opens the port; cutoff takes place when the valve cuts off the admission of steam; release, when the exhaust port opens, and compression when the exhaust port closes, retaining what steam is left in the cylinder, where it is compressed by the action of the piston. The particular points at which these different events take place are governed by the lap of the valve and the manner in which the valve is set. The points at which admission, release and compression take place are usually given in terms of the piston travel or fraction of the stroke.

Horsepower Developed by Engine

In stationary practice, will an engine driving a dynamo develop more horsepower through the dynamo than the horsepower of the engine? That is, will the electric current have a greater horsepower than the engine?

A. D. H.

An engine driving a dynamo cannot develop more horsepower through the dynamo than the engine itself develops; that is, the horsepower developed by the dynamo will not be any greater than the original horsepower developed by the engine; in fact, the dynamo will not deliver quite as much power as is developed by the engine, owing to the loss by friction in the engine and the heat losses in the dynamo.

Raising Boiler Pressure to Carry Load

If an engine with 45 pounds steam pressure and 26 inches of vacuum cuts off at quarter stroke, how much would the boiler pressure have to be raised to have the engine carry the same load with two pounds back pressure and still cut off at quarter stroke?

F. A. W.

The mean pressure of expanding steam at one-quarter cutoff is practically 0.6 of the initial pressure

$$0.6 \times 60 = 36$$
$$36 - 2 = 34,$$

which is the mean effective pressure in the cylinder with 45 pounds gage pressure, 26 inches vacuum and one-quarter cutoff. To do the same work against two pounds gage back pressure and cut off at quarter stroke would require a mean effective pressure of

$$34 + 17 = 51$$

pounds absolute;

$$51 = 0.6 \times 85$$

the absolute pressure necessary for a mean effective pressure of 34 pounds at one-quarter cutoff and two pounds gage back pressure. The boiler pressure required is

$$85 - 15 = 70.$$

Change of Connecting Rods

In case the connecting rod on the low-pressure side of a cross-compound engine should break, could the rod from the high-pressure side be used in the place of it?

T. C.

As both sides of standard cross-compound engines are designed for equal loads the pins and rods are of the same dimensions and are interchangeable.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T McKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for July, 1910, *was* 138,000.

August 2	34,000
August 9	34,000
August 16	34,000
August 23	34,000
August 30	34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

Boiler Inspection

Articles on boiler inspection appear from time to time in technical journals, and nearly all of them deplore the lack of intelligent supervision of steam boilers, and all advocate the passage of stringent laws by the municipality or the State. While admitting the need of proper laws relating to boiler supervision, we doubt that very much improvement in the operation of steam boilers would result from the passage of such laws if they are to be like most of those now on the statute books, and administered in the same manner. Most of our present laws (always excepting those of Massachusetts) are of little value because they do not recognize the two essentials that the boiler inspector should be under civil-service rules and that the steam user should be fined when an accident happens. Of course, no one would advocate fining a steam user if an impartial investigation showed that he was absolutely without blame, but he should at least pay the cost of the investigation, and we believe that such procedure would greatly decrease the number of "unavoidable" accidents. If you cannot determine mathematically just how this can be, turn to your textbook on human nature.

As it is now, a city or State boiler inspector's job is a political one, and is filled without that regard for his fitness for the office which characterizes the filling of responsible positions in the commercial world. The work, at the best, is hot, dirty and disagreeable, and the natural attitude of a hundred-dollar-a-month man is to shirk such work; when there is added to this natural attitude the fact that no matter how well he does his work he is likely to lose his position with a change of administration, and that no matter how poorly he inspects he will not lose his job until his party is turned out, one has a combination productive of a mental attitude that may be best characterized by the expression: "What's the use?"

The conditions cited account for the fact that the municipal and State boiler inspectors are usually below the standard of insurance inspectors and seldom enjoy the confidence of the steam users. Moreover, few capable boiler inspectors are appointed to the position of boiler inspector, because there are few men with adequate training and those few have good permanent jobs which they are unwilling to change for precarious political berths. Appointing a man to the position of boiler inspector does not make him one. It takes considerable experience in actually inspecting boilers to make a boiler inspector, and there is no greater fallacy than the current belief that a good boilermaker must, of necessity, be a good boiler inspector. As a general thing, the training of a boilermaker is inferior to that of an operating engineer for the making of a good boiler inspector.

The moral responsibility of the steam user is well recognized in England. In talking with an Englishman recently concerning a recent boiler explosion in this country which killed seven men, he asked what was done to the owner of the plant. Upon being told that the owner was not molested or criticized in any way, he said: "Do you mean to tell me that a man in this country may blow up a boiler and kill seven people, and no consequences whatever be visited upon him for that act?" That this is a privilege of every American citizen had to be admitted. The Englishman's surprise was natural. In his country they hold the steam users accountable for all steam-plant accidents that occur. For example, in the south of England some time ago the head of a thermal storage tank let go in the flange, killing two men and doing considerable property damage. The crack resulted from a strain in the metal of the flange caused by a hydraulic riveter which was used as a press to force the head flange and the shell plates together, which, of course, was done cold. The commission of the British Board of Trade which sat in the case assessed a fine of two thousand dollars against the company that built the drum; the manager of that company was fined two hundred and fifty dollars for letting the work go out of the shop; the chief engineer of the power plant was fined two hundred and fifty dollars, presumably for letting the tank come into the plant, and the resident engineer of the power plant was fined one hundred and fifty dollars for not cutting out the tank when leakage at the knuckle of the flange was first noticed.

We know of no better way to bring a man to a sense of his moral responsibility to the community than to touch his pocketbook. We know of no better way to reward a man for efficient work than to assure him that he may retain his position without fear or favor as long as that efficiency remains unimpaired. The logical deduction, therefore, is that in order to be efficacious any legislation enacted with the object of protecting the public against power-plant accidents should embody the two primary features of fines for the executives and owners and civil-service classification of the inspectors.

Verdict on Canton Explosion

The verdict of Coroner March, of Canton, O., which has just been made public, places the blame for the boiler explosion at the American Sheet and Tin Plate Company's plant, which occurred on May 17 and in which thirteen men were killed and many hurt, upon William Austin, the dead fireman, whose body was blown a distance of several hundred feet through a house, as reported in our issue of May 31.

In effect the coroner finds that Austin connected boiler No. 2, the one that exploded, with the rest of the boilers when the pressure in it was too low with reference to the other boilers on the line, thus causing water hammer, or sudden shock.

In making public his opinion the coroner says in part: "Two possible conditions could have caused the explosion. The first a condition of low water with a red-hot boiler above the water line and the sudden turning into the boiler of a large supply of comparatively cold water.

"The second is a condition known to boiler men as a water hammer, which is caused when a boiler carrying low pressure is too rapidly connected with or 'cut into' a battery of boilers of comparatively high pressure.

"Weighing the two causes as carefully as one without technical education might be expected to do, I have come to the conclusion that the explosion was due to water hammer, boiler No. 2 being cut into the battery of boilers by William Austin, whose body was thrown several hundred feet through the air, driving through a house and crushing through a picket fence beyond the house.

"I can think of no other cause for Mr. Austin's body having been thrown as it was with such force except that he was on top of the boiler when it let go, the exact position he would have been in if engaged in cutting in the boiler. Had he been turning water into a superheated boiler he would not have been thrown as he was, but would have been crushed to the ground, as in manipulating the feed valve he would have been standing in front and at the side of the boiler in an unfavorable position for being thrown such a distance."

The corner's investigations have established the fact that the stop valve was found two-thirds open, indicating that the boiler had been partially connected to the others when the explosion occurred. The valve had been damaged so that it could not be moved either way, remaining a mute witness to the cause of the accident. In reaching his conclusion the coroner has totally disregarded the testimony of Louis C. Hughes, who was injured in the accident, but whose life was saved by his temporary absence from the boiler room. He has also disregarded Hughes' suggestion that the accident might have been due to explosives placed in the coal by enemies of the company or by enemies of the employees of the mill. All evidence indicated that the force came from within the boiler and not from the firebox, as would have been the case had explosives been placed in the coal.

The coroner, since the explosion, has been in receipt of many letters from boiler men tending to corroborate the opinion that water hammer was the cause of the accident and citing the fact that sheared rivets, with plates intact as in this case, could be taken as a sign of a sound boiler. In making public the decision he acknowledges the assistance rendered him by the representative of POWER AND THE ENGINEER, who was on the ground immediately after the explosion, and by Edward Whelan, the local boilermaker, who testified at the inquest.

Acting on Good Advice

It is common practice for firms putting goods on the market to urge quality and make price of secondary consideration. This is really as it should be, as an article of first-class quality will demonstrate its value, thereby justifying the slight additional cost over the purchase price of an inferior product. Manufacturing establishments have built up on this principle a large volume of business, and the customers have been perfectly satisfied. When it comes to the internal arrangements of the manufacturing establishment, however, this policy is not always adhered to, the firm often scheming along with the cheapest of supplies and equipment, living as it were from hand to mouth behind the scenes while making a first-class showing in the open market.

Nobody feels this sooner than the chief engineer of the establishment. He is continually planning makeshifts in the effort to keep the machinery in motion and turn out the manufactured product with the least temporary expense, knowing that the policy of the company will not allow him to make those permanent improvements which in the end would lower the cost of production and put the plant on a better and higher mechanical plane. Even in the class of operatives such companies sometimes make the mistake of contenting themselves with low-grade men in important positions.

Not long ago the superintendent of such a plant approached a steam expert on the question of power-plant economy. His power was costing too much. There was an enormous waste in this department which it was impossible to detect. Results for the amount of expenditure were entirely out of all proportion and he invited the expert to look over his plant and report.

The result was a recommendation that he hire a high-class, high-priced operating engineer to take charge of the plant, giving him absolute authority in the plant. It was recommended that a salary of from $2500 to $5000 a year be paid to this man, and the belief was expressed that results would justify the expenditure.

Wonderful to relate, the advice in this case was followed; a $2500-a-year man was secured and set to work in the power plant. This man studied the case in all its phases, surrounding each particular power-generating equipment with the conditions most favorable to its efficient operation in the same manner that a physician would study the case of a patient whom he had removed from the slums and placed in a first-class hospital where every facility for combating disease was afforded. The flue gases were analyzed, the air leaks were stopped, the pipes were covered, the grate surface was reduced, the rate of combustion was increased and the ratio of grate to heating surface was readjusted. The drips were caught and utilized, the feed-water problem was attacked, the lubrication question was carefully considered, the personnel of the plant was studied carefully and everything bearing on the situation had the attention which its importance deserved. Two or three months passed by and the high-priced man had already saved his year's salary. As further improvements were instituted, the economies became more apparent and the firm voluntarily raised the salary of their engineer, knowing from past experience that his services were of sufficient value to justify the action. This was a case where competent advice was followed. In how many other cases is advice which should be followed ignored, simply through the reluctance of manufacturers to spend a few extra dollars to insure that their power plant and mechanical equipment will be placed on a dividend-paying basis? In many cases the power department is situated so that it can make or break a company, this being one of the largest items of expense of the organization. It would, therefore, appear that the very best advice should be obtained in the operation of this department and a man of high ability engaged to see that the operating details are carried out in proper manner.

Ottawa has a peat bog at her doors, estimated to contain over 5,000,000 tons of fuel, and three others within about 40 miles, aggregating over 20,000,000 tons.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

De Laval Centrifugal Pump Construction

The accompanying illustrations show that the De Laval pump has a relatively small diameter of the impeller, which implies high rotative speed, and is therefore well adapted to be operated by steam turbines or electric motors.

The material used for the impeller is Government bronze, consisting of 88 per cent. copper, 10 per cent. tin and 2 per cent. zinc.

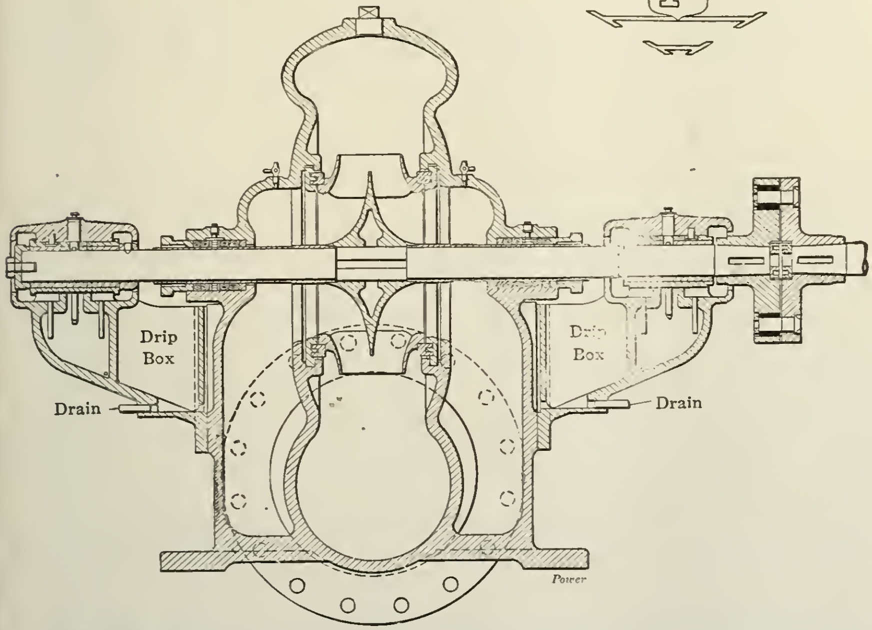


Fig. 1. Sectional View of Single-stage Double-suction Centrifugal Pump

The casings of the pumps, both single-stage and multistage, Figs. 2 and 6, are formed of two parts only; that is, they are split horizontally on the center line of the shaft and both the suction and discharge openings are in the lower half, or pump case. The top half, or pump-case cover, can be lifted off, after which the impeller and all passages are accessible, and the shaft and attached parts can be removed without disturbing the piping or connections or any other part of the machine. As the pump case and the cover are each solid castings, there is but one joint in the casing of the pump, a plain, flat surface, scraped to a fit and maintained tight by an oiled-paper gasket. Fig. 1 shows a single-stage double-suction centrifugal pump.

Cast iron is used for the casings of all De Laval pumps intended to handle noncorrosive liquids or fresh water. If corrosive or abrasive liquids are to be handled, other metals are used, such as acid-resisting bronze, nickel alloys or steel. The fitted surfaces of the cases are machined and the two parts are provided with taper dowels to insure correct assembling, even by unskilled labor. Portions of the casing subject to wear are machined to gage and fitted with removable wearing pieces, which, in effect, render the casing practically indestructible and make it possible to retain high efficiency indefinitely. One of the most important of these wearing parts is the pump-case protecting ring, in which is formed the labyrinth packing, preventing leakage from the discharge back to the suction. This joint is made by the combination of a removable ring attached to the pump casing and another removable ring attached to the revolving impeller, Figs. 3 and 4. Each ring is made to standard gage and can be replaced without fitting. The intermeshing grooves form a labyrinth through which any leakage from the discharge to the suction side of the impeller must find its way. Due to the use of these rings De Laval pumps can be built to successfully pump 500 feet or more per stage without suffering appreciably from slip. This type of packing, furthermore, permits of larger clearance, both axially and radially, without excessive leakage. The path of the water through the rings is tortuous, and the loss of head at each turn destroys head and prevents rapid flow. The water in the labyrinth is also set in rotation, creating a counter

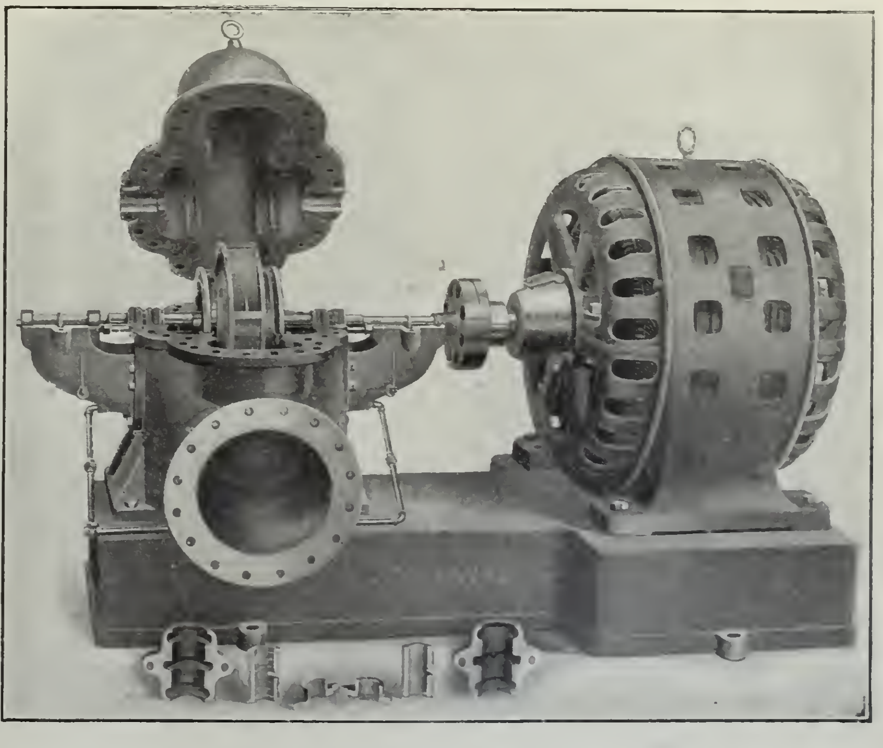


Fig. 2. De Laval Single-stage Motor-driven Pump

centrifugal force opposing the leakage. The rings are made of bronze.

Leakage of water out of the pump or of air into the pump along the shaft is prevented by packed glands containing soft packing under pressure, with a hollow skeleton ring in the middle of the packing, to which water under pressure is admitted, forming an air-tight seal, Figs. 5 and 7. The glands are of bronze and are split horizontally to allow their removal without disturbing other parts of the machine. The split gland, when removed, leaves ample room for the renewal of packing. By means of take-up studs the follower can be drawn up tightly against the packing when the pump is running. These studs are hinged, so that, by slightly slacking the nuts, the studs can be thrown out of the way for the removal of the gland. Large drip boxes, piped to a common drain, take care of leakage.

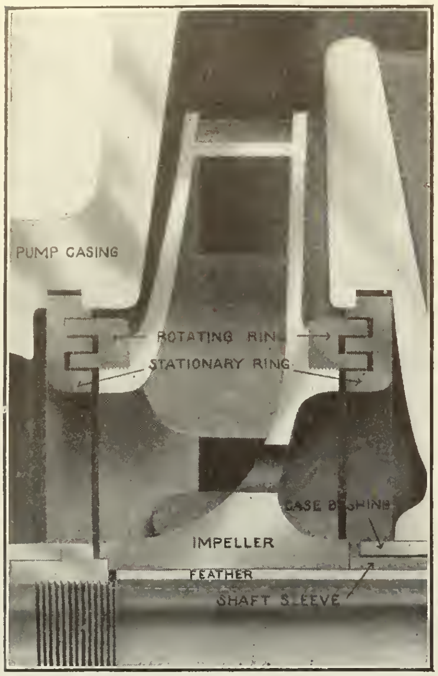


Fig. 3. Longitudinal Section

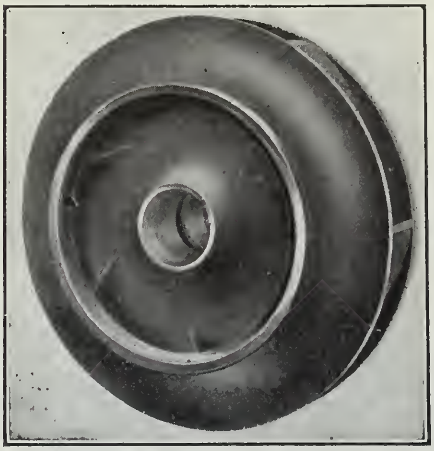


Fig. 4. Impeller

The shaft at this point is protected by a special bronze sleeve which extends inward to the impeller. Besides protecting the shaft from corrosion by the water, this bronze sleeve forms a bearing surface for the packing. Thus, should any scoring take place, it is received by the sleeve, which can be replaced.

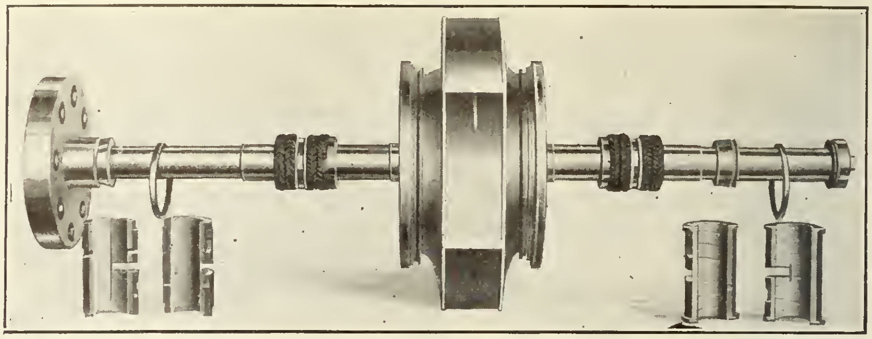


Fig. 5. Single-stage Double-suction Impeller and Shaft

The shafts are made from special hammer-forged steel, and are protected from contact with the liquid handled by the pump, and wear in the stuffing boxes, by bronze sleeves.

The bearings are of the vertically split, babbitt-lined, ring-oiled type. They are formed in separate shells resting in the brackets or pedestals. The latter are separate and distinct from the pump casing and stuffing boxes, and there is no possibility of water leaking past the stuffing boxes and finding its way into the bearings and oil reservoirs.

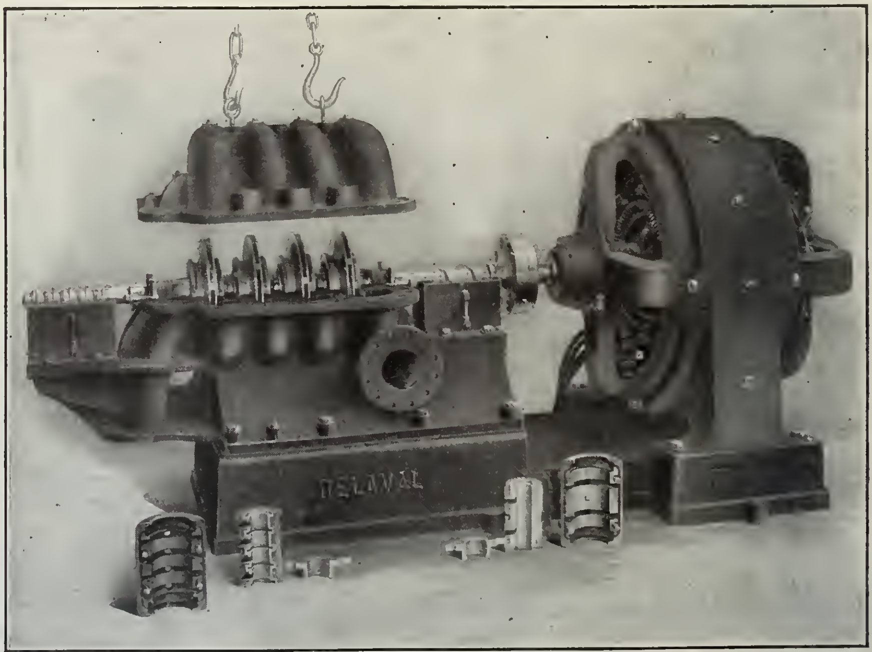


Fig. 6. Multistage Centrifugal Pump

The separate pedestal also permits of the removal of the bearings for inspection without disturbing the stuffing boxes, pump-case cover or other parts of the pump. The babbitted bearing shells are machined and ground on dead-center grinders and form a neat fit with the bearing bracket. This provides a quick and simple method of renewing the bearings, and also insures interchangeability. The vertically split type of bearing is easily removed, as it is only necessary to revolve the bearing about the shaft until each half is on top and free.

All bearings are supplied with special bronze oil rings of sufficient number and proper size to insure excellent lubrication. These rings dip in oil reservoirs, located in the brackets and large enough to insure freedom from agitation and to allow impurities in the oil to settle to the bottom. The reservoirs are equipped with suitable gage glasses and filling and drainage cocks. To prevent the creeping of oil along the shaft, from the bearings, the shaft is provided with grooves, commonly known as "oil slingers."

The preceding description of the single-stage type of pump applies in a large degree to the multistage pump. A multistage pump is in reality formed of several single-stage pumps arranged in series, the discharge from one stage being led into the inlet of the next impeller. Thus, the pressures developed in the several stages are added together and the

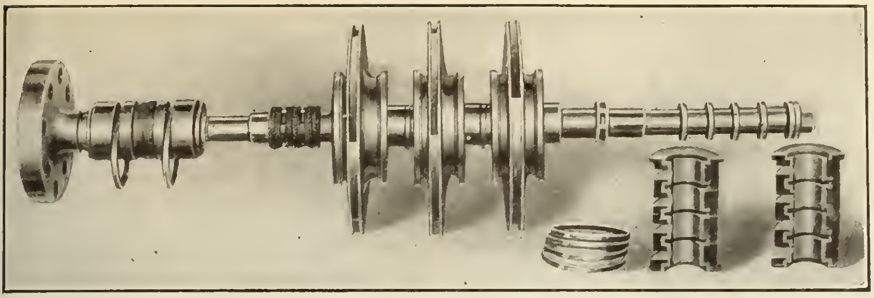


FIG. 7. SHOWING IMPELLER, SHAFT, FLANGE OF FLEXIBLE COUPLING AND THRUST BEARINGS

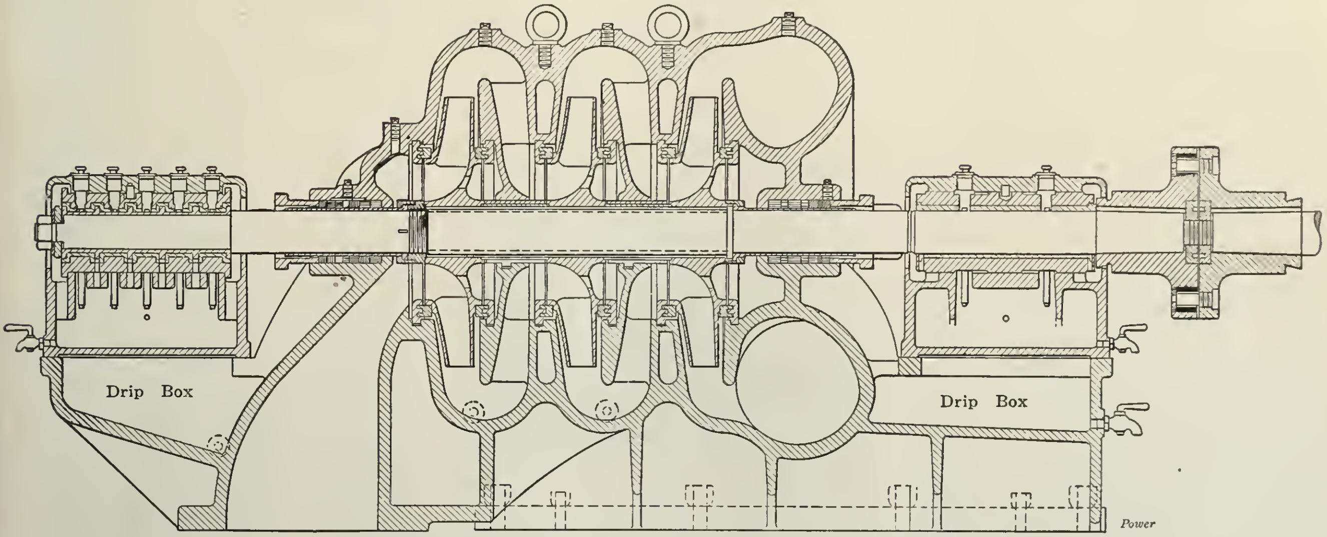


FIG. 8. SECTIONAL VIEW OF DE LAVAL MULTISTAGE CENTRIFUGAL PUMP

total head developed is equal to the sum of the heads generated by the individual stages.

The casing is made of two castings only, the passages from the delivery chamber of one impeller to the suction of the next being formed permanently in the solid casting.

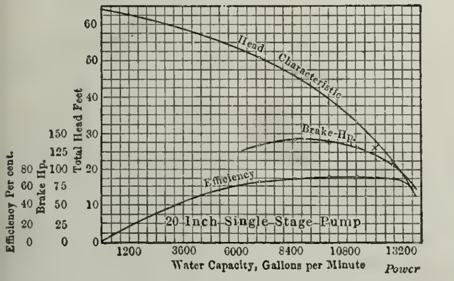


FIG. 9. LARGE MOTOR-DRIVEN PUMP, LOW-HEAD SERVICE

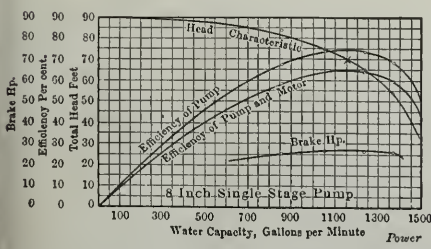


FIG. 10. SMALL MOTOR-DRIVEN SERVICE PUMP, MODERATE HEAD

The impeller of the multistage pump is of the inclosed type, and may be regarded as halves of double-suction impellers. Water enters the double-suction impeller axially from the two sides and the two currents of water join after having been turned into a radial direction; that is, two single-suction impellers, as shown in Fig. 8, put together back to back and the dividing wall between the impellers thinned out to nothing toward the periphery, would become a double-suction impeller as used in the single-stage pump.

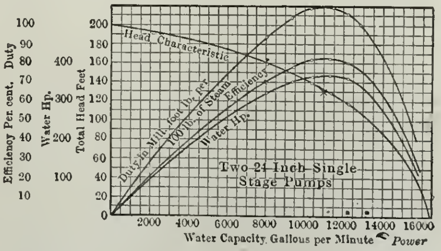


FIG. 11. LARGE TURBINE-DRIVEN GENERAL-SERVICE PUMP

A double-suction impeller in a multistage pump is not advisable, as it would involve complicated passages. It is possible, moreover, to balance a single-suction impeller by duplicating the conditions of pressure on both sides of the web of the impeller, by providing a balancing chamber on the back of the impeller, having the same cross-sectional area as the suction chamber, being inclosed by labyrinth rings of the same diameter as those encircling the suction opening. The impeller is pierced by a number of holes connecting the balancing chamber with the suction chamber, so that the static pressure in the suction chamber shall be transmitted to the balancing chamber, also that an extra pressure shall be generated in the balancing chamber approximately equal to the reaction of the axial column of water entering the suction of the impeller.

To provide for possible unbalanced end thrust, the multistage pump is provided with babbitt-lined, ring-oiled thrust bearings of the marine type forming the supporting bearing.

As compared with the single-stage pump, the multistage pump contains an additional water joint where the rotating shaft passes through the diaphragms sep-

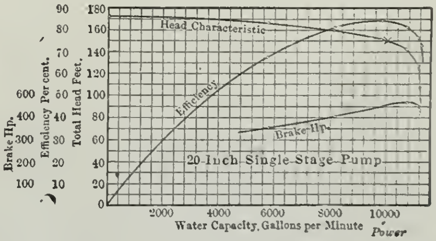


FIG. 12. MOTOR-DRIVEN SERVICE PUMP

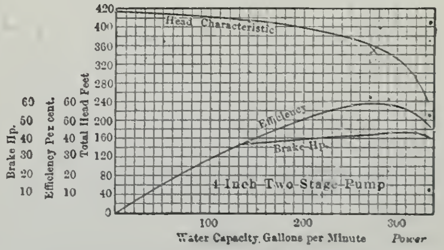


FIG. 13. TURBINE-DRIVEN BOILER-FEED PUMP

arating the several stages. This joint is formed between a protecting bronze sleeve keyed to the revolving shaft and a similar sleeve held by the diaphragm. It is long, with small clearance, but there is no rubbing contact and wear does not occur.

The general design and details of construction of De Laval vertical pumps are similar to those of the horizontal type. The pump case is split vertically in a plane passing through the center of the shaft, and the suction and discharge openings are in the main pump-case casting, thus allowing for the inspection and replacing of any of the working parts by removing the pump-case cover. In some cases the vertical type of machine is furnished with a single suction impeller where the conditions are more suitable for this type. A lower guide bearing is used in order to avoid the use of an overhung impeller which is liable to produce trouble by its tendency to vibrate and destroy the upper bearing. Both bearings are so arranged that they can be inspected while the pump is in operation, and can be removed without disturbing any part of the pump other than the bearing cap. In connection with the upper bearing, there is a thrust bearing to carry the weight of the running parts of the pump.

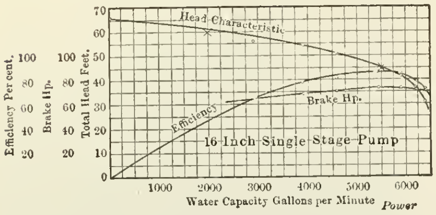
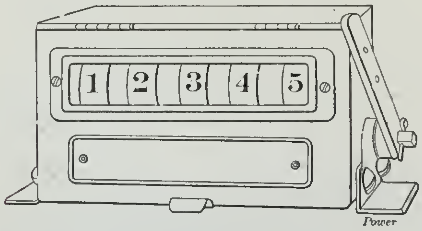


FIG. 14. CIRCULATING PUMP FOR BAROMETRIC CONDENSER

The diagrams, Figs. 9 to 14 inclusive, show characteristic curves of De Laval centrifugal pumps of several sizes designed for different classes of service. The three curves shown in each diagram indicate the head developed, the power required and the mechanical efficiency obtained through the range of output of the pump.

The Model B Durant Counter

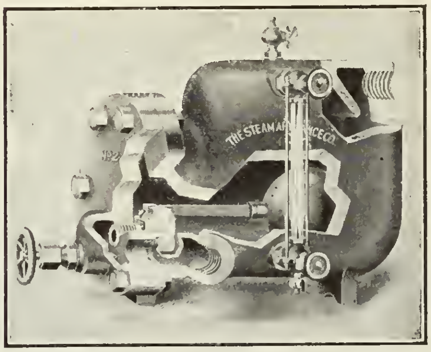
The counter illustrated herewith is built by the W. N. Durant Company, Milwaukee, Wis. It is of the full-geared type, without springs, and is provided with a quick-resetting device which leaves the gears constantly in mesh. It is 4 inches long by 2½ inches high. It can be operated on a very short stroke and is accurate at high speeds.



MODEL "B" DURANT COUNTER

Peters Corliss Valve Steam Trap

The essential features of this trap are shown herewith. In operation, water enters at the upper part of the trap against a baffle, which deflects the current down the inside walls. Water therefore cannot fall on the float and subject the latter to continued oscillation and excessive wear from this cause. Sediment is washed by the current to the opposite end of the trap into a sediment chamber, the latter being drained by a valve opening to the atmosphere. The working valve of the trap is placed above this sediment chamber, thus eliminating the wear which the sediment might have on the valve. A bypass is placed in the cover, permitting a discharge independent of the valve opening, and a water glass is fitted to show the water level.



PETERS CORLISS VALVE STEAM TRAP

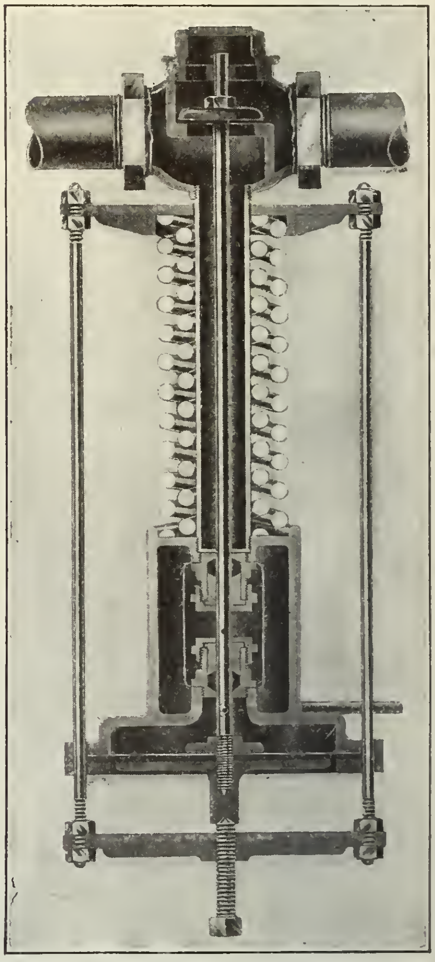
The feature of the trap is in the valve, which is of the Corliss or semi-rotary type and consists of but three parts, the body, the valve and the T handle lever to which is attached a heavy seamless copper float.

This trap has a continuous flow only at maximum capacities, or where the inflow is equal to or greater than the outflow, which latter condition sometimes occurs. Its normal operation is not to open as the water rises, but the valve opening is delayed until a sufficient amount of water has accumulated, when the pressure of steam against the valve is overcome by the action of the water on the float. Then the trap opens wide suddenly. When the valve closes, the same conditions exist and a sudden closing is obtained, which prevents wire drawing. The water level is always maintained above the valve, preventing loss of steam. Full opening and quick closing are two valuable features which result in a permanently tight valve under long continued service. This trap is being manufactured by the Steam Appliance Company, Milwaukee, Wis.

The Boardman Pump Governor

The Locke Regulator Company, Salem, Mass., has perfected an improved governor which is recommended by the Factory Insurance Association of Hartford, Conn., to be used in connection with Underwriters' steam fire pumps controlled by an auxiliary system.

This governor has been designed for the special purpose of meeting the requirements of fire-pump service and embraces the following new features:



BOARDMAN PUMP GOVERNOR

It has a single seated valve instead of a double balance valve, with the valve so constructed that it may be renewed at any time without removing the governor from the pipe.

The governor is constructed with an independent water chamber which is entirely separate from the steam. The water chamber being constantly under pressure from the sprinkler system keeps the diaphragm cool and protects it from the heat that would otherwise destroy it. The general arrangement of the springs and all the parts is shown in the accompanying illustration.

The desired pressure can be obtained by setting the screw, shown at the bottom, which governs the tension of the springs.

SOCIETY NOTES

The combined National Association of Stationary Engineers associations of Philadelphia, Nos. 1, 9, 12 and 20 had a summer outing on Saturday afternoon and evening, August 20. There were about 800 engineers and their wives present. A feature of the afternoon's entertainment was a ball game between associations Nos. 1 and 20, the former winning out by a score of 6 to 2. Supper was served to all and dancing until 12 p.m. filled out a program which was thoroughly enjoyed by all.

BOOKS RECEIVED

SOLENOIDS. By Charles R. Underhill. D. Van Nostrand Company, New York. Cloth; 342 pages, 5x7½ inches; 223 illustrations. Price, $2.

ELECTRIC MOTORS. By Francis B. Crocker and Morton Arendt. D. Van Nostrand Company, New York. Cloth; 291 pages, 6x9 inches; 158 illustrations. Price, $2.50.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

ROTARY ENGINE. William H. Thompson, Jr., St. Thomas, N. D., assignor to International Roturbine Engine Company of America, Fargo, N. D. 966,622.

ROTARY ENGINE. George Walter Brown, Ledgedale, Penn. 966,648.

MOTOR. George E. W. Luehrmann, St. Louis, Mo., assignor to Wayne Manufacturing Company, St. Louis, Mo., a Corporation of Missouri. 966.757.

INTERNAL COMBUSTION MOTOR OR ENGINE. Herbert William Rowing and Victor Watson Riley, London, England. 966,953.

INTERNAL COMBUSTION ENGINE. Joseph A. Nickelson, New York, N. Y., assignor to New York Yacht, Launch and Engine Company, Morris Heights, N. Y., a Corporation of New York. 966,948.

INTERNAL COMBUSTION ENGINE. Sigurd Wiebe, New York, N. Y., assignor of one-half to John C. Quinn, Philadelphia, Penn. 966,972.

EXPLOSIVE ENGINE. George A. Parker, Hope Valley, R. I., assignor of one-half to Helen Gertrude Simmons, Hope Valley, R. I. 967,046.

INTERNAL COMBUSTION ENGINE. Percy V. Woodward, London, Ontario, Canada, assignor of one-half to Albert W. Woodward, London, Canada. 967.097.

ROTARY ENGINE. Ernest P. Dargin, Denver, Colo., assignor to Mary E. Dargin, Denver, Colo. 967.108.

CURRENT WHEEL. John W. Ziegler, Spokane, Wash., assignor of one-half to L. W. Willett, Spokane, Wash. 966,511.

BOILERS, FURNACES AND GAS PRODUCERS

BOILER. Peter A. Deasy, Oakland, Cal. 966,532.

MECHANICAL STOKER. David F. Hervey, Logansport, Ind. 966,547.

LIQUID FUEL BURNER. Walter R. Montgomery, Shreveport, La. 966,576.

BURNER FOR LIQUID FUEL. Carl W. Weiss, New York, N. Y. 966,628.

GAS PRODUCER. Walter Thomas, Vancouver, British Columbia, Canada. 966,718.

OIL BURNER. August J. Garloff, Stony Point, Cal. 966,899.

OIL BURNER. Benet F. McMahon, Beattl, Kan. 967,037.

SMOKE CONSUMER. Charles Sorensen, Gulfport, Miss. 967,074.

OIL BURNER. Henry L. Wadley, Sentinel, Okla. 967,089.

WATER GRATE AND PROTECTOR. John G. B. Johnson, Pine Valley, Manitoba. 967,128.

POWER PLANT AUXILIARIES AND APPLIANCES

AUTOMATIC PUMP FOR INTERNAL COMBUSTION ENGINES. Robert W. J. Smith, Terrell, Tex., assignor of one-fourth to John C. Anderson and one-fourth to Robert J. Rowe, Terrell, Tex. 967,073.

APPARATUS FOR GENERATING COMBUSTIBLE VAPOR. Percy Curral Pace, Loxwood, Billinghurst, England. 967,044.

PUMP. Chester Comstock, Ridgewood, N. J. 966,428.

APPARATUS FOR PRIMING AND VENTING WATER PUMPS. Albert Blauvelt, Chicago, Ill. 966,423.

GAGE GUARD. Peter G. Olson, St. Paul, Minn. 966,468.

HEATING OF TURBINE PARTS. Charles Algernon Parsons, Newcastle-upon-Tyne, England. 966,588.

COMBINED MANUALLY AND AUTOMATICALLY ACTUATED VALVE. Frederick C. Ellison, Charleston, W. Va. 966,672.

VALVE OR GATE. Frank P. Snow, Covina, Cal., assignor to Keilar-Thomason Manufacturing Company, Covina, Cal., a Corporation of California. 966,713.

STEAM TRAP AND AUTOMATIC CHECK VALVE. Samuel V. Sharwood, Brockton, Mass., assignor of one-half to Edmund L. Reed, East Bridgewater, Mass. 966,709.

REBOUNDING IGNITION AND SPACER DEVICE FOR EXPLOSIVE ENGINES. Stephen Dudley Field, Stockbridge, Mass., assignor to Pittsfield Spark Coil Company, Dalton, Mass., a Corporation. 966,737.

SUPERHEATER. William F. Buck, Chicago, Ill. 966,792.

RECORDING PRESSURE GAGE. Edgar H. Bristol, Naugatuck, Conn., assignor to the Industrial Instrument Company, Waterbury, Conn., a Corporation of Connecticut. 966,790.

VALVE. Rudolph Conrader, Erie, Penn. 966,797.

ENGINE VALVE. John Dillander, Temple, Tex. 966,810.

FLAT ROPE OR BELT. Robert A. Hammond, Sandwich, Mass. 966,830.

CHIMNEY FLUE CLEANER. Iver I. Oium, Westby, Wis. 966,853.

METHOD OF MANUFACTURING GASKETS. Joseph M. Towne, East Orange, N. J., assignor to Safety Car Heating and Lighting Company, a Corporation of New Jersey. 966,873.

HAND OILER. William Rubly, Tuckahoe, N. Y. 966,861.

UNION. William H. Stoddard, Somerville, Mass. 966,870.

VALVE. Carl G. Sprado, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 966,867.

VALVE ARRANGEMENT FOR STEAM ENGINES. Thomas G. Aultman, Fairmont, W. Va. 966.880.

GOVERNOR MECHANISM FOR INTERNAL COMBUSTION ENGINES. Frederick T. Flinchbaugh, York, Penn. 966,896.

VALVE. William Hallowell, Manchester, England, assignor of one-half to Henry Storey and William Edward Storey, Manchester, England. 966,909.

PACKING. Leopold Katzenstein, New York, N. Y. 966,922.

BLOWER FOR BOILERS. Edward B. Barnhill, Marion, Ind., asignor to Ernest W. Micklin, Detroit, Mich. 966,987.

BOILER-TUBE CLEANER. Frank Efraim Carlson, Brooklyn, N. Y. 966,998.

RELIEF VALVE. Felix H. Carssow, San Francisco, Cal. 966,999.

PACKLESS VALVE. Charles S. Frishmuth, Philadelphia, Penn. 967,009.

MEANS FOR COOLING THE CYLINDERS OF INTERNAL COMBUSTION ENGINES. Leopold Durand, Paris, France, assignor of one-half to Rudolf August Louis Lehmann, London, England. 967,117.

WATER HEATER. Donald Barns Morison, Hartlepool, England. 967,142.

ELECTRICAL INVENTIONS AND APPLICATIONS

SYSTEM OF MOTOR CONTROL. Samuel H. Keefer, Plainfield, N. J., assignor to Niles-Bement-Pond Company, Jersey City, N. J., a Corporation of New Jersey. 967,133.

VARIABLE SELF-INDUCTANCE COIL. Joseph Ray Jesse, Birmingham, Ala. 966,555.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

BUSINESS ITEMS

The Mexican-Northwestern Railway Company, of 25 Broad street, New York City, has recently placed a large order for motors with the Westinghouse Electric and Manufacturing Company. The order includes 168 induction motors, aggregating 3738 horsepower of the types MS and HF, ranging from 3 to 200 horsepower. These motors will be shipped to the company's property at Madera, Chihuahua, Mexico, to be used in the operation of the saw and planing mills.

The Burke Furnace Company, 59 Clark street, Chicago, Ill., report recent sales of the Burke smokeless furnace as follows: Three of 75 horsepower for the Ellsworth building, Chicago; two of 125 horsepower for the Mallers building, in the same city; two of 150 horsepower for the city of Batavia, Ill.; three of 350 horsepower and two of 225 horsepower for the city of Lansing, Mich.; two of 150 horsepower for the Great Western Life Assurance building, of Winnipeg, Manitoba; one of 300 horsepower for the Weinbrenner Boot and Shoe Company, and one of 225 horsepower for the Ziegler Candy Company, of Milwaukee. It is a remarkable fact that all of the above installations are in connection with the Continental type of boiler, with corrugated furnaces. This combination is becoming popular and some excellent economies are shown with these outfits.

NEW EQUIPMENT

Point Grey, B. C., will spend $50,000 for a waterworks system.

The city of Seaforth, Ont., will install a hydroelectric power plant.

Farmington, Minn., has voted $10,000 bonds for a waterworks system.

Rush City, Minn., will vote on issuing bonds for its waterworks system.

The Salem (N. J.) Electric Company will erect a new modern power plant.

A $20,000 boiler house will be erected for the Reading (Penn.) Iron Company.

The city of Mulvane, Kan., voted to issue $30,000 bonds for waterworks system.

The American Valve Company, Coxsackie, N. Y., is building a new power house.

The New Richmond (Wis.) Roller Mills will install an auxiliary steam power plant.

H. J. Heinz Company, Salem, N. J., is increasing the capacity of its power plant.

J. S. Collins & Son, Mooreston, N. J., will increase the capacity of its power plant.

The Frank C. Schilling Company, Green Bay, Wis., will erect a cold-storage plant.

The city council, Rushford, Minn., will make extensions to its electric-light system.

The city of Timmonsville, Okla., will vote on issuance of $30,000 bonds for waterworks.

The Birmingham (Ala.) Horseshoe and Rolling Mills Company will erect a power house.

F. E. Perry, Mill street, Gardner, Mass., is receiving propositions for a water-wheel governor.

The New York State Training School for Girls, Hudson, N. Y., is erecting a new power plant.

The Standard Sanitary Manufacturing Company, Toronto, Ont., is building a new power house.

The refrigerating plant of Swift & Co., Troy, N. Y., was destroyed by fire. Loss, $40,000.

Outlook, Sask., will equip a large new power house in connection with its water-supply system.

Calgary, Alberta, will spend $125,000 for an addition to its power house and new equipment.

The citizens of Morriston, S. D., will vote on the issuance of $10,000 bonds for a waterworks system.

The Central Iowa Produce Company, Atlantic, Iowa, will install an ice and cold-storage plant.

Contracts will be awarded soon for new boilers for a municipal electric-light plant at Redfield, S. D.

The village trustees, New Athens, Ill., are reported to be considering the construction of waterworks.

The Consumers Power Company will spend $30,000 on improvements at its gas plant in Faribault, Minn.

The Laclede Power Company, St. Louis, Mo., will erect a $25,000 power house at the foot of Mound street.

The Quinebaug Company, Danielson, Conn., will install two new steam engines of 300 and 500 horsepower.

The Young Men's Christian Association, Lawrence, Mass., will install a horizontal-tubular steam boiler.

The city of Fairmount, Minn., will issue $30,000 bonds for improving its electric-light plant and waterworks.

R. C. Hart, city engineer, Clinton, Iowa has been instructed to prepare plans for waterworks to cost $600,000.

Sloan & Robson, San Francisco, Cal., will prepare plans for waterworks and a sewer system for Alturas, Cal.

The Stanley Works, New Britain, Conn., will soon install a new 500-horsepower steam turbine and generator set.

The Cleveland (Ohio) Electric Illuminating Company has secured a permit for the erection of a new power house.

The city of Floresville, Tex., will install an engine and air compressor in its waterworks plant. Address the mayor.

The Hammonton & Egg Harbor City Gas Company, Hammonton, N. J., will increase the capacity of its power plant.

The Salem (N. J.) Glass Company will increase the capacity of its power plant. New boiler will be installed.

The city trustees, Porterville, Cal., are said to be considering an expenditure of $25,000 for an auxiliary water plant.

An election will be held at Olathe, Kan., to vote on issuing $100,000 bonds for waterworks. Mattie A. Burns, town clerk.

Plans have been completed for a new power house for the Mt. Vernon Railway and Light Company, at Columbus, Ohio.

The Crystal Ice Manufacturing Company, Argenta, Ark., will double the capacity of its plant, expanding about $50,000.

The Ohio Valley Scenic Route, of Toronto, Ont., has purchased a building, which will be converted into a power house.

The Citizens Light, Heat and Power Company, Montgomery, Ala., will spend $50,000 on new equipment for its plant.

The Cumberland Glass Company, Bridgeton, N. J., is to increase the capacity of its power plant. Will install new boiler, etc.

An addition is being erected to the power house of the Toledo (Ohio) Railway and Light Company, at the foot of Madison street.

The San Jacinto Rice Company, Elena, Tex., is in the market for a 30-inch centrifugal pump for irrigation purposes. J. F. Jones, engineer.

The city of Warrenton, Mo., voted $25,000 bond issue for waterworks and sewer system. Fuller-Coult Company, Chemical building, St. Louis, engineers.

Bids will be received until September 20, by the board of water commissioners, Detroit, Mich., for a pumping engine of 25,000,000 gallons capacity.

The city council, Clinton, Iowa, has passed a resolution instructing the city engineer to prepare an estimate of the cost of installing a modern municipal water plant.

The Navy Department, Bureau of Supplies and Accounts, Washington, D. C., will open bids September 8, for 36 steam gages, 1500 water-gage glasses as per Schedule No. 2830.

The Sterling-Moline Traction Company has awarded contract for the construction of a road. Power plant will be erected at Lyndon, Ill. The offices of the company are at Sterling, Ill.

The Navy Department, Bureau of Supplies and Accounts, Washington, D. C., will open bids September 6, for two turbo-generating sets for the Brooklyn navy yard as per Schedule No. 2826.

J. G. White & Co., Inc., of New York, have been awarded contract for the construction of a hydroelectric power plant for the Eastern Tennessee Power Company, at Parksville, Tenn. This is to cost about $2,000,000.

Bids will be received by the board of water commissioners, Evansville, Ind., until September 29, for two steam-driven pumps or pumping engines for supplying water to its filtration plant, each to have a daily capacity of 12,000,000 gallons.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

SALESMEN—We have an A-1 proposition for high-grade salesmen. Address Box 247, POWER.

WANTED—Thoroughly competent steam specialty salesman: one that can sell high-grade goods. Address "M. M. Co.," POWER.

MASTER MECHANIC for blast furnace plant in Virginia; state qualifications and salary expected. Address "W. H.," POWER.

AN ENGINEER in each town to sell the best rocking grate for steam boilers. Write Martin Grate Co., 281 Dearborn street, Chicago.

AGENTS WANTED on high-class steam specialties; give full particulars as to territory, etc. Address Box 246, Care POWER, 150 Michigan Ave., Chicago, Ill.

WANTED—Engineer to take charge of gas engines, refrigerating machinery and producer plant; young man who is ambitious, familiar with vertical type of producer-gas engine. Box 248, POWER.

STEAM SPECIALTY SALESMEN—Some open territory for the best steam specialty ever offered; excellent side line; pocket sample; virgin field. Weiss Brothers Mfg. Company, Detroit, Mich.

WANTED—Waterworks engineer to take charge of machinery and distributing system of about three million gallons capacity daily; no one but man of experience wanted; references required. Address James J. Sweeney, President, Owensboro Water Works Co., Owensboro, Ky.

WANTED—Agents to sell on commission highest grade steam specialties on market; energetic, capable specialty salesmen can make big profits; none others need apply: a gilt-edged proposition for high-grade salesmen from a very reliable and established manufacturer. Address W. T. Jameson, 100 Surrey Road, Mansfield, Ohio.

Power and The Engineer

Vol. 32 NEW YORK, SEPTEMBER 6, 1910 No. 36

TIME has proven that the pioneers in steam-engine design, Sickels and Corliss, had the right idea when they designed the releasing valve gear. Although nearly seventy years have elapsed since the first of these gears was put into use, the germ of the invention, more or less disguised to meet the requirements of modern practice, is to be found in practical operation in thousands of engine rooms at the present time.

Because of the fact that Corliss did more to perfect this type of gear than any other man, his name is usually associated with it, and engines so equipped are commonly known as Corliss engines.

All Corliss gears have two essential features: First, four separate ports to which steam is distributed; namely, two admission and two exhaust ports. Second, an external mechanism by which the rotary valves are operated quickly and positively at the various points of the steam cycle, the governor controlling the point of cutoff at which the valve is released by a spring or dashpot.

As there are separate ports for admission and exhaust, the former can be made narrow, which, with the quick opening and closing of the admission valve, insures full steam pressure right up to the point of cutoff and prevents wire-drawing the steam. Also the rapid opening and closing of the exhaust ports produces an almost constant back pressure. These features make it possible to obtain an indicator diagram resembling very closely the theoretical diagram.

The valves are independent of one another and are easily adjusted for any desired distribution of steam. This, with a variation in cutoff by the governor to suit the load, insures an economical consumption of steam. For these and many other reasons the Corliss engine is unequaled for certain kinds of service.

The distinguishing feature between the valve gears of the various makes of Corliss engine lies in the external mechanism between the governor and the valve stem. Some of these are examples of very ingenious kinematic systems, while others are of very simple design.

Every engineer should be familiar with the numerous makes of Corliss valve gear; not because of the fact that he may be called upon at any time to operate any one of them, for the engineer who knows his business could easily solve their method of working after a few minutes' study, but a familiarity with the various types enables him to make a more intelligent selection when choosing an engine to meet certain requirements.

This issue contains the first instalment of an article upon the "Development of Releasing Valve Gears," which is unusually complete in its treatment of the subject. The article covers the development of the releasing gear from the rather crude forms of its earliest application to the highly-perfected gears on the market at the present time. Each gear is illustrated, its distinctive features are pointed out, and its operation explained; special care has been taken to make the illustrations uniform so that the sketches may be readily compared.

Aside from its historical value the subject of releasing valve gears is a live topic which no engineer can afford to pass by.

Training Engineers at Franklin Union

By F. L. Johnson

> *This institution, which owes its establishment to Benjamin Franklin, is an industrial school for men who desire an advanced knowledge and a wider experience in their vocations. Though covering a wide field in industrial pursuits it furnishes exceptional opportunities to the engineer.*

In 1791, Benjamin Franklin bequeathed to his native town of Boston the sum of $5000, to be held in trust for one hundred years, during which time the principal and its accruing interest were to be lent at 5 per cent. per year for the purpose of aiding "married artificers starting in business."

At the end of the first century the fund was to be divided: A portion to be spent in such "public works which may be judged of most general utility to the inhabitants, such as fortifications, bridges, aqueducts, public buildings, pavements or whatever may make life in the town more convenient to its people and render it more agreeable to strangers resorting thither for health or a temporary residence."

In 1891 the first portion of the fund became available for the purpose intended. But owing to several reasons, among which were litigation and an attempt of the city government to spend a large portion of it in a junketing expedition to find out how to invest the rest of it, it was not used at that time.

It was finally decided that the Supreme Court of Massachusetts should be asked to construe the meaning of the terms of the bequest; this resulted in the appointment by the court of a board of twelve managers of whom but one, the mayor, could possibly be a public official or politician.

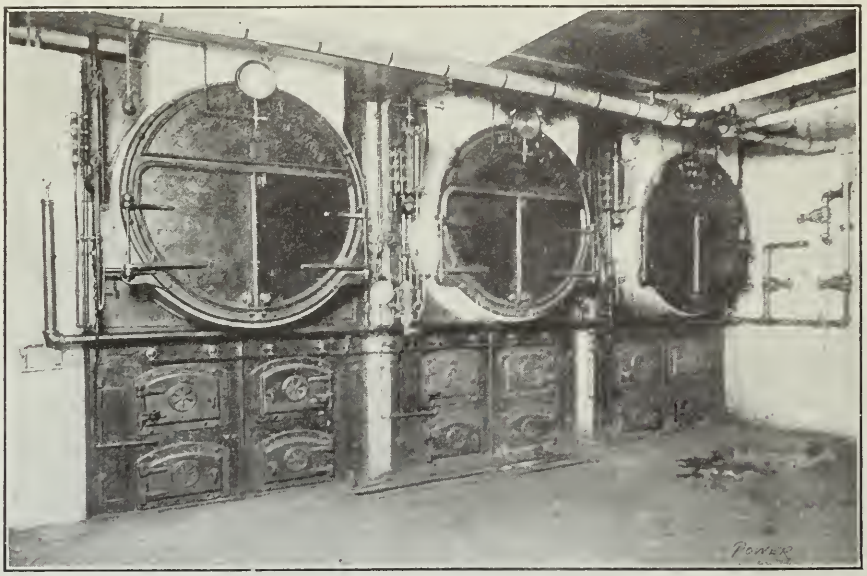


Fig. 1. Front View of Boilers

The men selected for the board of managers were Richard Olney, president; Nathan Matthews, vice-president; James J. Storrow, secretary; William Endicott, treasurer; Rev. Charles W. Duane, Frank K. Foster, Charles T. Gallagher, Mayor George A. Hibbard, Rev. Alexander K. MacLennan, Rev. Charles E. Park, John A. Sullivan, Prof. George F. Swain. In the judgment of this board an industrial school was deemed the one public work of the "most general utility to the inhabitants of Boston," and some $400,000 of the fund was expended in building and equipping the Franklin Union.

As practically all of the available fund was spent in buying the land, erecting the building and supplying its equipment, nothing remained for the support of the institution. Andrew Carnegie, learning of the situation, gave a fund which provides for an annual income of $20,000 which, with the nominal sums received for tuition, furnishes the entire revenue of the institution. This, though inadequate, permits about one-half of the building to be utilized for the purposes intended.

In its equipment for practical teaching the institution is radically different from the ordinary manual-training school. Instead of competing with existing schools of a kindred nature, it supplements their work.

The foremost idea from the beginning has been the teaching of men. In the selection of the entire equipment this object has been kept in view. It aims to reach men having more or less limited experience and knowledge of their vocations with a view to increasing their experience and perfecting their knowledge.

The instruction is less of a technical nature than that given at some older and larger institutions, and is more practical than can be obtained at the evening classes of the public schools, the Young Men's Christian Association classes or the regular trade schools. As the best methods of teaching trade practice and theory are to be found in the commercial shops and enterprises, the methods of the Franklin Union have been shaped to closly follow present-day shop practice.

Instructors, so far as is possible, are selected from the shop, the engine room, the commercial laboratory and from the building trades and professions. Shop language and shop problems afford the student, with his practice instruction in shop work, the most practical way for advancement.

Though the course of instruction covers a broad field embracing several trades, there is but little that is not either directly or indirectly related to the engineer and his work; and there are probably few who would not be benefited by a course at the institution.

The character of the instruction being practical a student who is qualified may have additional work as an equivalent for that in which he is already proficient. The arrangement of the courses is the result of the most careful planning on the part of the instructors, who have not hesitated to take advantage of many valuable suggestions made by the students, making the studies of the highest value to the man from the shop or power plant.

Evening classes are held on Monday, Wednesday and Thursday evenings, from the last of September to the last of March. Two-year courses are given in the following subjects: Machine construction, industrial electricity, steam engines and boilers, structures, architectural working drawing, industrial chemistry.

Classes, without certificates, continuing for one season: Sheet-metal drafting, mechanical drawing, industrial arithmetic and shop formulas, estimating for architects and builders, firemen's class, heating and ventilating, practical mechanics, heat and hydraulics, gas and gasolene engines, ship and yacht drafting.

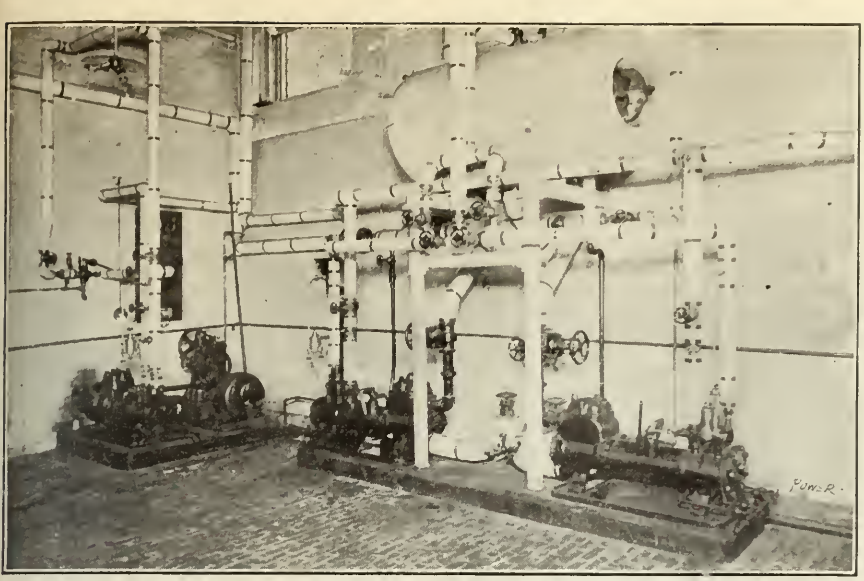


FIG. 2. FEED AND CIRCULATING PUMPS IN BOILER ROOM

Saturday afternoon classes are held in industrial chemistry, steam engines and boilers, industrial electricity, gas and gasolene engines and industrial arithmetic. The instruction covers the same ground as the evening courses.

In the steam laboratory, of which the boiler-room equipment is a part, there are three 6x18-foot horizontal return-tubular boilers with pumps, damper regulator, pressure-reducing valves and all the necessary boiler-room appliances. The equipment of this laboratory comprises also, one 8x24-inch Cooper Corliss engine, used with one or two eccentrics, one 12x12-inch Ames engine coupled to a 75-kilowatt General Electric direct-current generator, one 8x10-inch Armington & Sims engine belted to a 17-kilowatt Westinghouse generator, one 4x4-inch Sturtevant engine coupled to a generator, one 3 and 6 by 5-inch compound piston-valve marine engine, one 8x10-inch Payne engine, used for plain slide- and riding-cutoff valve setting; one Miller engine, one small upright slide-valve engine, one small horizontal slide-valve engine, eight steam pumps of various makes, one Wheeler surface condenser, one air pump, one jet condenser, one small horizontal-tubular boiler, traps, damper regulators, pump governors, reducing valves and safety valves of standard makes.

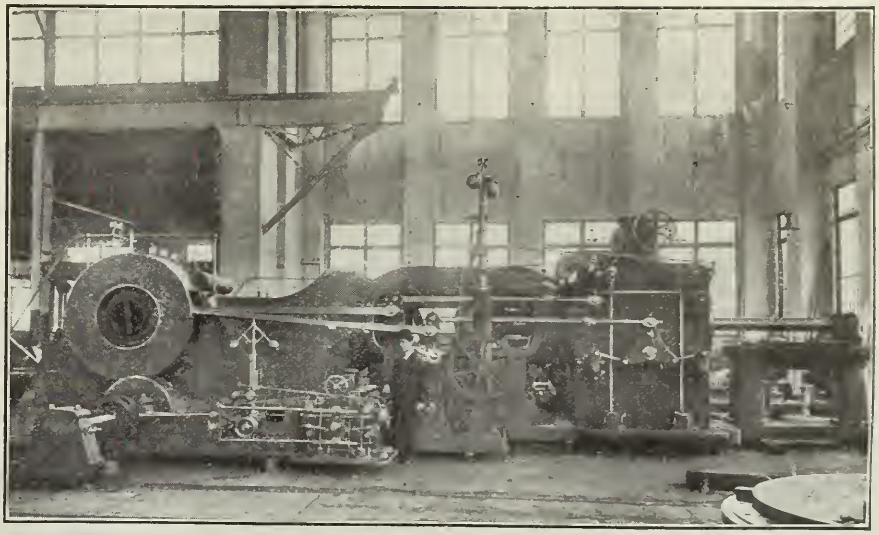


FIG. 3. ENGINE FOR FRANKLIN UNION ON THE ERECTION FLOOR AT THE SHOP

These engines and pumps are so installed that they may be worked condensing or noncondensing, using either the surface or jet condenser.

In the gas-engine laboratory there are: One 9-horsepower Foos gas engine, one 6-horsepower Mietz & Weiss oil engine, one 10-horsepower Atlantic special two-cylinder, two-stroke gasolene engine, one 25-horsepower Johnson four-cylinder, four-stroke gasolene engine, one 5-horsepower Lathrop one-cylinder, two-stroke gasolene engine, one 3-horsepower Ferro one-cylinder, two-stroke gasolene engine, one 8-horsepower Atlantic three-cylinder, two-stroke gasolene engine, one 5-horsepower Wolverine one-cylinder, four-stroke gasolene engine, one 8-horsepower Clifton two-cylinder, four-stroke gasolene engine, one 2-horsepower Tufts one-cylinder, two-stroke gasolene engine.

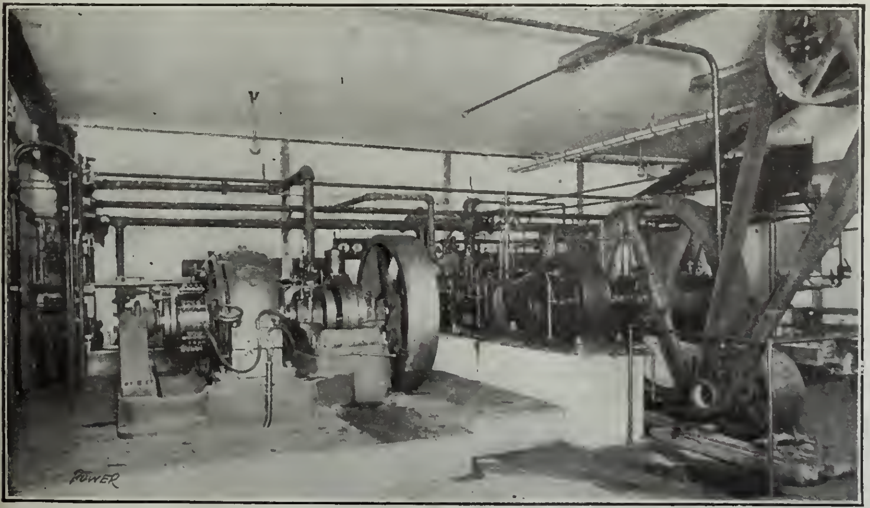


FIG. 4. ONE END OF STEAM-ENGINE LABORATORY

All these engines are equipped with prony brakes, and with scales and tanks for weighing fuel. In addition there is an assortment of standard carbureters, timing devices and magnetos.

The electrical-laboratory equipment consists of one 45-horsepower direct-current Westinghouse motor coupled to a 30-kilowatt alternating-current Westinghouse generator, one 7½-kilowatt Westinghouse rotary converter, three 6-horsepower direct-current Westinghouse

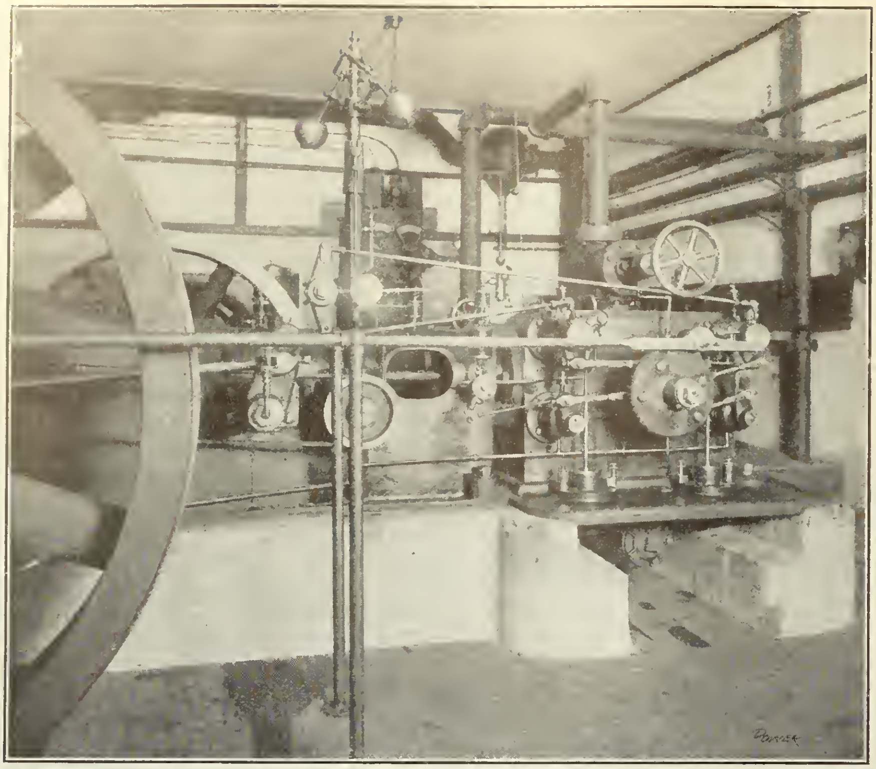


Fig. 5. Cooper Corliss Engine, 8x24 Inches

motors coupled to a 3½-kilowatt generator, one 5-horsepower Westinghouse variable-speed shunt motor with prony brake, one 5-horsepower direct-current Westinghouse motor with prony brake, one 5-horsepower Westinghouse variable-speed motor coupled to a Holtzer-Cabot 3-kilowatt generator, one 5-horsepower Westinghouse induction motor coupled to a 3-kilowatt Imperial electric generator, one 5-horsepower General Electric shunt-wound motor coupled to a 3¼-kilowatt Sprague generator, one 6-horsepower Edison bipolar motor, one 4½-horsepower 230-volt Crocker-Wheeler motor, one 5-horsepower General Electric motor coupled to a 3-kilowatt Holtzer-Cabot generator.

There are also banks of portable lamps, water rheostats, transformers, voltmeters and ammeters.

A glance at the course of instruction will show that the steam, mechanical and electrical courses constitute only a small part of the field covered by the curriculum; these, however, form the portion of greatest interest to engineers.

All of the apparatus in the steam- and gas-engine laboratories is directly under the supervision of George H. Bird, superintendent of the building, who is an indefatigable worker, spending the greater part of his waking hours in original investigation and experiment in the steam-engine laboratory.

One line of his investigation has been in the direction of finding the cost of leaks in the valves and pistons of steam engines. After ascertaining that the piston and valves of the 8x24-inch Cooper engine were tight, he drilled holes through them into which he fitted plugs, which could be removed to introduce any desired kind of a leak in the piston, steam valve or exhaust valve, individually or in concert. Loss from one or a combination of leaks was measured by weighing the water drawn from the Wheeler surface condenser.

Any problem in steam-engine construction or repairs is "nuts and raisins" to Mr. Bird, who never seems more interested than when he has possession of an old slide-valve engine and, surrounded by the class in steam engines and boilers, directs its reconstruction and repair, putting in new pins and brasses, adding a shifting eccentric or a riding cutoff, and so on.

Junk shops and second-hand machinery depots are ransacked for anything that will furnish work or instruction for the students who by actual hand work get, in one short term, an experience under able supervision that probably would not come to them in years of service in an ordinary power plant.

In an almost endless variety, problems in steam-engine installation, adjustment, operation and repairs are furnished. One group may use the Riehle testing machines on test pieces of boiler plate or concrete prepared by another group. A small class may today assemble an engine or a pump which was taken apart yesterday. Here a few are adjusting a riding cutoff designed by others while their own design will be taken in hand next week by others, and so on through the entire course, including boiler tests,

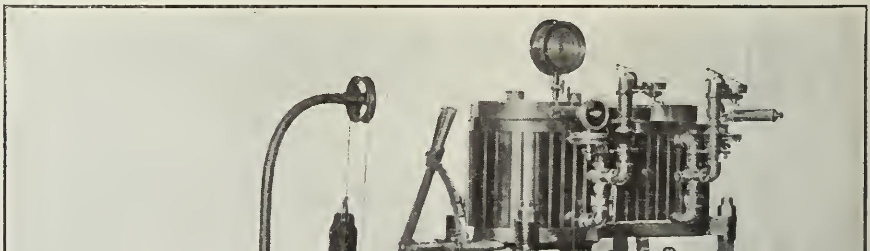


Fig. 6. Marine Compound Engine, 3 and 6 by 6 Inches

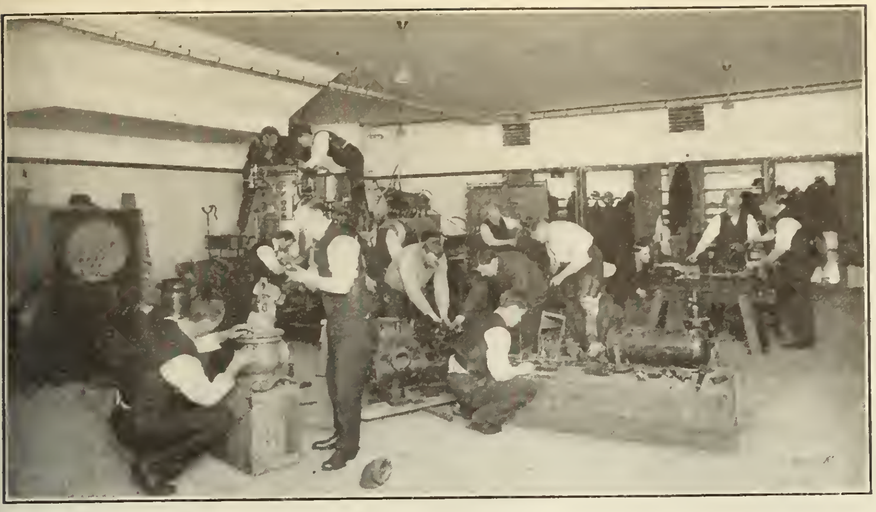


Fig. 7. Steam-engine Class at Work

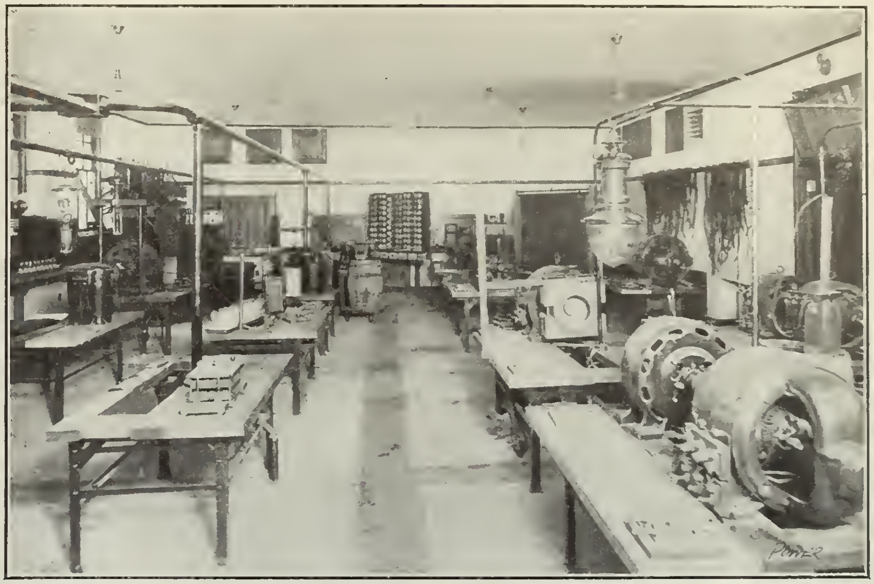


Fig. 8. Electrical Laboratory

flue-gas analysis, design, operation and repairs of feed pumps, condensers, circulating and air pumps, lubricators, prony brakes, safety valves and every appliance connected with steam engineering. In the electric, gas and gasolene engine and the chemical laboratories the methods of teaching follow the same lines as closely as possible, making the instruction practical in every feature.

Examination question No. 19, selected at random from the list, is a fair example of the methods by which every student is required to demonstrate that he has an intelligent grasp of the subject which he has been studying.

It is interesting to watch the classes at their work; so intent are all in making every minute count for something gained. One of the illustrations shows a group at work in a corner of the engine room; some are at one thing and some at another, while at the left is seen a relic of other days—an old horizontal-tubular boiler built in 1862.

That part of the curriculum of greatest interest to engineers is embraced in steam engines and boilers, which covers two years and comprises lectures and laboratory practice for engineers, oilers and steamfitters. In the first year the subjects are boiler details; fittings, setting, installation, fuel and combustion; construction and operation of pumps and simple engines; setting plain slide valves. Those for the second year are Corliss and other automatic cutoff engines; heat and the generation and properties of steam; compound engines, condensers, indicators, elevators and refrigerating machinery.

This may and should be, if the student honestly desires the utmost advancement in knowledge and experience, supplemented by studies in machine construction which embrace machine-shop arithmetic, elementary mechanics, mechanical drawing, logarithms and the use of the slide rule, electricity in the shop, machine drawing, mechanics, mechanism, testing materials and machine design. He should also make selections from the course in industrial electricity which is intended to meet the needs of men working in the various electrical trades. The first year consists of lectures, recitations and laboratory work covering static and dynamic electricity, and magnetism, which every student must understand before he can handle intelligently any type of modern electrical machinery. The course includes instruction in figuring electrical direct-current circuits, in drawing wiring diagrams, in the theory and wiring of commercial dynamos and motors, and in solving practical problems. The laboratory work involves experiments illustrating the principles brought out in the

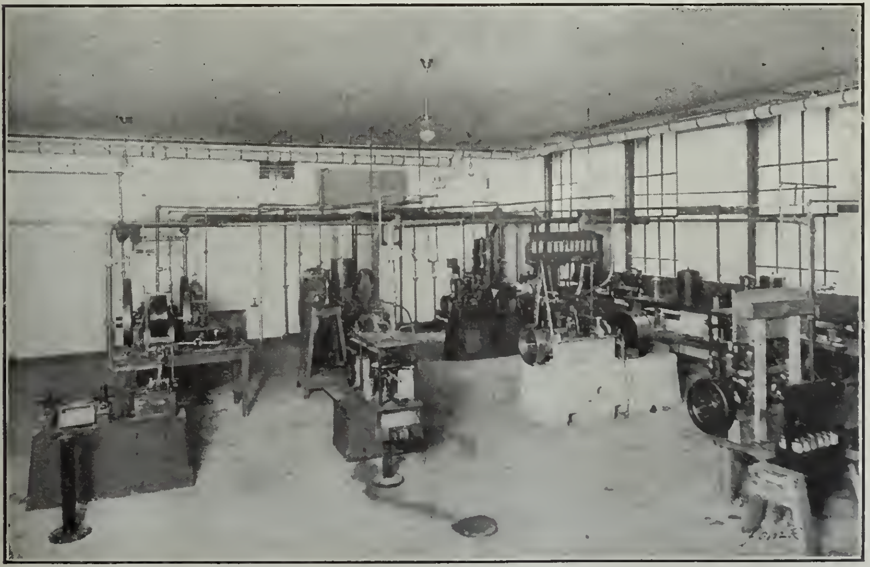


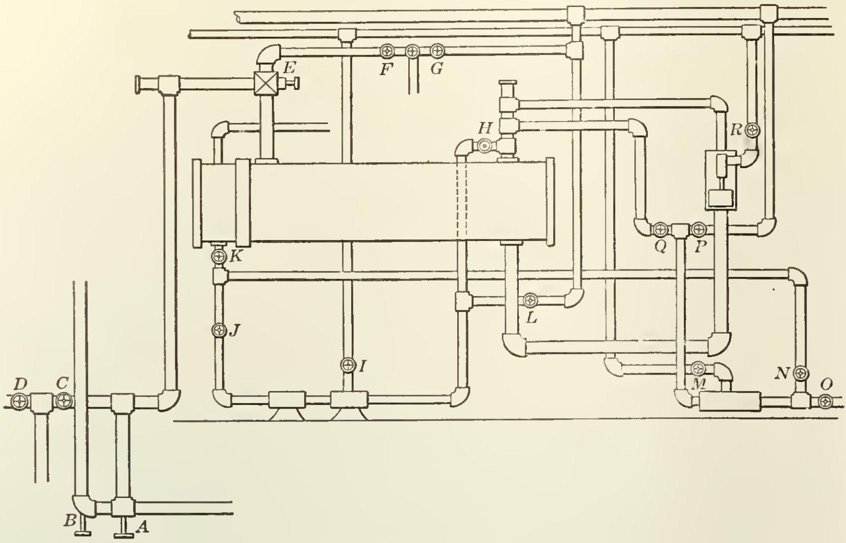
Fig. 9. Gas- and Gasolene-engine Laboratory

FRANKLIN UNION, Boston.

Steam Laboratory Practice.

Second Year

19. Condenser and steam piping in steam laboratory.



Make a list of the valves, showing which should be opened and which should be closed, in order to run the condenser with the Ames Engine, using the Davidson pump for the circulating water. Include the valves for both the air and circulating pumps, indicating the valves by the letters in the sketch.

Power

FIG. 10. SAMPLE EXAMINATION QUESTION

class room and the use of electrical measing instruments.

During the second year the laboratory work embraces the actual testing of direct-current and alternating-current motors, generators, arc lamps and transformers, and the use of measuring instruments. All the laboratory apparatus is of commercial size. The motors range from 5 to 50 horsepower and the generators from 3 to 30 kilowatts, and are typical of those produced by five different manufacturers. The laboratory work is supplemented by lectures, recitations and the solution of practical problems.

The course in industrial chemistry, though intended for young men employed in chemical-manufacturing plants, mills and drug houses, and for men who wish to acquire a practical knowledge of the principles and applications of chemistry, should not be passed by the engineering student if time can be afforded for even only a part of the course. The first year includes lectures and laboratory practice in elementary general chemistry. The second year includes qualitative and quantitative analysis. The third year offers an opportunity for instruction in commercial analysis and for specialization on industrial problems.

The entire course covers lectures, recitations and experiments in elementary chemistry, general chemistry, qualitative analysis, quantitative analysis, organic chemistry and commercial analysis.

Besides these there are complete and partial courses in sheet-metal drawing and mechanical drawing. Study in heating and ventilating can be followed to advantage by the practical steam and gasolene engineer as the power plant in the building is available for this work.

The "steam engines, boilers and firemen's class" is fortunate in being directly under the instruction of Cerlton A. Read, professor of steam engineering at the Worcester Polytechnic Institute, who is assisted in his work by George H. Bind, superintendent and chief engineer of the Franklin Union building; Edward H. Kearney, chief engineer of the John Hancock building, Devonshire street, Boston, George W. Hayes, chief engineer of the Puffer Manufacturing Company, Winchester, Mass., Frank G. Wright Factory building Trust Company, South Boston, and Robert E. Thayer, Massachusetts Institute of Technology, who are all able and eminent engineers in direct charge of important steam- and gas-engine plants in Boston and the immediate vicinity.

Walter B. Russell, a graduate of Massachusetts Institute of Technology, also graduate of and later instructor at the Pratt Institute, Brooklyn, N. Y., who was associated with C. W. Cross, superintendent of apprentices for the New York Central Railway lines and was in no small measure responsible for the efficient apprenticeship system of these roads, was selected by the board of managers for director of the Franklin Union. The steady improvement in the effectiveness of the instruction, the positions in industry secured by graduates and the fact that the applications for admission far exceed the capacity of the institution furnish ample evidence that no mistake was made in this selection.

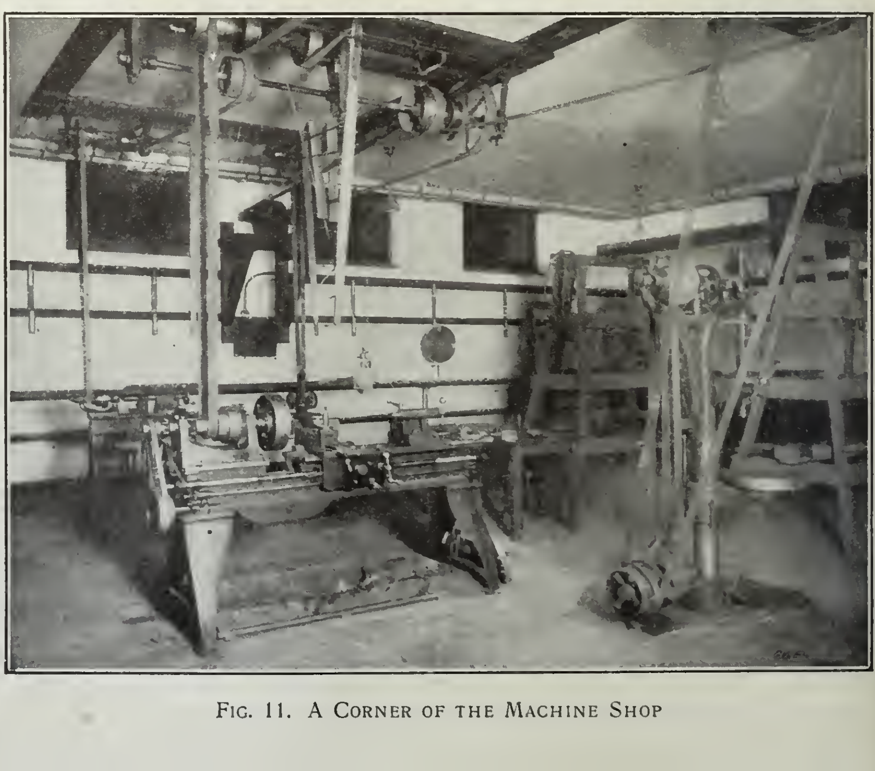


FIG. 11. A CORNER OF THE MACHINE SHOP

Development Releasing Valve Gear

By C. A. Tupper*

A review of the development of this type of valve gear from the days of Watt to the present time. Each distinctive gear is illustrated, its operation explained and its salient features pointed out. A second instalment will conclude the discussion of the releasing valve gears used in current practice.

D. K. Clark, in his work on "Steam Engines," is authority for the statement that the disconnecting valve gear invented in England by James Watt, for opening and closing poppet valves, as shown in Fig. 1, discloses the germ of the modern gear; and that Watt advocated, if he did not practise, the employment of the governor to vary the cutoff instead of wire-drawing the steam. From the explanation accompanying Fig. 1, however, it is evident that this gear was operated without either governor or wristplate.

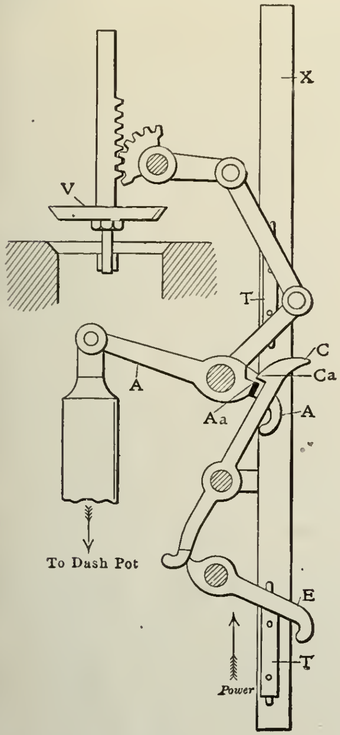


FIG. 1. ORIGINAL WATT GEAR

$A\ A'$ = Drop lever which actuates the valve by connecting link, arm, rack and pinion, and is also connected to the dashpot rod.
Aa = Catch on A.
C = Hook.
Ca = Catch on C.
E = Trip lever.
T = Tappets.
V = Valve.
X = Rod connected to engine beam.

OPERATION

When X is moved in one direction by the engine beam, then by means of tappets T the opening and closing of the valve is caused, and in the opposite direction tappets cause the curved lever E to trip or liberate Ca on C from Aa on A'. The dashpot immediately draws A downward, to act either for opening or closing the valve.

Releasing or tripping mechanisms for valve gears were first brought out in 1841, by an American, Frederick E. Sickles, who in May, 1842, and September, 1845, obtained U. S. letters patent on "the combination of a cam, arm or wiper moved by an independent motion with a reciprocating trip and catch to liberate a weight, spring or other force that acts to close a cutoff valve." From this claim it would be inferred that the invention of Sickles covered the essential features of the liberating valve gears in use at the present time, and considerable confusion exists in the public mind as to the exact degree of credit due him.

The Sickles mechanism, as practically developed, appears to have been first utilized by an engineering firm in New York City in the motive power of a steamer named the "South America." It was applied to poppet valves lifted by cams. When the cam had lifted the valve stem to the desired point, a latch connection between the stem and the lifting mechanism was released and the valve closed independently by dropping. As the lifting mechanism descended, it

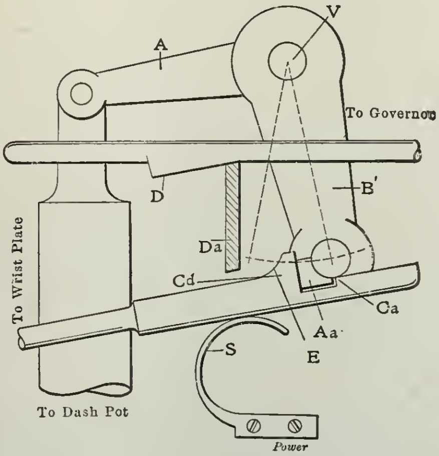


FIG. 2. FIRST GEORGE CORLISS GEAR

A = Drop lever, mounted on the valve stem which it actuates, and connected to the dashpot rod.
B' = Lever carrying catchblock and receiving its motion from the wristplate.
Aa = Catchblock on B'.
Ca = Catch on wristplate rod.
Cd = Stop on wristplate rod.
D = Incline plane or knock-off cam.
Da = Trip bolt.
E = Projection for tripping.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da forcing E away, thus causing Ca to release B', A is then immediately drawn downward by the dashpot, closing the valve.

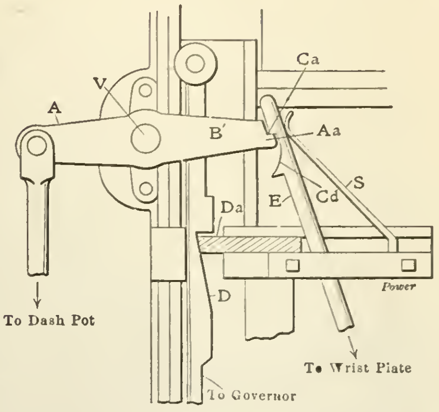


FIG. 3. SECOND GEORGE CORLISS GEAR

A = Drop lever mounted on the valve stem and connected to the dashpot rod The valves are actuated by small rollers and toothed segments.
B' = Lever carrying catchblock and receiving its motion from the wristplate.
Aa = Catch on B'.
Ca = Catch on wristplate rod.
Cd = Stop on wristplate rod.
D = Incline plane or knock-off cam.
Da = Trip bolt.
E = Tripping surface on wristplate rod.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate Ca engages Aa, thereby raising A until E comes in contact with Da and forces E away; this causes Ca to release B'. A is then immediately drawn downward by the dashpot and closes the valve.

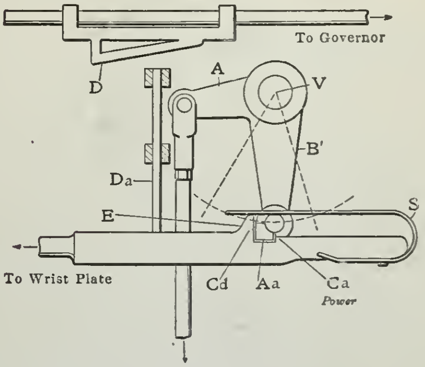


FIG. 4. THIRD GEORGE CORLISS GEAR

A = Drop lever or bell crank, mounted on the valve stem which it actuates, and connected to the dashpot rod.
B' = Lever carrying catchblock and receiving its motion from the wristplate.
Aa = Catchblock on B'.
Ca = Catch on wristplate rod.
Cd = Stop on wristplate rod.
D = Incline plane or knock-off cam.
Da = Trip bolt.
E = Projection for tripping.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da and forces E away, causing Ca to release B', A is then immediately drawn downward by the dashpot and closes the valve.

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displaced the latch; but after it had passed, the latch was forced forward into position to be caught again by the lifting mechanism on the next stroke. This latch release, however, could be adjusted only by hand, and the next step in the development of such mechanisms was to make them subject to the automatic control of a governor.

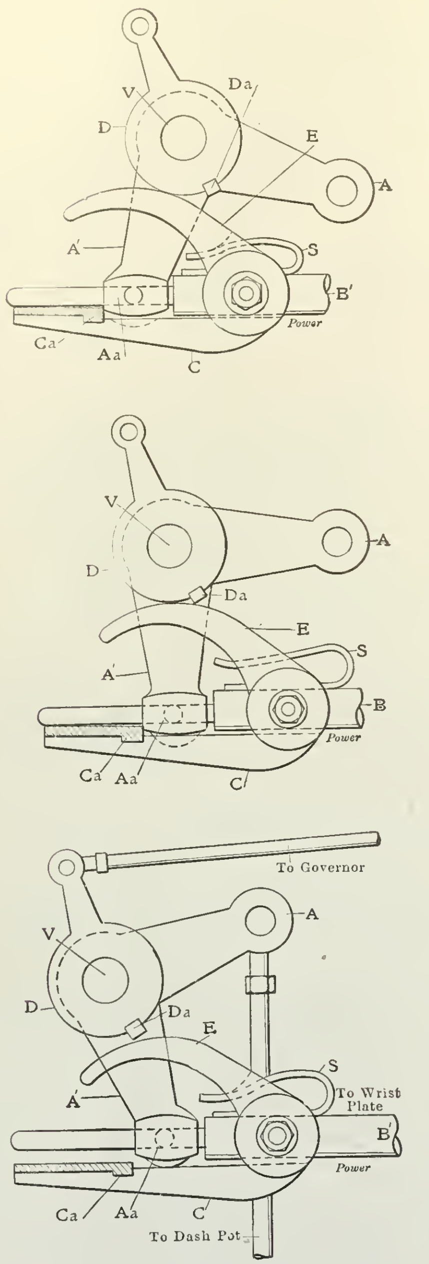


FIG. 5. CRAB-CLAW GEAR OF GEORGE CORLISS IN OPERATING POSITIONS

A A' = Drop lever or bell crank, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Block carried by A'.
B' C E = Crab-claw from which this type of gear takes its name. B is a rod, deriving its motion from the wristplate, which carries the hook C and tripping lever E, the two last named being in one piece.
Ca = Catchblock on C.
D = Knock-off cam, loosely mounted on on the outer end of the valve stem. The position of this cam is determined by the governor.
Da = Projection for tripping.
E = Trip lever, as explained above.
V = Valve stem.
S = Spring.

OPERATION

When B' is pulled toward the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da and is forced away. This depresses C sufficiently to release Aa, and A is immediately drawn downward by the dashpot, closing the valve.

To Zachariah Allen is accredited the invention, in 1834, of an expansion gear depending upon the governor for its regulation; but the first engineer who caused the governor to act on a tripping gear was George H. Corliss, of Providence, R. I. His invention is usually dated from March 10, 1849, at which time he was granted a patent, which expired in 1870.

The characteristics of all Corliss valve gears may be said to consist of: First, the four separate steam-distributing parts, two for admission and two for exhaust, so arranged as to reduce clearance to a minimum; second, the disengagement of the external mechanism through the governor at the end of the steam admission periods, whereby the work of the engine is regulated and satisfactorily performed under relatively constant boiler pressure. To accomplish the required results it is necessary to use rotating valves which can be easily moved and reliably operated by a mechanism entirely exterior to the steam cylinder.

In this connection the first feature is the operation of the four valves from points on a wristplate, which is made to oscillate back and forth through a considerable angle by its connection with the eccentric, the arrangement being such that the steam valve will open rapidly as the engine passes its dead center and the exhaust valve on the opposite end of the cylinder will be opened wide before that point is reached. The valves usually work in diagonal pairs, the action of each pair being independent of the other.

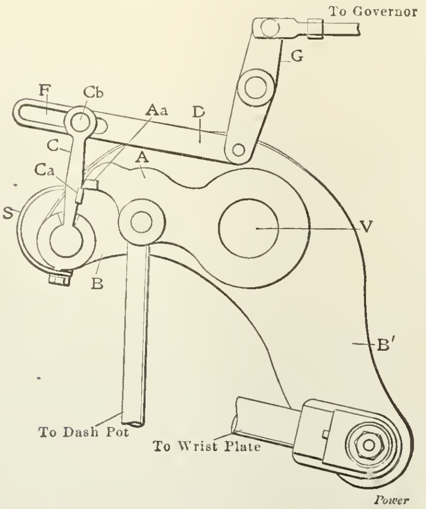


FIG. 6. LATER GEORGE CORLISS GEAR

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Block carried by A.
B B' = Curved lever carrying, at the end of the shorter arm B, a latch playing on a cylindrical socket joint. This lever receives its motion from the wristplate.
C = Latch.
Ca = Block carried by C.
Cb = Pin sliding in D (trip roller).
D = Slotted bar, performing a function similar to that of the knock-off cam, now generally used. This is supported at one end by the pin Cb, and at the other end by the arm G, the position of which, and consequently of F, is determined by the governor.
F = Slot in D, the position of which controls the points of release and cutoff.
G = Connection to governor.
S = Spring holding hook in its normal position.
V = Valve stem.

OPERATION

When B' is pulled toward the wristplate Ca engages Aa, and raises A until Cb slides to the end of the slot in D, moving C far enough outward to release A, which is immediately drawn downward by the dashpot and closes the valve.

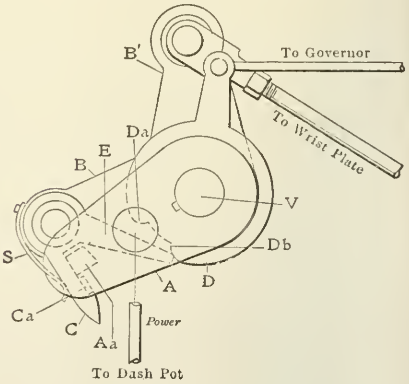


FIG. 7. REYNOLDS-CORLISS GEAR

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on A.
B B' = Bell crank, which carries hook C and tripping lever E. This bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by hook.
D = Knock-off cam mounted on the valve stem between A and B. The position of this cam is determined by the governor.
Da = Projection for tripping.
Db = Projection for safety stop.
E = Trip lever.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da. This forces E away and causes C to release A, which is immediately drawn downward by the dashpot and closes the valve.

The second feature is the release or trip of the steam-valve rods, by which the hold of the wristplate upon the valve is released and the valve is closed suddenly by a weight or spring. The differences in the design of this releasing mechanism distinguish the various Corliss engines from one another; but it is an essential characteristic of all forms that the adjustment of the gear be made by the governor, so that the speed of the engine will automatically vary the length of admission by causing cutoff when the trip occurs. The exhaust valves are positively connected to the wristplate, so that release and compression will be constant for any given valve setting, while cutoff and expansion vary according to the work of the engine.

A third feature common to Corliss gears is the closing of the valve, when disconnected from the wristplate, by means of a weight or spring attached to a dashpot; although other methods are also used.

The development of these features from the earliest designs, and the difference in detail at the present time, is briefly discussed and illustrated in the following: It will also be made manifest, as far as practicable, to what individual minds the important variations in type are principally due.

Fig. 2 represents the first Corliss gear, while Figs. 3 and 4 show the salient points of the later forms that led to the introduction by Corliss of the celebrated "crab-claw gear," illustrated in Fig. 5.

From this point the development of releasing gears proceeded independently along quite a number of parallel lines, which may be treated in two distinct divisions, namely, trip gears and cam and roller gears.

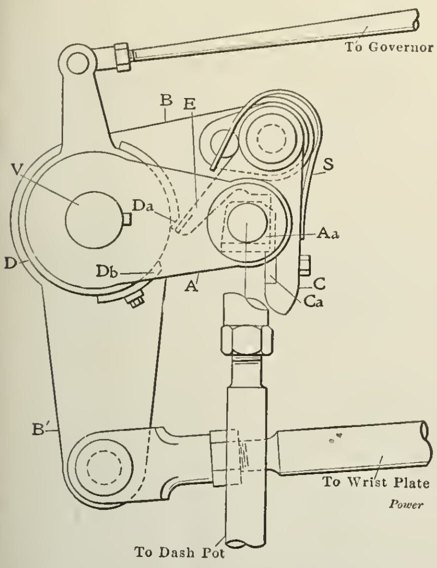


FIG. 8. REYNOLDS-CORLISS GEAR

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on *A*.
B B' = Bell crank, which carries hook *C* and tripping lever *E*. This bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by hook.
D = Knock-off cam mounted on the valve stem between *A* and *B B'*. The position of this cam is determined by the governor.
Da = Projection for tripping (operating cam).
Db = Projection for safety stop (safety cam).
E = Trip lever.
S = Spring.
V = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, thereby raising *A* until *E* comes in contact with *Da* (as shown in the figure). This forces *E* away and causes *C* to release *A*, which is immediately drawn downward by the dashpot and closes the valve.

In the first of these, release is effected by one leg of the hook sliding over the cam, and in the other the same result is attained by means of a roller working upon a cam. The relative merits and demerits of the two types, depending upon their adaptability to various kinds of service, are briefly indicated, but the reader will be left to judge these points chiefly for himself.

Fig. 5, showing the crab-claw gear previously alluded to, represents the first Corliss releasing mechanism to be extensively employed over a long period of time, and its appearance marked a distinct advance in the design of steam engines. For the slow speeds then in use, the device was well adapted.

In Fig. 6 is illustrated the latest gear designed by Corliss, which was used by the Corliss Steam Engine Company and later by its successor, the International Power Company, now the American & British Manufacturing Company.

The first important outcome from the crab-claw gear made by an inventor other than Corliss was the Reynolds-Corliss gear brought out by Edwin Reynolds in 1877, a distinguishing feature of which lies in the substitution for the crab claw of a V-shaped hook, as shown in Fig. 7. The advantage of this mechanism over the crab-claw gear is that the strains on the valve stem are greatly reduced. The hook takes hold of the lifting arm at the same distance from the center of the stem as does the dashpot, or very close to it, practically doing away with the bending moment due to the pull of the dashpot. The stem is mainly subjected to torsional strains caused by the friction and inertia of the valve. In the crab-claw gear, which is now a thing of the past, the stem had to withstand the bending moment due to the pull of the dashpot, also the forward strain resulting from the resistance of the valve; this caused the valve-stem bearing in the bonnet to wear rapidly.

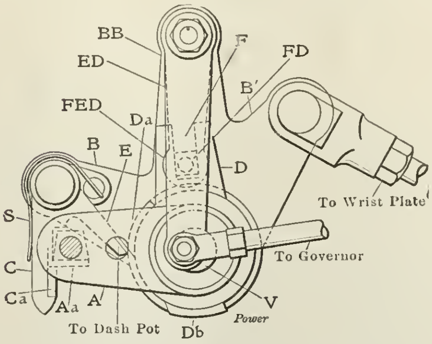


FIG. 9. REYNOLDS "WORLD'S FAIR" GEAR

A = Drop lever on crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on *A*.
B, *B'*, *BB* = Three-arm steam lever. *B* carries hook *C* and spring *S*; *B'* is connected to the wristplate; *B B* holds floating lever *E D* and arm *F E D* pivoted on the stud.
C = Hook.
Ca = Catchblock on *C*.
D = Slotted arm on cam sleeve (detaching cam). The position of this cam is determined by the action of mediums *E D*, *F E D* and *F D* (the latter slides up or down in *F*).
Da = Projection for tripping (operating cam).
Db = Projection for safety stop (safety cam).
E D = Floating lever pivoted on stud, which stud projects horizontally forward from the upper extremity of *B B*. The movement of this lever is controlled by the governor.
F = Slot in *D*, the position of which controls points of release and cutoff.
F D = Sliding block on *F E D*.
F E D = Arm cast in one piece with *E D* and suspended from far end of sleeve on *E D*. This sleeve projects horizontally back from the upper extremity of *E D*. The arm holds the block *F D* at its lower extremity and is pivoted on a stud.
S = Spring.
V = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, raising *A* until *E* comes in contact with *Da*. This forces *E C* outward and causes *C* to release *A*, which is immediately drawn down by the dashpot and closes the valve.

The crab claw was also heavy and was, therefore, adapted only to slow running engines. It required a stiff spring to bring it into engagement with the steam lever, as the whole weight of the claw had to be lifted. In the Reynolds gear, however, the weight of the hook directly assists in the operation, therefore it requires only a very light spring. Moreover, in this gear all the edges of the steel on the hook or steam arm can be used as working edges, which is not the case in the crab-claw gear.

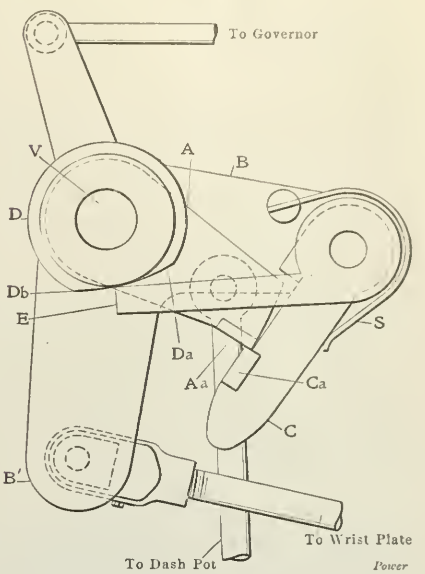


FIG. 10. FRASER & CHALMERS GEAR

A = Drop lever or crank arm mounted on the valve stem which it actuates, and connected to the dashpot rod.
Aa = Block carried by *A*.
B B' = Bell crank which carries a hook *C* on its inner side and a tripping lever *E* on its outer side. This bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by *C*.
D = Knock-off cam loosely mounted on the outer end of the valve stem, the position of which cam is determined by the governor.
Da = Projection for tripping.
Db = Projection for safety stop.
E = Trip lever.
S = Spring.
V = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, thereby raising *A* until *E* comes in contact with *Da*. This forces *E* away and causes *C* to release *A*, which is immediately drawn downward by the dashpot and closes the valve.

Since Edwin Reynolds brought out his gear, most Corliss engine builders, with the exceptions noted in the following remarks, have adopted his design outright or with slight modifications. At the pres-

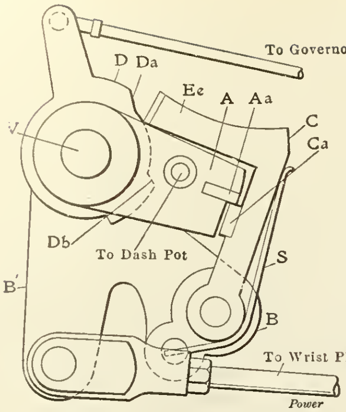


Fig. 11. Hoffman & Billings (Sederholm) Gear

A = Drop lever or bell crank, keyed to the valve stem inside of the outer bonnet bearing which it actuates, and connected to the dashpot rod.
Aa = Block carried by A.
$B\,B'$ = Double arm, which carries hook C. This arm receives its motion from the wristplate.
C = Hook journaled in B'.
Ca = Block carried by C.
Ee = Right angular extension and support of lever C, in position approximately horizontal.
D = Knock-off cam mounted on the valve-stem bearing, the position of this cam is determined by the governor.
Da = Rise to a surface concentric with groove surface.
Db = Rise to a surface concentric with groove surface.
S = Spring.
V = Valve stem.

Operation

When B' is pulled toward the wristplate, Ee ascends along the groove in D, keeping Ca and Aa in the relative position shown, until Ee comes in contact with Da, then it suddenly deviates from the valve-stem center and forces C outward, disengaging the blocks. This releases A, which is immediately drawn downward by the dashpot and closes the valve.

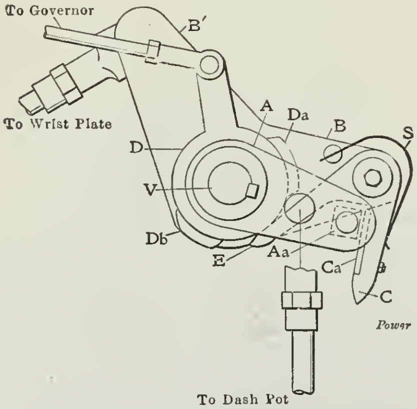


Fig. 12. Murray Gear

A = Drop lever or crank pin, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on A.
$B\,B'$ = Bell crank, which carries hook C and lever E. This bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by hook.
D = Knock-off lever mounted on the angular bracket between A and $B\,B'$, the position of this lever is determined by the governor.
Da = Knock-off cam.
Db = Safety knock-off cam.

Operation

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da. This forces E away and causes C to release A, which is immediately drawn downward by the dashpot, and closes the valve.

ent time the essential differences in the Corliss valve gears used by the great majority of builders are mainly due to workmanship and to designing the parts with more or less ample wearing surfaces.

The form shown in Fig. 7 has always proved satisfactory in operation and is similar to that now used by the Fulton Iron Works. As a standard it has been very closely adhered to by other leading builders, a fact which will be recognized at once by comparing it with the next few gears illustrated.

Fig. 8 is a modified form of that shown in Fig. 7, and has been used within recent years by Allis-Chalmers Company, for whom Edwin Reynolds long served as chief engineer. This gear is adapted to both medium- and long-range cutoff. Its application to the latter purpose is clearly indicated by Hubert E. Collins in his book on "Valve Setting," in which he states that the long-range gear is designed to give a maximum cutoff. The essential features of the steam valves are that they have a negative lap or opening when in mid-position, the cutoff being made entirely by the governor through the knock-off cam. In the engine which was built by Edwin Reynolds for the Columbian Exposition, an auxiliary eccentric operated levers on the governor in such a manner as to cause the knock-off block to follow the hook until cutoff occurred at the latest practicable point in the stroke; but in all essential details the releasing mechanism embodied the standard Reynolds-Corliss design. This gear is shown in Fig. 9.

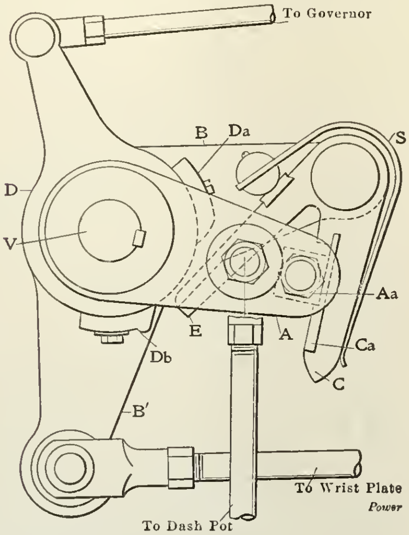


Fig. 13. Griffith & Wedge Gear

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on A.
$B\,B'$ = Bell crank which carries hook C and tripping lever E. This bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by hook.
D = Knock-off cam mounted on the valve stem between A and $B\,B'$. The position of this cam is determined by the governor.
Da = Projection for tripping (operating cam).
Db = Projection for safety stop (safety cam).
E = Trip lever.
S = Spring.
V = Valve.

Operation

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da. This forces E away and causes C to release A, which is immediately drawn downward by the dashpot and closes the valve.

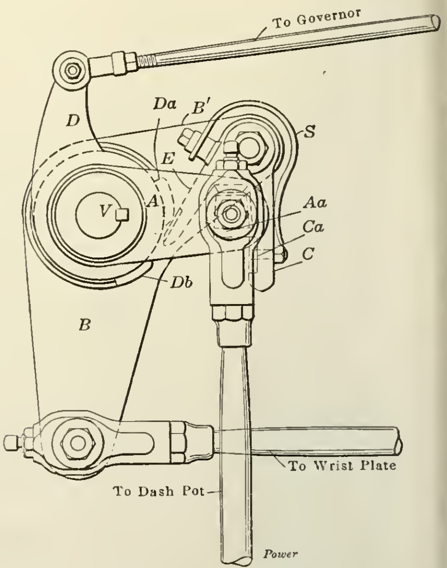


Fig. 14. Cooper Gear

A = Drop lever or steam crank, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Block-pin block carried by A.
$B\,B'$ = Bell crank, which receives its motion from the wristplate and carries hook C.
C = Hook.
Ca = Pick-up block carried by hook.
D = Knock-off cam mounted on the valve stem between A and $B\,B'$. The position of this cam is determined by the governor.
Da = Knock-off clip.
Db = Safety clip.
E = Trip lever.
S = Spring.
V = Valve stem.

Operation

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising A until E comes in contact with Da. This forces C away, thus causing C to release A, which is immediately drawn downward by the dashpot and closes the valve. When the governor reaches its lowest position, the projection Db prevents Ca from engaging Aa and keeps the valve closed. Similarly when the governor reaches its highest position, Da is moved around until Ca is kept from engaging with Aa and opening the valve.

Practically identical with Fig. 8 is the gear used by the Bates Machine Company, except for the fact that the wristplates are eliminated and the motion imparted directly by the eccentric rod, a feature which was also used by Reynolds.

The Bates Machine Company at one time employed a form of valve gear which differed radically from any of those described in this article, a folding mechanism being substituted for the usual hook or crab-claw rig; but it possesses merely historical interest and is not of sufficient importance to be illustrated here.

Very similar to Fig. 7 is the gear illustrated in Fig. 10, which was formerly used, among others, by Fraser & Chal-

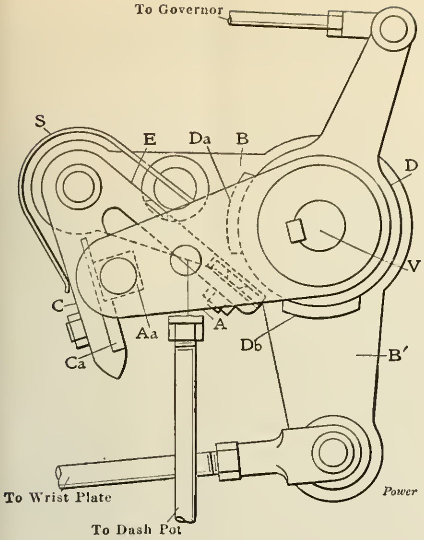


Fig. 15. Heilman Gear

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on *A*.
B B′ = Bell crank, which carries hook *C* and tripping lever *E*. This bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by hook.
D = Knock-off cam mounted on the valve stem between *A* and *B B′*; its position is determined by the governor.
Da = Projection for tripping (operating cam).
Db = Projection for safety stop (safety cam).
E = Tripping lever.
S = Spring.
V = Valve stem.

Operation

When *B′* is pulled in the direction of the wristplate, *Ca* engages *Aa*, thereby raising *A* until *E* comes in contact with *Da*. This forces *E* away and causes *C* to release *A*, which is immediately drawn downward by the dashpot and closes the valve.

mers, of Chicago. In operation it is practically the same. A distinguishing feature of the engines upon which it has been used lies in the fact that the dashpots are set at a slight deviation from a right angle with the cylinder foot. It was claimed by the builders, whose plant has since been merged into the works of another company, that this position allows greater leverage on the point of the hook when raising the plunger of the dashpot and opening the valves, and at the same time affords the plunger greater leverage over the valve during the first part of its downward movement when closing the valve, thus insuring a very rapid closing of the valves and consequently a quick and sharp cutoff.

Fig. 11 shows a gear invented by E. T. Sederholm in 1885 for the Hoffman & Billings Manufacturing Company, of Milwaukee, which later appears to have been used by the Rarig Engineering Company. In this the drop lever is keyed on the valve stem inside of the outer bearing of the bonnet. The knock-off cam is mounted on the bonnet and projects sideways in such a manner that the hook comes in contact with it. This gear is noteworthy chiefly as being the first to have the valve stem supported at its outer end.

Fig. 12 is a sketch of the gear made by the Murray Iron Works. A feature of this is the fact that the knock-off lever is fitted with double cams for use with an improved automatic safety stop.

Fig. 13 shows the gear used by the Griffith & Wedge Company; and Fig. 14 has been sketched from the mechanism shown in an engraving of one of the C. & G. Cooper Company's engines. Fig. 15 illustrates the releasing mechanism employed by the Heilman Machine Works, and Fig. 16 is the Frick Company's gear.

These have all been successful in operation but follow so closely the standard design of Fig. 7 as to require no extended comment. Among other gears of the same general design are those which have been used by the Hooven, Owens, Rentschler Company; the William Todd Company; the Lane & Bodley Company; Lake Shore Engine Works; Knowlson & Kelley; St. Louis Iron and Machine Company; Watertown Engine Company; Robert Wetherill & Co.; H. N. Strait Manufacturing Company and Wellman-Seaver-Morgan Company.

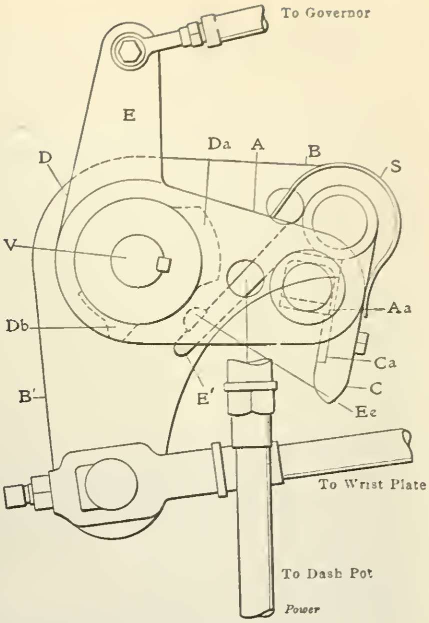


Fig. 16. Frick Gear

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock, which is a part of *A*.
B B′ = Bell crank, which carries the hook or latch *C* and receives its motion from the wristplate.
C = Hook.
Ca = Block carried by hook.
D = Knock-off cam mounted on the valve bonnet between *A* and *B B′*. The position of this is determined by the governor.
Da = Projection for tripping (operating cam).
Db = Projection for safety stop (safety cam).
E = Trip lever.
E′ = Part of *C*.
Ec = Knock-off button on *E′*.
S = Spring.
V = Valve stem.

Operation

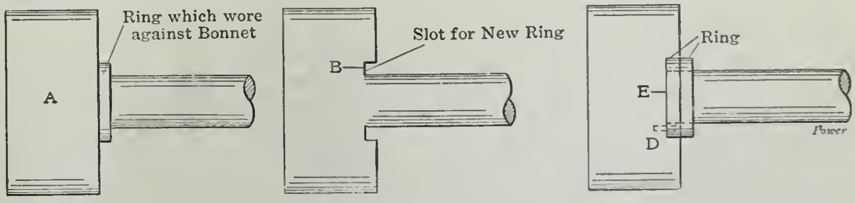
When *B′* is pulled in the direction of the wristplate *Ca* engages *Aa*, thereby raising *A* until *Ec* comes in contact with *Da*. This forces *Ec* away, thus causing *C* to release *A*, which is immediately drawn downward by the dashpot and closes the valve.

(Other gears in common use will be described in next week's number.)

Corliss Valve Stem Repair

While talking shop with an engineer at one of the cotton mills located at Anniston, Ala., a method of repairing the valve stem of the Corliss engine was shown the writer, the original design of which is illustrated at *A* in the accompanying illustration. The collar, originally made solid with the valve stem, was designed to have a bearing against the valve-stem bonnet, but gave trouble from wearing, which allowed considerable end play to the valve.

In order to eliminate this trouble, the engineer turned off the original collar and cut a recess in the head of the valve stem, as shown at *B*, in which he fitted a ring of soft steel so that the outer edge came about even with the edge of the valve-stem head. Another ring of the same size was made an easy fit on the valve stem, to form a bearing with the ring *E*, set in the head, both being of the same size as the original ring on the stem, so as to fit in the recess in the bonnet.



Showing Valve-stem Repair

By this arrangement the wear is now removed from the bonnet and placed on the two rings, which are easily removed when worn, thus eliminating the expense of a new valve stem. The inner ring is prevented from turning by the small dowel pin shown at *D*.

Compact Condensing Installation

By Warren O. Rogers

In a small plant a change from reciprocating engines to turbines was made and it was decided to install condensers. It was necessary to economize in floor space, so the type of condensing unit shown in the illustrations was selected.

The power plant of the Bristol (Tenn.) Gas and Electric Company is not noted for its size, but it ranks among the pioneers in the use of the new Leblanc condenser, which, although No. 13, has been giving the best of satisfaction.

The main generating units consist of two 300-kilowatt, three-phase, 60-cycle generators direct connected to Westinghouse turbines. The most interesting feature of the plant, however, is the condenser installation. When it was decided to put in turbine units to supplant the reciprocating engines, the type and design of condenser had to be considered.

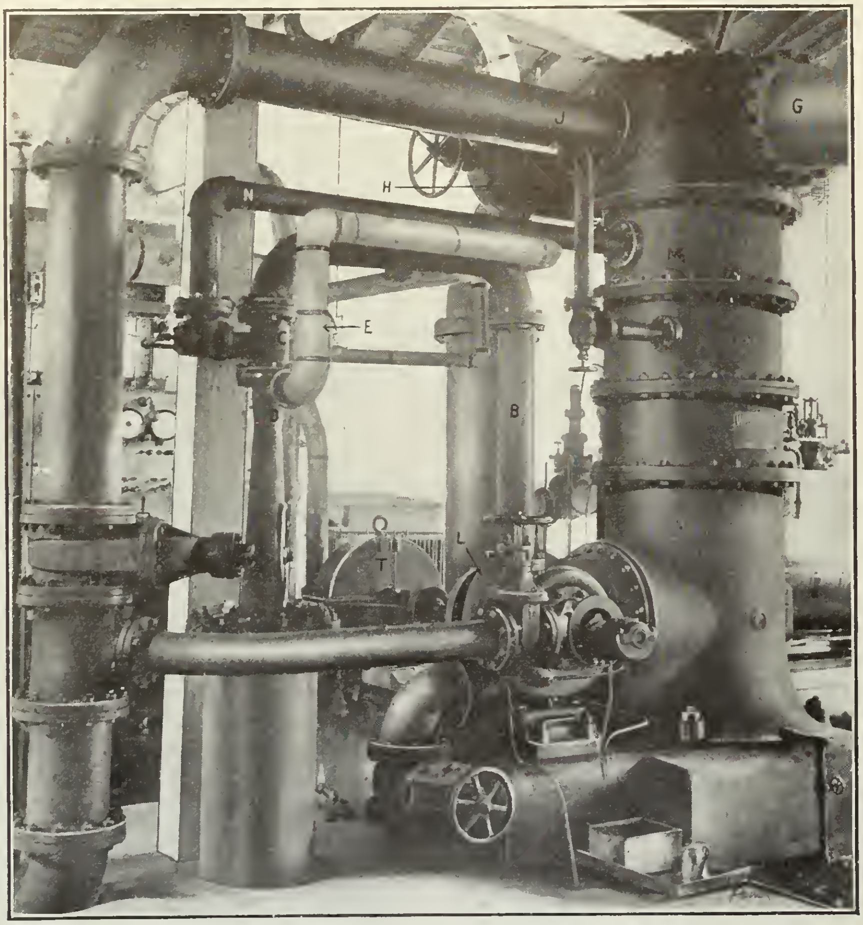


Fig. 1. Leblanc Condenser in Bristol Plant

It was decided to install the Leblanc condenser shown in Fig. 1. It was located at the head end and midway between the two turbines. Instead of occupying a large floor space and requiring several foundations for the air pump and other auxiliaries, a light cement foundation was all that was required, and the condenser was placed upon it as shown.

When the condenser is in operation the water and air are removed separately by means of a pair of small turbine-type rotors attached to a shaft extending through the casing in which the rotors are placed. This casing is connected to the lower portion of the condensing chamber and is made slightly larger than the exhaust opening of the turbine.

The air pump operates on a principle differing from usual condenser practice. A series of slugs of water, between which are small pockets of air drawn from the upper portion of the condenser body, are forced through the discharge nozzle. In passing through the small nozzle the air mixes with the water, but owing to the velocity at which the water is traveling, no air can return to the condenser.

The condensing water is taken from a nearby creek at a lift of 18 feet, and the water used in the air pump is drawn from the main inlet pipe by suction into an annular chamber *A*, which is overhung by the bucket of the pump rotor. As the water passes out of an outlet connecting with the pipe *B*, it is projected downward in the form of slugs, with the small pockets of air between, and the whole is carried to the nozzle *C*. This nozzle has an auxiliary nozzle *D*, to which is connected a steam pipe *E*. When the engineer is about to start up the condenser, steam is turned into the auxiliary nozzle to create sufficient vacuum (about 10 inches) to cause water to flow through the air pump, which is then started. If the water supply were taken from a point above the air pump, it would not be necessary to use the steam jet in starting the condenser.

Fig. 2 illustrates the only moving element in the condenser, the two rotors of the pumps. This rotor shaft is driven by a steam turbine, as shown at *T*, Fig. 1, but if desirable, a motor can be used. In the Bristol plant the turbine running the pumps has 30 horsepower capacity when running at 2500 revolutions per minute.

Referring to Fig. 1, the exhaust steam enters the condenser through either of the pipes *G* or *H*, depending on which turbine is being used. The cooling water enters through the pipe *J*, and is projected downward through a spray nozzle lo-

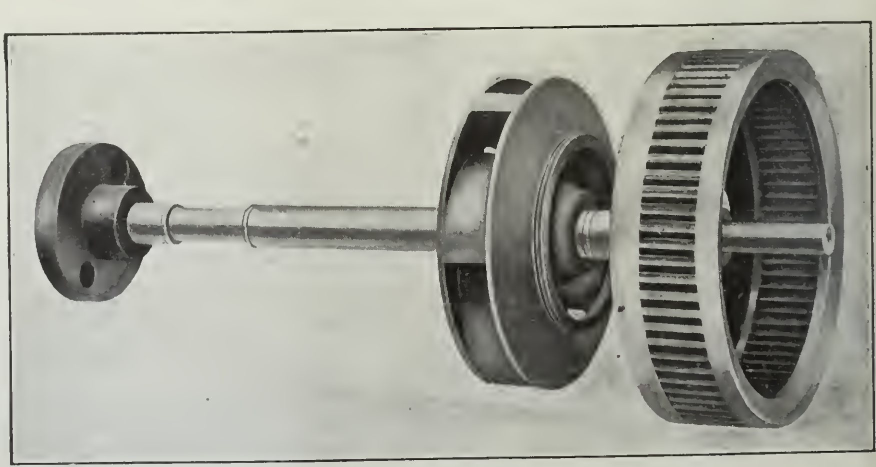


Fig. 2. Only Moving Element in the Condenser

cated within the condenser in the section *R*. This injection water and condensed exhaust steam, which is drawn down and condensed by the water, after passing through the nozzle, flows to the centrifugal discharge pump *L* under a head of about 2.5 feet. The space *M* in the condenser, just below the nozzle and above the water passing through the discharge pump, is occupied by water vapor and air released from the injection water and from the exhaust steam. This space is connected to the air-pump nozzle *C* by means of the pipe *N*, and the vapor and air are removed as already described. The Bristol plant presents a good illustration of the advantages of this type of condensing unit.

Handy Chart for Barometer Readings

By G. A. Glick

Anyone who has worked with a calorimeter or with any barometer reading in inches of mercury will know the inconvenience incurred in changing these readings to a form in which they can be used. It is the object of the chart to facilitate such transformations, and by its use one is able to change inches of mercury to pounds per square inch; to find the corresponding boiling temperature at any barometer reading; and to find the altitude corresponding to any barometer reading. The chart is plotted only between 28 and 31.5 inches of mercury, for the reason that the larger number of calculations come within this range, and the small range permits considerable accuracy, that otherwise would not be obtainable. The scales used are all rather fine and this will permit close readings, which are liable to be more accurate than can be obtained on a slide rule.

A few constants are given below to permit calculations beyond the limits of the chart. With regard to altitude it may be said that, as a rough approximation, the pressure of the atmosphere decreases one-half pound per square inch for every 1000 feet of ascent, or in descending toward the center of the earth the barometer readings increase by 1 inch of mercury for every 900 feet drop. To change inches of mercury to pounds per square inch, the well known rule of multiplying the reading in inches by 0.4908 gives the corresponding pressure in pounds per square inch. Roughly speaking, pure water will boil at 1 degree Fahrenheit less for every 550 feet ascension up to an elevation of one mile, after which the increment becomes 560 feet.

The chart can be used interchangeably, that is, pounds per square inch can be used instead of inches of mercury to find the corresponding altitude and boiling point, or even the barometer reading, if it is desired. In this case all the values would have to be reduced to inches of mercury as a common standard. For example: Suppose the barometer reads 29.925 inches of mercury. Looking on the chart at this reading we find the corresponding pressure to be 14.68 pounds per square inch. The altitude corresponding to this barometer reading is 70 feet, and water will boil at 211.87 degrees Fahrenheit. Care must be taken that each reading is taken from the proper curve plotted for that condition.

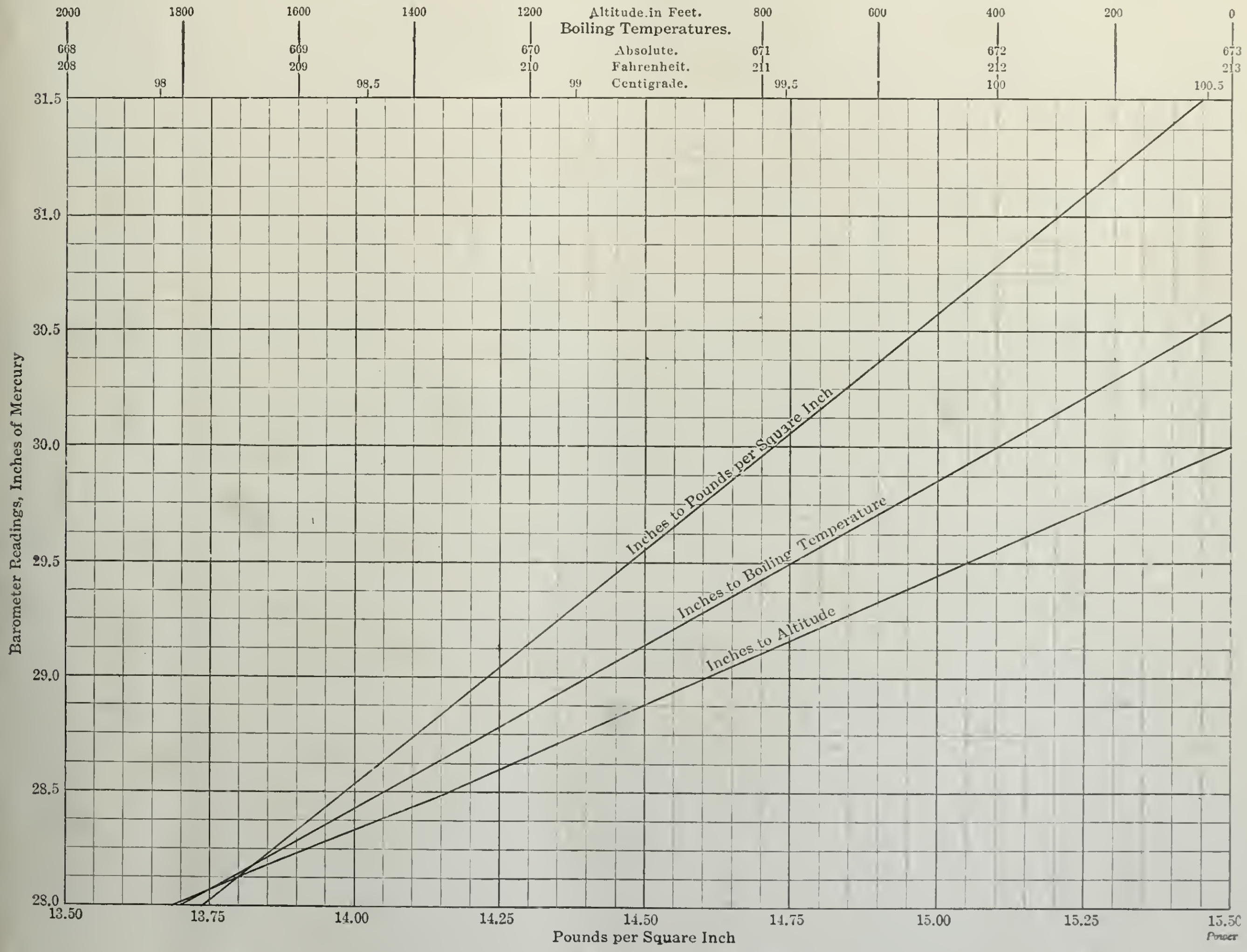


Chart for the Conversion of Barometer Readings

Uncle Pegleg's Philosophy

By again using the platform car as an illustration Uncle Pegleg shows that a reaction nozzle must run as fast as the jet to get all the energy out of the water.

In fulfilment of his promise to explain why it is that a reaction nozzle must run as fast as the jet to get the energy all out, Uncle Pegleg produced at our next talk a sketch like Fig. 1.

"I've fixed this up," he said, "to represent the case where the water is taken aboard as fast as it is played out behind, so the process can be a continuous one. That tank we talked about the other day would run dry after a little and then the car would have to stop and take on dead water and store a lot of energy in it to get it into motion, and fool you about stopping it. Here's a long trough between the tracks and a scoop picking the water up the way they do it in locomotives. If there wasn't any friction the water would simply run in the front end and out the back as fast as the car ran. Is that clear?"

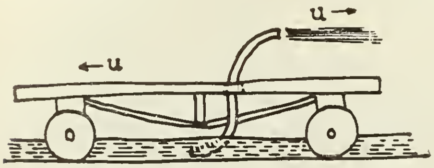


FIG. 1

I could see that, with the exception of the little work needed to lift the water which he told me to forget along with the friction.

"Supposing the car to be running at *u* feet per second, you would have a jet coming out of the nozzle with a velocity of *u*, eh?"

I agreed to it.

"But it wouldn't be pushing the car any, would it? It would be coming out because something pushed the pipe through it, just as it would come out of the end of a straight tube that you pushed through the tank.

"Now, if you put in a pump here, Fig. 2, and hiked that water up to *v* feet per second you wouldn't have to do so much work on it to get it up to *v* feet as though you started with it standing still, would you? If it is already moving *u* feet in a second, all the motion you have to put into it with the pump is *v-u* feet a second. That's clear, isn't it?"

I allowed that it was.

"Well, now, it don't take so much force to accelerate a thing fifty feet a second as it does a hundred, does it?"

"No."

"And instead of taking $w v \div g$ pounds to get the jet up to a velocity of *v* feet per second as it would if the water started from rest, it would take only $w\ (v - u) \div g$ pounds."

I thought over his illustration of sliding the cake of ice on the pond, remembered that one pound of force would accelerate a pound of matter *g* feet per second and could see that to accelerate a weight *w*, *a* feet per second, would take a force equal to $w a \div g$. Of course, if the stuff was going with a velocity of *u* to start with and had to get up to *v* in a second, the acceleration was *v-u*. So far, so good.

"If the pump only has to push that hard on the water, the water will push back only that hard on the pump. Action and reaction are equal and opposite."

"Then the faster the nozzle moves the less it pushes," I said.

"Yes, in this case."

"I said that before and you said it didn't," I reminded him.

"But that was a different case. That nozzle wasn't taking in water in motion. It was drawing from a tank at rest with reference to itself," he replied.

"Now, let's see what we've got," he went on. "We have a jet reacting or pushing against the car with a force of $w\ (v - u) \div g$ pounds, and the car moves *u* feet per second, then the work done is as follows:

$$\text{force} \times \text{space} = \text{work}$$

$$\frac{w(v-u)}{g} \times u = \text{work}$$

"Let us try it with some figures and see how it works out. Suppose the jet to be big enough to discharge 32 pounds a second, with a velocity of 100 feet a second, then the work done on the car would be:

$$\frac{w\ (v-u)}{g} \times u = \frac{32\ (100-u)}{32} = 100u - u^2$$

"Give *u* the values in the first column and you will get the values in the fourth column for the work expended on the car:

"Figure it out for 10 feet velocity of the car so you can see how it was done."

I put down

$$u = 10$$
$$100 \times u = 1000$$
$$u^2 = 100$$
$$100u - u^2 = 1000 - 100 = 900 \text{ foot-pounds}$$

"You see," he said, "when the car don't move, that is, when $u = o$ there is no work done on it. The number of foot-pounds of work that is done on the car runs from 900 per second at a velocity of 10 feet up to 2500 foot-pounds per second at 50 feet or half the velocity of the jet; after this it runs down again as the fourth column shows until, when the velocity gets up to that of the jet, the energy has got back to nothing. Energy is the product of force and space. When the car is standing still the jet is pushing its hardest, but there is no energy absorbed by the car because there is no motion. When the car gets to going as fast as the jet there is plenty of motion but no force."

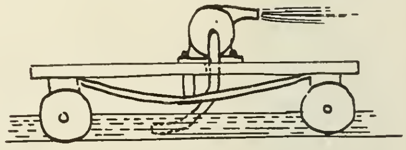


FIG. 2

"It looks then as though it got the most power at half speed just like the impact wheel," I remarked.

"So it does."

"But you started in to prove that it got the most out when it ran as fast as the jet."

"No, I didn't, my boy. I started in to prove that it came nearest to getting all there was in the jet out of it when the nozzle ran as fast as the jet did. The faster the car runs the less work the pump can get in on the water for a given jet velocity, so the less there is to take out. Look at that fifth column. The energy in the water, due to the velocity with which it approaches the jet is $w u^2 \div 2g$, isn't it?"

"Well," I argued, "There isn't any energy in the water lying in the trough."

"Take it this way, then. The total energy of the jet is $w v^2 \div 2g$. See that?"

"Yes."

u	100 u	u^2	Work Done on Car.	Work Done on Jet.	Efficiency, Per Cent.
0	0	0	0	5,000	0
10	1,000	100	900	4,950	18.2
20	2,000	400	1,600	4,800	33.3
30	3,000	900	2,100	4,550	46.2
40	4,000	1,600	2,400	4,200	57.1
50	5,000	2,500	2,500	3,750	66.7
60	6,000	3,600	2,400	3,200	75.0
70	7,000	4,900	2,100	2,550	84.0
80	8,000	6,400	1,600	1,800	88.9
90	9,000	8,100	900	950	94.7
100	10,000	10,000	0	0	0

"Well, the pump didn't have to get up the full velocity v because the water would flow out of the nozzle with the velocity u due to the scoop. It was moving relatively to the nozzle with a velocity u and it had in it, relatively to the nozzle, $wu^2 \div 2g$ foot-pounds of energy per second."

I gave in.

"Then all the energy the pump had to put in was

$$\frac{w(v^2-u^2)}{2g} = \frac{32(100^2-30^2)}{2\times 32}$$

See that?"

I saw that it must be so if the thing over to the right of the equal mark was the difference between the other two. The old man told me to try it some time with figures instead of letters, and I did, and it came out the same.

"Just figure with that formula the energy put by the pump into the jet when the velocity of the car equals 30," he said.

I worked it out so, using the same values for v w and g as in the other columns:

$$\frac{wv^2}{2g} - \frac{wu^2}{2g} = \frac{w(v^2-u^2)}{2g}$$

$$\frac{10000-900}{2} = \frac{9100}{2} = 4550.$$

He showed me that this was the same that he got in the fourth line of his fifth column and said that all the other lines were figured the same way.

"You see," he said. "the amount of energy that the pump puts into a jet of that size at 100 feet runs from 5000 foot-pounds per second when the water coming to it has no velocity (the scoop no motion in the trough) down to nothing when the water is scooped in with the same velocity with which it runs out. (Remember that the quantity w that is taken in per second is the same in any case, but its velocity may be more or less.) Notice, too, that although the work done on the car becomes less after its speed exceeds that of the jet, the *proportion* of the work which is put into the jet by the pump (column 5), which is applied to the propulsion of the car (column 4) gets greater, the greater the speed. I have figured these in percentages in the last column by dividing the figures in column 4 by those in column 5, and multiplying by 100 to give it in percentage. When you get up near the velocity of the jet you are getting out the largest percentage of the work in the jet, but there wouldn't be much if you got the whole of it."

As he folded up his papers, he said: "Thought you had the old man stuck, didn't you?" and as I grinned he admitted, "Well, you did for a while."

A Siphon Heating System

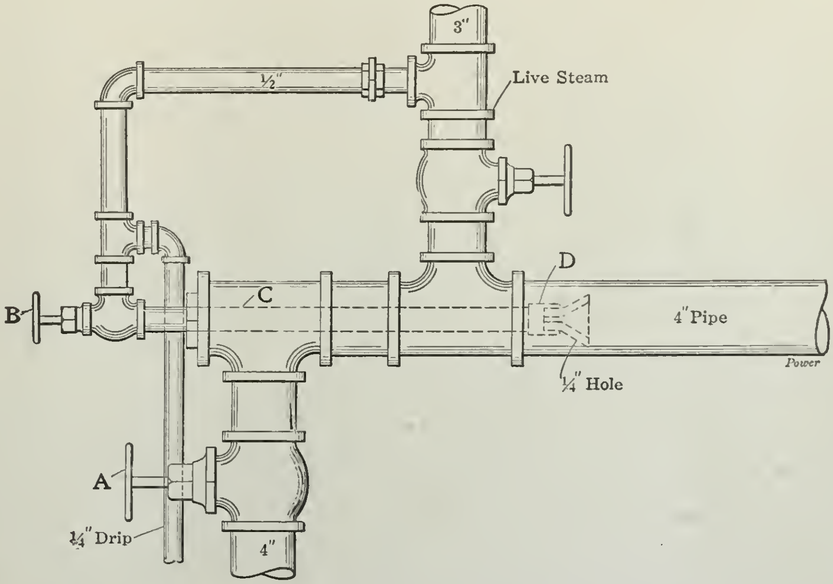
In the engine room of a cotton mill in the town of Anniston, Ala., a unique method of utilizing the exhaust steam from an engine unit is employed to heat the building which is a structure three stories high and several hundred feet long. The circulation of steam is effected without the aid of a vacuum pump or a back-pressure valve on the exhaust pipe.

When a heating engineer visited the plant, he suggested that a vacuum pump be installed to draw the exhaust steam through the heating coils in the factory, but the engineer decided that a better scheme could be utilized, and, therefore, devised a siphon system, which the "wise ones" claimed would not work; but it does.

A particular arrangement in this plant is that there are two exhaust pipes instead of one. The engine is made with a 14-inch exhaust outlet reduced to 12 inches from a tee placed between the cylinder and heater, and from the heater to the atmosphere. From the tee an 8-inch pipe extends out and up to a point above the roof, exhausting to the atmosphere. Just beyond the tee a 4-inch pipe is connected to the exhaust pipe which connects to the 1½-inch heating pipes in the mill.

The illustration shows the arrangement of the piping and the method employed to get the steam to circulate through the heating pipes. The 4-inch pipe from the engine exhaust line is fitted with a globe valve, in which is fitted a nipple to a 4-inch tee. In one end of the tee is a reducing plug, in which is fitted a ½-inch pipe. This pipe is connected to the 3-inch live-steam line, which also connects with the 4-inch pipe line leading to the factory. This live-steam line is used on the heating system when the engine is not running. The steam, when so used, is reduced to a pressure of 85 pounds per square inch.

age of live steam. The current of steam in issuing from the flared end draws exhaust steam from the exhaust pipe, which has from 1½ to 2 pounds back pressure, and forces it through the heating coils in the mill. By this method the engineer overcame the necessity of purchasing a vacuum pump, and at the same time obtained most satisfactory results with the steam jet.



Siphon Heating System

The operation is simple. When exhaust steam is required for heating, the valves *A* and *B* are opened, the valve *A* being on the exhaust-steam line, and valve *B* regulating the live steam for the steam jet *D*. The ½-inch pipe *C* extends in past all joints and is fitted on the inner end with a coupling, in which a flared end piece is screwed. This nozzle has a ¼-inch hole through its center for the passage of live steam.

The propensity of English builders of gas engines to cling to the hit-and-miss method of governing is manifest in the British section of the Brussels Exposition.

Elements of Compression System

By F. E. Matthews

The principles of operation of the compression refrigerating system are explained and the essential members of the direct-expansion and brine systems illustrated. Systems with semi-automatic and complete automatic control are given attention.

Fig. 1 is a diagrammatic representation of the essential members of a complete compression-refrigerating system in which *E* represents the direct-expansion coil in which the working medium is evaporated as in the test tube, *P* the compressor or pump for increasing the pressure of the gasified ammonia, *C* the condenser for cooling and liquefying the gasified ammonia, and *V* a throttling valve by which the flow of liquefied ammonia under condenser pressure is controlled as it passes from the receiver *R* to the expansion coil *E*, in which a materially lower pressure is maintained by the pump in order that the refrigerating medium may boil at a sufficiently low temperature to absorb heat from, and consequently refrigerate, the surrounding air which is already cold. In practice the systems are somewhat more elaborate.

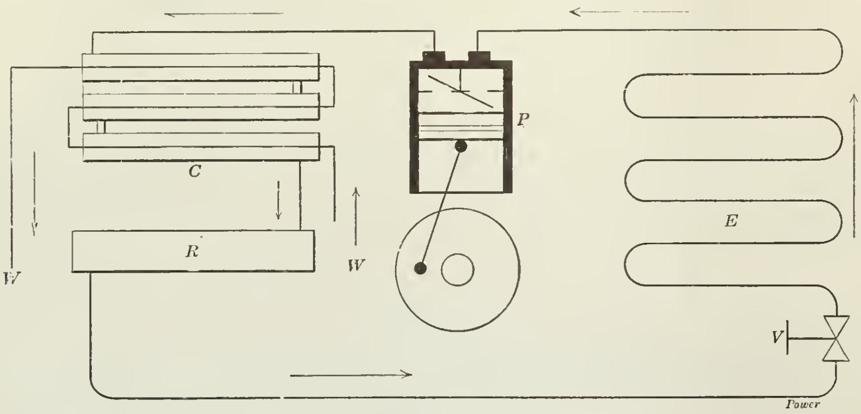


Fig. 1. Essential Members of Compression System

In order that some work of refrigeration may be carried on while the plant proper is shut down either in case of accident or in order to avoid the expense of skilled attendance during a part of the twenty-four hours, the brine-circulating system is employed. Such a system is represented in the conventionalized diagram, Fig. 2. In addition to the usual members of the direct-expansion refrigerating system represented in Fig. 1, the brine system employs a brine tank *T*, Fig. 2, a series of air-cooling or brine coils and a brine pump *B B*. In this case the initial cooling effect of the evaporating ammonia is expended in the brine which is circulated by the brine pump through the air-cooling coils installed in the cold-storage room *R*.

Salt, either sodium chloride ($NaCl$) or calcium chloride ($CaCl$) is required for making the brine, and suitable insulation must be provided for the brine tank, the brine cylinders of the pump and all brine piping outside of the cold-storage compartments in order to reduce to a minimum the losses due to the radiation of cold, or more accurately speaking, the absorption of heat. In addition to the first cost of the previously mentioned items the operating expense of power required to operate the brine pump must be considered; first, for the actual transfer of brine from the brine tank through the cooling coils back to the tank; second, to overcome the friction of the brine in traversing the above cycle; third, to produce sufficient additional refrigeration to make up for the heating effect produced mechanically by the circulation of brine and the heat actually absorbed through the brine tank and brine-piping insulation; and fourth, to make up for the greatly reduced efficiency occasioned by the necessity of operating the refrigerating plant proper at a materially lower back pressure in order to produce a sufficiently lower ammonia temperature to make up for the second heat transfer encountered between the atmosphere of the cold-storage compartments and the ammonia gas. The increased operating expense directly resulting from the inherent low efficiency of the brine system is sometimes as high as 25 per cent., and isolated instances have been noted in small plants where the cost of circulating the brine alone was almost as great as that of cooling it.

Instead of the brine system some builders employ what is known as the "congealing-tank" system, represented diagrammatically in Fig. 3. The first cost of the brine pump and the power required to operate it, and the cost of brine pipe and tank insulation, as well as the losses through the same, are avoided in this system by virtually splitting up the main brine tank and installing the pieces commonly called "congealing tanks" in the several cold-storage compartments. In this case the heat-absorbing surface of the several smaller tanks entirely replaces that of the brine coils. Here, however, the desirability from a mechanical standpoint of making the tanks as small as possible debars the carrying of sufficiently large volumes of brine to provide for refrigerating the rooms for any great length of time, by the rise in temperature of the brine. This involves only the specific heat of the brine, the amount of refrigeration produced being the product of the pounds of brine, the number of degrees rise in temperature and the specific heat of the brine.

$$B.t.u. = lb.\ brine \times (t - t_1) \times specific\ heat\ of\ brine.$$

In order to store more refrigeration in the comparatively small volume of the congealing tanks a weak solution of brine is employed, which, in freezing, stores refrigeration proportional to the number of pounds of brine frozen multiplied by the latent heat of fusion of the brine ice plus the cold required to chill the brine down to the freezing point.

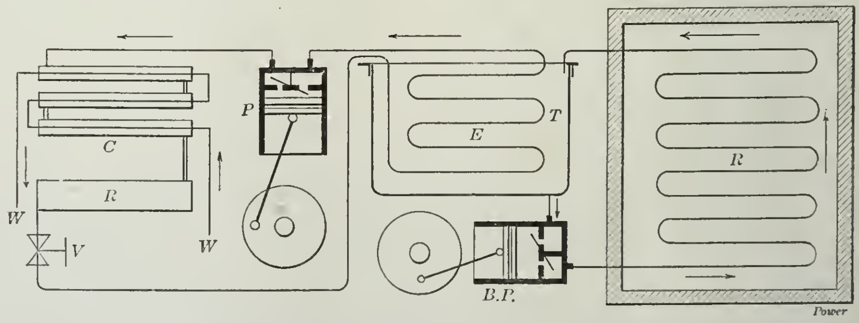


Fig. 2. Ammonia System May Be Shut Down and Brine Circulated

This expression does not provide for the small additional amount of refrigeration required to cool the brine ice below its freezing point. When the temperature of the ice is below the freezing point the additional refrigeration will be found from the expression

lb. ice × specific heat of the ice × $(t_1 - t_2)$.

In the above expressions t is the temperature of the brine before cooling, t_1 the temperature at which the brine freezes and t_2 the temperature to which the ice is cooled after freezing.

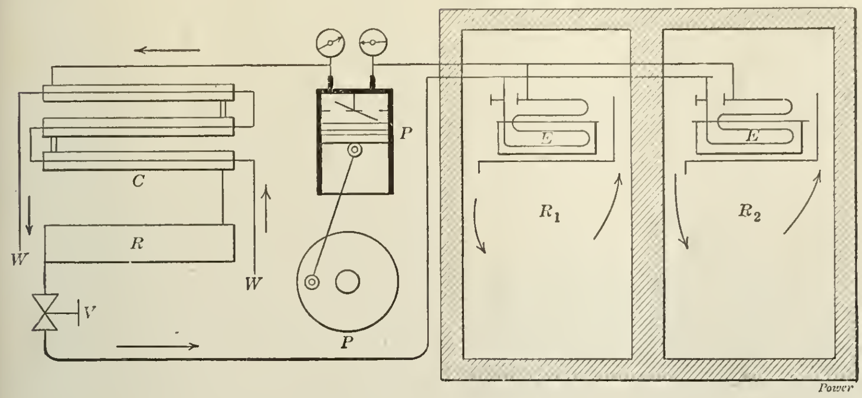


Fig. 3. Congealing-tank Brine System

In order that the rooms may be cooled more quickly when the refrigerating plant resumes operation after several hours of inaction, it is often customary to install from one-third to two-thirds of the total expansion piping outside the tanks in direct contact with the atmosphere of the rooms, the remaining two-thirds or one-third being submerged in the weak brine of the congealing tanks which have been substituted to take the place of one large brine tank.

It is evident from the foregoing that so far as decreased efficiency entailed by the double-heat interchange of the brine-circulating system is concerned, the congealing-tank system is only a compromise. While a part of the piping cools the air by direct radiation, the greater part must transmit cold first to the surrounding brine, or ice, as the case may be, which medium in turn must transmit it on through the walls of the tanks to the air; and a sufficiently low ammonia evaporation or "back pressure" with its entailed loss of efficiency, must be maintained in both the submerged and the exposed coils, to carry out the double-heat interchange in the former. The fact that the refrigeration stored up during the hours of operation of the plant for use during the hours of rest is in the form of a coating of ice on the coils instead of brine kept in constant circulation, further increases the necessity for a lower back pressure; first, because ice is a poorer conductor of heat than brine; second, because the congealing-tank surface as well as that of the expansion pipes, is often insulated with ice; and third, because the stagnant brine, or, what is even worse, ice in the case of the congealing-tank system, absorbs and gives up heat less readily than the moving brine.

As it is mechanically impracticable to make the thin, light congealing tanks indefinitely continuous, as is done in the case of ammonia or brine-cooling coils, the ammonia-expansion coils installed in such tanks must be made shorter. This necessitates the use of an increased number of return bends or fittings for the installation of the system as a whole and accordingly tends to increase the initial cost of the expansion piping for both material and labor.

While local conditions must be the deciding factors in every case, congealing-tank systems of small capacity usually compare quite favorably with the brine-circulation systems of similar capacities; but it must be remembered that both systems have the disadvantage of being inherently inefficient because of the necessity of operating under lower back pressures in order to produce the correspondingly lower temperatures required to effect the double-heat transfer. At the lower pressures fewer pounds of the refrigerating medium are passed through the compressor per cubic foot of piston displacement and a greater number of pounds are necessary to produce a given refrigerating effect, both of which conditions make it necessary to speed up or otherwise increase the displacement of the compressor in order to produce the same amount of cooling duty.

By the direct expansion of the refrigerant, cold is produced just where it is required and more nearly at the temperatures required, and incidentally, aside from the saving in power required to operate the compressor and to pump the brine, a saving in investment for a suitable room and foundations for brine tanks and pumps as well as for the tanks and pumps themselves, necessary piping and insulation, etc. Thermal losses due to radiation through the insulation of these members and through that due to the heating affect of pumping the brine are entirely eliminated.

The type of system best adapted to a given set of requirements can be determined only after carefully considering the relative importance of such factors as cost of power, cost of attendance, allowable temperature variation, availability of capital and probable earning power of plant.

While the use of electric power undoubtedly reduces the duties of the attendant, safety and the necessity of adjusting hand-expansion valves require more or less constant attention, and the slight additional attention required by the steam or combustion engine is usually too slight to make up for the usual comparatively high cost of electric power.

The application of reliable safety devices, however, which protect the plant in case of abnormal pressure resulting from failure of water supply or the accidental closing of the wrong valves, together with a reliable automatic expansion valve, eliminates these two most important duties of the attendant. When the compressors are of the inclosed-crank self-oiling type, and electric motors are employed, very little attention need be paid to lubrication, and the item of attendance becomes very small indeed.

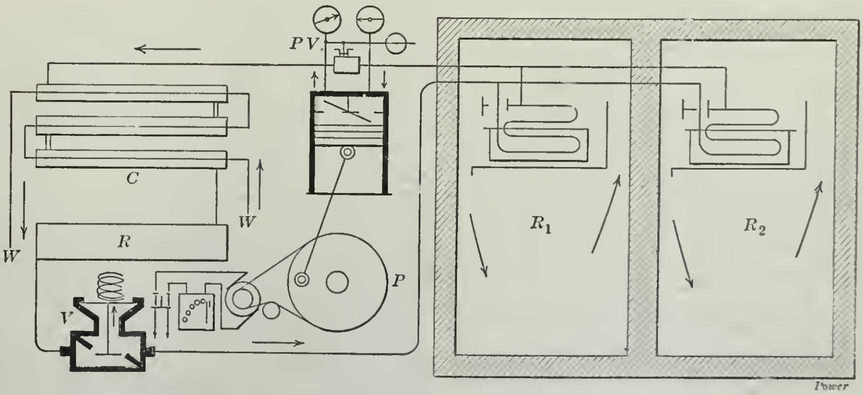


Fig. 4. Semi-automatic Refrigerating System

The semi-automatic system usually employed to meet such conditions is illustrated diagrammatically in Fig. 4. It will be noted that this system is essentially the same as the congealing-tank system illustrated in Fig. 3, except for the addition of an automatic expansion valve *V* and some suitable safety device *P V* for preventing the occurrence of abnormal pressures. In the case of other than electrical power, this may be a simple mechanical device, such as a spring or a weight-loaded "pop" valve between the high-pressure and the low-pressure sides of the system. In the case of electric power a device actuated by pressure such as is shown diagrammatically in Fig. 5, may be employed. Such a device consists essentially of either a Bourdon tube or a diaphragm *D* arranged to move a lever arm *A* in such a way as either to disengage a latch holding a spring-opening electric switch closed or, where there is a no-voltage release coil in the motor-controlling circuit, the device may be arranged as at *P D*, Fig. 6. Here the diaphragm *D* bowing outward under abnormal pressure *P* is employed simply to make an electric contact *C* for short-circuiting a no-voltage coil which, being thereby deënergized, allows a spring to open the motor circuit and interrupt the operation of the refrigerating system, just as would occur in case of failure of line voltage. It is almost unnecessary to add that a thermostat *T* may be similarly employed to stop the refrigerating machine when the desired temperature has been produced in the cold-storage compartments.

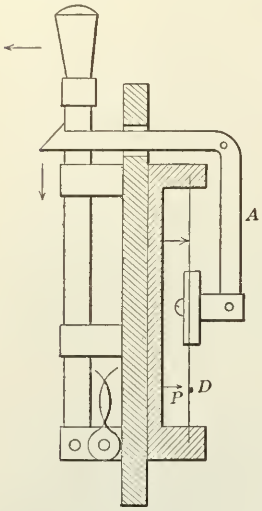


Fig. 5.

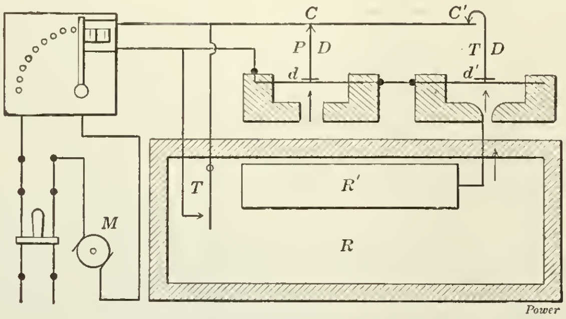


Fig. 6. Thermostatic Motor Control

In the case of a spring-opening switch an outside source of power, such as a storage battery, would have to be used in connection with the thermostat to supply sufficient power to operate the latch, or instead of the laminated-blade form of thermostat illustrated in Fig. 5 a thermostatic device *T D*, Fig. 6, actuated by the expansion force of a gas under pressure may be employed. In this case a small amount of some volatile liquid, such as anhydrous ammonia, is placed in a small closed receiver *R'* and connected to the thermostatic device *T D* by a small tube. A decrease in the temperature of the cold-storage compartment *R* causes a corresponding decrease in pressure in the receiver *R'* and beneath the diaphragm *d*, which allows it to return to its normal flat position where it effects the making of an electrical contact at *c* which stops the machine.

Still more elaborate systems have been devised to the end of eliminating all attendance. Such "completely automatic" systems divide themselves into two classes according to the two possible cycles of cause and effect on which the systems may be operated. In the former systems, illustrated diagrammatically in Fig. 7, variations in temperature in the cold-storage room *R* effect the starting and stopping of the refrigerating machine, which through the resulting variations in back pressure effects the regulation of the flow of refrigerant to the expansion coils and through the resulting variations in head pressure effects the regulation of the flow of water on the condenser coils.

In the latter systems, illustrated diagrammatically in Fig. 8, variations in temperature in the cold-storage room *R*, instead of effecting the starting and stopping of the refrigerating machine, and indirectly the regulation of the flow of the refrigerating medium, in this case directly controls the flow of the refrigerant through a thermostatic-expansion valve *V* which in turn controls the starting and stopping of the machine through the resulting pressures. The cycle of operation of each system is given in detail in the following.

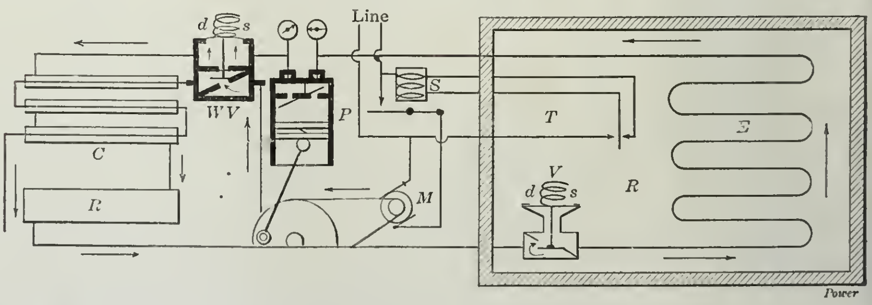


Fig. 7. Completely Automatic Refrigerating System

The thermostatically controlled system shown in Fig. 7 employs a laminated-blade or other reliable type of thermostat for making and breaking two electric circuits which, without going into full detail regarding the complete circuit, stops and starts the machine through the operation of an appropriate automatic motor-controlling panel. The electrical solenoid switch *S*, when energized, raises the motor-controller arm slowly over the rheostat segments. The deënergizing of this solenoid by short-circuiting when the thermostat makes the circuit through the stopping contact, allows the rheostat arm to drop, open circuiting and stopping the motor.

The operation of the machine is to draw the refrigerant vapor from the expansion coils *E* and discharge it into the condenser *C*, which operation reduces the pressure in the condenser coils. The reduction in pressure in the expansion or low-pressure side of the system allows a properly adjusted spring *s* or weight on the expansion valve *V* to overcome the back pressure of the gas exerted beneath the diaphragm *d*, which forces the attached valve stem down, opens the valve and allows the refrigerant to flow from the liquid receiver *R* to the expansion coils. In a similar manner the increased head or condenser pressure operates on a water-controlling valve *W V*, which instead of closing with increasing pressure, opens and admits cooling water to the condenser *C*.

The evaporation of the liquid refrigerant admitted to the expansion coils by the expansion valve produces an increase in pressure which overcomes the pressure of the spring above the diaphragm and closes the valve until the pressure is so reduced by the drawing away of the vapors by the compressor that the spring again overcomes the pressure of the gas below the diaphragm, when the valve opens and more liquid is allowed to pass. As a matter of fact there is no appreciable variation in back pressure and accordingly no intermittence in the feed of the liquid. The valve acts simply as a pressure-reducing valve maintaining that back pressure for which it is adjusted until the machine stops, when the evaporation of the residual liquid in the expansion coils produces an abnormal rise in back pressure which overcomes the tension of the spring and tightly closes the valve.

When the machine is stopped and no more hot, high-pressure gas is discharged into the condenser, the cooling water soon reduces the temperature and corresponding pressure in the high-pressure side of the system, which allows the spring in the water valve *W V* to overcome this reduced pressure and close the valve, interrupting the flow of water to the condenser.

In the second method of control, illustrated in Fig. 8, an increase in temperature causes the thermostatic fluid in the thermostatic tube *T* to expand and exert an increased pressure under the diaphragm *d* of the thermostatic expansion valve *V*, overcoming the adjusting spring *s*, opening the valve and admitting liquid refrigerant to the expansion coils. This increase in expansion-coil pressure caused by the introduction of the liquid overcomes the pressure of the spring *s* above the diaphragm *d* of the pressure-actuated, motor-controlling device *M C* making the electric circuit to the motor and starting the machine. The resulting decrease in back pressure allows the spring of the expansion valve to assert itself, forcing the diaphragm down and throttling the incoming liquid to the requirements of the compartment. When the temperature has been sufficiently reduced, the correspondingly reduced pressure in the thermostatic tube *T* allows the spring *s* to close the thermostatic expansion valve *V* and the reducing pressure in the expansion coils is now overcome by the spring in the motor-controlling device *M C,* which, operating in the opposite direction due to the decreased pressure, stops the motor.

The automatic water valve shown in Fig. 8 operates on the same principle as that described in Fig. 7. In the semi-automatic systems illustrated diagrammatically in Figs. 5 and 6, as well as in the completely automatic systems, Figs. 7 and 8, electric power is employed. Small systems may be satisfactorily operated semi-automatically at the expense of somewhat more attendance, and usually a considerable saving in cost of power is effected by the substitution of some efficient type of combustion engine. The advisability of such a substitution at the present day of an unquestionably reliable combustion engine depends almost entirely on the relative local cost of attendance, and of power developed by electricity, illuminating gas, gasolene, kerosene, fuel oil or producer gas.

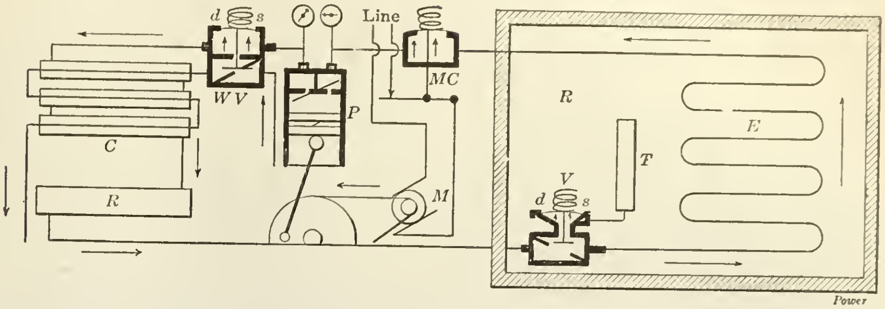
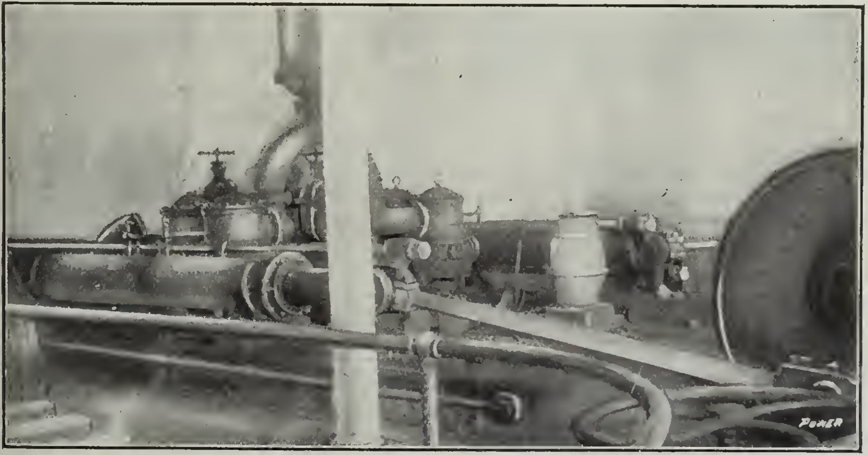


Fig. 8. System Controlled by Thermostatic Expansion Valve

Old Water Works Pump at Lynchburg

When visiting the city of Lynchburg, Va., it is an interesting diversion to view the old city pumping station located on the banks of the James river. This old plant did faithful work up to within a few years, when it gave way to a more modern system.



Old Water-works Pump

Four waterwheel-driven pumps and one steam pump comprise the present installation. One of these old timers is illustrated herewith. It was installed in 1882, or 28 years ago, by a local firm, and is driven by a 56-inch water-turbine wheel. The water is obtained from a raceway which branches off at a dam some distance above the pumping station.

This duplex pump consists of four 15½-inch cast-iron cylinders, each made with a flange on both ends. Two of these cylinders are bolted end to end on each side of the pump with a blank plate placed between each pair. This plate forms a central cylinder head for the two cylinders. The two outer ends of the cylinder are fitted with stuffing boxes, and each cylinder is fitted with a 10½-inch plunger with a 36-inch stroke. The two crank-end plungers obtain motion from a crank rod on each side, as shown. The crosshead slides in a flat guide, the sides of which are beveled inward at the top to prevent the crosshead from tipping.

The head-end plungers are driven by two side rods which extend from the crosshead on the crank-end plunger to that of the head-end plunger. The crosshead guides of the head-end plunger is the same as that on the crank end of the pump. The cylinders are each connected to a valve pot, or chamber, which contains a discharge and suction-valve seat deck, each deck containing nineteen valves. The suction pipe is 12 inches, and the discharge pipe 14 inches in diameter. An air chamber is located on the arched pipe, shown between the water cylinders.

The two crank pins are set 90 degrees apart, both crank disks being hung on the same crank shaft. Near the center of the shaft is a beveled gear which meshes with a smaller beveled gear keyed to the waterwheel shaft. The average lift of the pump is 5 feet, and the pressure pumped against is 156 pounds per square inch. The pump runs at a speed of 18 revolutions per minute.

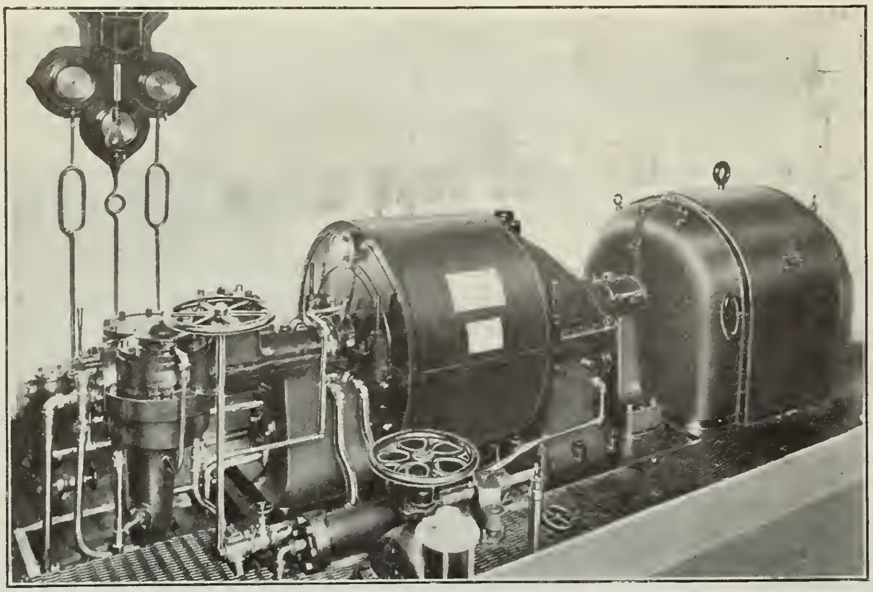
Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Use of Synchronous Electrical Machines to Connect Low Pressure Turbines and Reciprocating Engines

By Charles B. Cooke, Jr.

The low-pressure turbine has greatly facilitated the extension of existing power plants with minimum increase in floor area and complementary equipment. An interesting problem in this connection developed with the recent addition of a Westinghouse low-pressure turbine to the Bernon Mills power plant in Georgiaville, R. I. Previous to the enlargement of the power equipment, the steam plant of this company included one 23x60 twin-cylinder Corliss engine which developed approximately 420 indicated horsepower under normal operating conditions. The engine is belted directly to a line shaft and carries a characteristic mill load.



Low-pressure Turbine and Synchronous Generator

With additional power requirements, however, it was found necessary nearly to double the steam-plant output, and, moreover, it was essential that this should be accomplished without any material extension of the station limits. To fulfil these demands it was finally decided to operate a low-pressure turbine with the exhaust steam of the reciprocating engine, the turbine to drive an alternating-current generator electrically interlocked with a synchronous motor coupled to the line shaft. Excitation for both the generator and motor is furnished by a small direct-current dynamo belt driven from the synchronous motor.

While in many cases the induction motor is given preference over the synchronous type, chiefly because of its large starting torque, which makes it unnecessary to use auxiliary starting apparatus, the conditions at the Bernon Mills made this qualification unnecessary, the motor being brought to speed by the Corliss engine.

The capacity of the low-pressure unit was determined from the full-load steam consumption of the engine, 12,400 pounds per hour, and by assuming for the turbine a steam rate of 36 pounds per kilowatt-hour, an allowance of 7 per cent. being made for moisture in the engine exhaust. This method gave an available turbine-generator power of 320 kilowatts and accordingly a low-pressure turbine normally rated at 300 kilowatts was installed. This turbine is of the standard Westinghouse single-flow construction, operating between 15 pounds absolute pressure and 28 inches vacuum, and is operated in conjunction with a barometric condenser.

The unusual character of this installation necessarily involves an interesting procedure on starting up. The reciprocating engine is first started slowly, exhausting to the barometric condenser. The available natural head of 28 feet existing at the plant being insufficient for priming purposes, a slight vacuum of 5 to 6 inches is first produced by bypassing part of the condensing water to the tail pipe of the barometric tube. Through the medium of this partial vacuum, the turbine is started by closing a valve which, when open, bypasses the steam from the engine to the condenser, the turbine inlet pressure increasing to about 5 pounds gage and remaining about at that point until a vacuum of 12 to 15 inches is formed by the condensation of steam through the agency of tail-pipe injection. The valve which passes the injection water to the tail pipe is then closed, and the 15-inch vacuum elevates the condensing medium to the condenser head, readily establishing the working vacuum of 28 inches or over.

The synchronous motor coupled to the line shaft having been brought up to speed by the engine, its field and that of the generator are excited, following which the voltage of both machines is raised to 400 volts, the machines are synchronized, and finally thrown in parallel.

This procedure, while necessitating several steps, has proved very simple to the operators, the entire operation from starting the reciprocating engine to placing the station under full load rarely consuming more than five minutes.

The electrical principles involved in this method of power development render the provision of a turbine governor unnecessary, because when the generator and motor have been put in parallel, the interlocking or synchronizing force developed when either machine tries to depart from synchronism effectually prevents any variation in the speed of the turbine. If, for instance, the motor tends to increase its speed, the engine will be partly relieved of its load and the steam consumption consequently reduced. Then, with less steam passing to the turbine, its speed and output decrease, causing a similar reduction of speed in the motor, until a balanced condition is again obtained. This action follows any change in the rate of work in the main unit with but very small lag.

The electrical equilibrium of the system is further maintained by the employment of a synchronous motor of larger capacity than the turbine, this being, in effect, the electrical analogue of the flywheel on an engine.

If, for any reason, less power is required from the shaft, the engine and motor share the decrease in the demand, the former through the action of its governor and the latter by reason of its phase relation with the generator.

A New Alternator

In building large alternating-current generators, of course, the features of regulation and wave form receive full consideration along with the more familiar features of efficiency, temperature rise, etc., but purchasers of small generators of this class are usually contented if the machines run reliably and do not overheat; consequently, many (not all) small alternators are built without a great deal of consideration for the finer details of performance, such as wave form, behavior under different power factors, etc. The conditions of use, however, have steadily changed in such directions as to necessitate more attention to the points mentioned, and in response to this development many builders of small alternators are now giving relatively as much effort to the perfection of the hitherto neglected features as is bestowed upon the same features by designers of large machines.

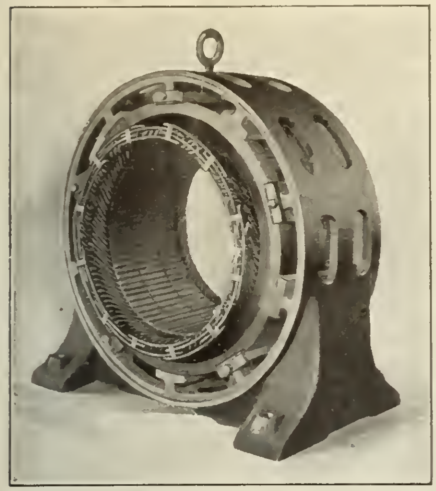


Fig. 1. The "Ideal" Armature

Amongst the small alternators designed on "large" lines is the "Ideal" machine illustrated herewith. As Fig. 1 indicates, the armature is the stationary member. The core is mounted in a one-piece housing of cast iron and is held in place by clamps at the ends, to avoid the eddy-current difficulties entailed by the use of bolts through the core. Ventilating ducts are left at intervals along the core and the "spacer" in each duct is made with fins which bear against the core teeth on each side of the duct, to prevent vibration of the teeth and resultant humming. The coils are form-wound and individually insulated, the slots being of the entirely open form which permits the direct insertion of a taped coil.

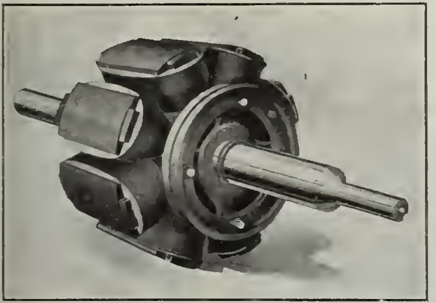


Fig. 2. Field Magnet

The field magnet is a steel casting with solid shoes bolted on the ends of the magnet cores. The object of using solid instead of laminated shoes is to facilitate parallel operation by supplying a path for the eddy currents set up by hunting and irregularities in angular velocity, without the use of squirrel-cage or other forms of dampers. The air-gap length and the slot width are proportioned with a view to restricting the pole-face losses to satisfactory values during normal operation, the aim being that excessive eddy currents shall be induced only by variations in angular velocity which are serious enough to jeopardize parallel operation; such eddy currents exert, of course, a synchronizing influence, opposing any tendency to fall or pull out of step. Fig. 2 illustrates the construction of the field magnet. The magnet coils are composed of square wire in smaller sizes and thin copper "ribbon" wound on edge in larger sizes. The "space factor," therefore, is obviously about 30 per cent. higher than could be obtained with round wire, and the coil is mechanically more sturdy. The pole shoes are shaped with a view to producing a true sine wave of electromotive force under average operating conditions—that is, at or near full load and with a power factor between 80 per cent. and unity. Of course, a true sine wave cannot be obtained at all loads and power factors, and the next best course is to get it under predominating conditions. The same argument holds as to efficiency, and instead of yielding maximum efficiency at full load, the "Ideal" alternate is designed for greatest efficiency at around 80 or 90 per cent. of full load, which, in everyday work, is carried a much longer proportion of the running time than full load, as a general rule.

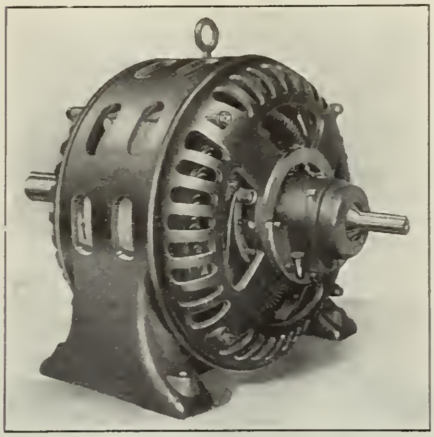


Fig. 3. Complete Machine

Fig. 3 shows the appearance of the complete machine, and close inspection will enable the reader to see that a split journal box is used and that the lower half of it is bolted to a circular seat on the end bracket of the machine. The object in using this construction is to make it easy to remove the journal box or to take out the field magnet. The pulley of a belted alternator need not be removed in order to do either of these things, and the journal box of a direct-coupled machine can be removed without taking down the shaft.

These alternators are built in sizes ranging from 20 to 200 kilovolt-amperes by the Ideal Electric and Manufacturing Company, Mansfield, Ohio.

LETTERS

Handling a Large Motor Starter

The following plan was used to prevent arcing on a heavy-duty starting box which is used for starting a large motor and is not equipped with magnetic blowout:

A piece of wood about 2 inches wide and a foot long was whittled down to a chisel edge. When ready to start the motor, this was inserted between the first and second contact blocks, and the sliding contact pulled over against it. When the motor had come up to speed on this contact the stick was put between the second and third notches and the sliding contact again moved over, repeating this process until the last two or three blocks were reached, after which it was unnecessary.

The reason for using the wooden wedge was that the sliding friction of the brush makes it difficult to move the lever of a large starter just far enough to get full bearing surface on a given contact plate and to avoid overrunning the contact and forming an arc with the next plate.

The idea originated, I believe, with C. C. Constantine, of the Baltimore Copper Works.

Donald M. Liddell.

Grasselli, Ind.

Wanted: An Automatic Water Rheostat for a Pump Motor

A triplex pump, 9x12 inches, is to be placed underground and driven by a 220-volt shunt-wound motor of 50 horsepower, running at 970 revolutions per minute. This pump will draw water from a sump supplied by sinking pumps and from other sources, and the flow will be variable, up to its maximum capacity. It is desired to vary the speed of the motor automatically as the water in the sump rises and falls, the motor speed and pump work increasing with the rise in water level and decreasing with the fall. It is desired that the pump continue running, even at lowest water level.

There appear to be many self-starters on the market; but it is desired to construct a water starter which, by closing the gap as water level rises and opening it as water falls, will vary the motor-circuit resistance and alter the motor speed. The usual form of starting box will be used, the water resistance being connected between the starting box and the motor.

The principal difficulty seems to be to proportion the resistance of the water rheostat so that it will not be too large at low level and stop the motor. I should be glad to see diagrams of arrangements suggested by other readers. The water rheostat should short-circuit itself at the high point. The mine water contains so much acid that it would probably not be necessary to add any chemical, and there could be a steady flow through the rheostat.

W. H. Earle.

Ocampo, Chihuahua, Mexico.

Charging Storage Batteries in Parallel

On page 1426, of the issue of August 9, Mr. Meade gives some diagrams of switchboard wiring for series or parallel charging of storage batteries, which are open to criticism. As a rule, small isolated plants are 110-volt installations and in all probability a battery in such a plant would have to be charged in parallel. Mr. Meade's diagrams show a main-line rheostat only. Theoretically, this arrangement would work, but practically it would seldom if ever give any degree of satisfaction if equal charging of the two sections is to be considered. Anyone who has ever tried to charge a battery in parallel sections without a separate rheostat in each section has realized that it is impossible to balance the charging current. A person who installed a battery according to the diagrams of Mr. Meade's article, if unfamiliar with that class of work, would have trouble on his hands.

The addition of the balancing rheostats would not complicate the wiring and certainly would aid in the successful operation of the battery. The increased life of the battery due to regular charging would offset the additional cost of the balancing rheostat many times over.

It would be preferable to leave out the main-line rheostat and install two balancing rheostats rather than use a main-line rheostat only, the latter being decidedly bad practice to advocate for a piece of apparatus, the cost of which would be a large part of the total cost of a small installation, and of comparatively short life, even with the best of care.

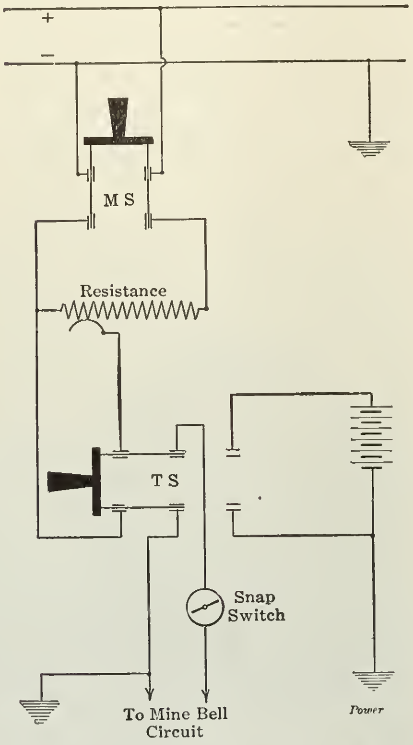
A. C. Kerr.

Ft. Monroe, Va.

Operating Bells in Mines

Signal bells around a mine are as necessary as water in the boilers, and the quantity of dry batteries ordinarily used up makes a considerable expense item in the course of a year. As all our signal bells are of large size the current consumption from batteries was necessarily large, and a battery of six and eight cells would not last very long.

To avoid using batteries except when the power plant was shut down (which is very seldom) the circuit arrangement shown by the accompanying diagram was installed. It has not given one particle of trouble, and the cells will last until they lose their strength through age. It is advisable to ground one side of the bell circuit because the negative side of the power circuit is grounded for trolley return in the mine. There is a sliding contact on the resistance by means of which the voltage on the bell lines can be raised when the lines become leaky or poor connections are made. The snap switch is included in the circuit to cut out the bells in case of a short-circuit on the lines.



Supply Connections for Mine Bells

All circuits for bells are taken from the double-pole double-throw transfer switch, *T S*. The main switch *M S* is provided to cut the whole equipment off from the power circuit if it should become necessary, as in case of trouble in the resistance.

L. Earle Brown.

Roundup, Mont.

Unusual Cause of a Dead Dynamo

Some time ago a small shunt-wound direct-current generator used for excitation purposes gave trouble. While in service the machine let down, failing to generate any voltage whatever. Since trouble of this kind had been encountered before, we tried the usual means of correction. Previously the trouble was due to an insulating coating on the commutator, of particles from a number of buffing wheels located nearby. After sandpapering both the commutator and the brushes, the machine built up again, but generated for only a short time, when it again refused to work. A superficial examination failed to throw any light on the trouble, and we settled down to test. After testing all the lines and finding no trouble, we tested the field winding and found that it seemed to be all right. We were then at a loss, because the machine acted exactly as an open field circuit would make it do. Having another machine running, I connected the field circuit of the "dead" machine to the busbars of the operating machine, through the rheostat of the first machine. The machine built up nicely, but upon careful examination we noticed a small arc at the bottom of one of the field-magnet coils of the affected machine.

Upon dismantling the machine and examining the coil we found that in winding the coil a layer of insulation was put around it before the final layer of wire was wound on, and along a heavy strip of insulation was laid a strip of german silver about 1 inch wide and 1/32 inch thick. The top of this had been carefully insulated to within ¾ of an inch of the lower end. The last layer of wire was then wound on beginning at the top of the strip and running to the lower end. As the final convolutions came to the bare surface of the german-silver strip, the insulation was scraped off and the last six turns were soldered, as the assembler supposed, to the german-silver strip, but the solder failed to "run" and only the surfaces of the wires had been joined with solder. The only connection to the strip was the contact due to the tension of the wire wound on the bare end of the strip. The strip was used to carry the current to the other end of the coil, for convenience in connecting. Expansion and contraction of the strip caused the machine to fail at one time and work at another. When cool, the contraction would make a good joint, thus enabling the operator to start the machine. This was also the reason why all of the tests failed to locate the trouble.

When the excitation was effected from the duplicate machine, the poor contact due to expansion caused the arcing at the defective point.

C. A. Davies.

Cincinnati, O.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Generation of Power from Peat*

By T. Arthur Mighill

There is as yet, to my knowledge, no plant in operation where peat fuel is handled entirely by machinery from start to finish. I believe that digging the peat, moving pallets, or gathering dried peat by hand is impractical if peat fuel is to be prepared cheaply enough to compete with coal. Although some processes have eliminated most of the hand labor, there is much yet to be done. We hear of successful peat-power plants in Europe; however, in America we are not able to operate under European conditions. We must pay male laborers higher wages than prevail in Europe, and we do not employ women and children for such work.

As peat is a fuel of low calorific value, averaging from 6000 to 9000 B.t.u. per pound of dry substance, one pound of good coal is equal to one and one-half to two pounds of average peat. Although peat when burned under a boiler will generate steam, the quantity required is nearly twice that of coal, so that a larger grate area is required and the fuel must be more frequently fired. The ashes are light and can easily be removed. The flame is long and should put the heat where it will do the most good, in the tubes. From its light and pulverent nature, when the draft is forced, considerable fuel might find its way into the stack before complete combustion.

For small plants it may not be practical or desirable to use a gas producer and engine on account of the simplicity of operation of a steam boiler and engine. I am advocating the use of the gas producer and engine in large plants not on account of any difficulty in using peat as a steam-boiler fuel but on account of the higher heat efficiency of the gas engine.

In contemplating the use of peat as a source of power we must consider the economic conditions connected with the locality. We must consider the price of coal at the locality and the cost of preparing peat fuel. The farther the bog is from the coalfields the greater the possibilities of the peat proposition. The matter ends here if the power is to be used for factory purposes, but if electrical energy is to be sold, consideration must be given to the situation of the bog relative to the community to be served. As electricity can easily be transmitted over a considerable distance, the bog may be some distance from the region served, but the generating plant must be at the bog. Freight charges must be eliminated.

The preparation of peat for use as fuel consists of digging the peat, drying it and storing it. The gasification of it is the most important process of the series. I have made considerable inquiry as to whether peat will clinker badly in a producer but have received no satisfactory answer. In a series of experiments at Pittsburg I found some clinkers, but I cannot say that they were any worse than those formed by coal. From the fineness of the ash I judge that the possibilities of forming a large amount of clinker are not very great. However, if the peat is too finely divided the draft will form holes and here the combustion will be most intense, favoring clinker formation. The lightness and porosity of peat is a drawback, for the combustion may be so rapid in places that holes or "chimneys" are formed. This necessitates considerable poking of the fire; and every poke of the iron is likely to make a path for the air. The lightness of the fuel and the low heat value greatly increase the size of the peat producer over a coal-gas producer of the same power capacity.

Proportion of Moisture

The content of water in air-dried peat makes the operation of a producer uncertain, for to my mind there is quite a difference between running steam through incandescent coke, as in an anthracite producer, and using the steam liberated in the fuel bed itself at a rate which, from the nature of the operation, must be irregular. Under such conditions the producer may run sometimes too hot and sometimes too cool and the heat value of the gas may vary greatly.

This raises the question, how much water can peat contain and be used successfully in a producer? This can be determined only by long tests of fuels containing various quantities of water, not by such short runs of which accounts are available. It has been said that peat containing as much as 50 per cent. of moisture can be used in a producer. If this were true it would cut down the drying expense very much, but I believe that 30 to 35 per cent. of moisture is about the limit that can be allowed.

The evaporation of water in a producer only to have it condense again unaltered is inexcusable unless we are going to recover ammonia, as in the Mond process, or else find some compensation in the preparation of the fuel whereby the producer is worked extra to save the drying field. It has been shown that to attempt to recover byproducts, except in plants of about 3000 horsepower and over, is impracticable.

If the temperature of the producer gets too low, as through the evaporation of too much water, the carbonic acid is not decomposed and the heat value of the gas is correspondingly lowered.

Preparation of Fuel

The form in which the fuel is applied is of great importance. There are records of runs in this country where machine peat and briquetted peat were used. There should be no trouble in using briquets but I believe that the expense of manufacturing them does not warrant their use, and I consider properly machined peat just as good. I am not a believer, however, in the idea that a producer can be run on anything that is combustible.

In a run of about two weeks, at Pittsburg, it was found that peat dug out of the bog and not machined but thrown on the ground to dry did not make a fuel suitable for producer purposes. We did not dig the peat out in blocks as is sometimes done in Europe, but dumped a car load of fresh peat on the ground and spread it out to a thickness of several inches. The dried peat was shipped by rail from Massachusetts to Pittsburg. In all, it was shoveled over about ten times before it was burned in the producer. The result was that there was so large a percentage of "fines" in it that the draft through the producer was greatly restricted, and the suction rose so high that the operation of the producer became a source of danger. As a result, we got poor gas, high in carbonic acid.

To operate a producer successfully on peat the fuel must be prepared by machining it and drying it so that it hardens into lumps and does not produce much "fines" when handled.

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*Abstract of a paper read before the American Peat Society, at Ottawa.

RESOURCES REQUIRED FOR THE CONTINUOUS DEVELOPMENT OF 1000 HORSEPOWER

Reports of producer tests show a consumption of from 1½ to 2½ pounds of dry peat per horsepower-hour. Suppose the average is two pounds, containing, say, 9000 B.t.u. per pound, which would represent an excellent quality of peat. As it is impractical to use perfectly dry peat, more pounds of less heat value must be used. Assuming that the peat contains 33 per cent. of moisture, which seems to be the most practical state of dryness to attain, the two pounds of dry peat would be increased by one pound of water, making three pounds of wet peat necessary to develop one horsepower-hour. This would require excavating from the bog 240 tons of raw peat, with a content of 90 per cent. of moisture, each 24 hours to keep up a supply of 1000 horsepower. Bog peat has about the same weight as water (62½ pounds per cubic foot). I am assuming a bog undrained or imperfectly so, as the density of peat increases with the dryness of the bog.

An acre of such a bog would furnish 1360 tons of raw peat for each foot of depth, which, with 90 per cent. moisture, would yield 136 tons of dry peat, equivalent to 204 tons of peat containing 33 per cent. of moisture. Since a 1000-horsepower plant would consume 36 tons of fuel per day, one foot-acre of peat would last about six days, and a square mile one foot deep would last about ten years. These figures are based on an average, not a maximum, output of 1000 horsepower throughout the 24-hour day.

DRYING THE PEAT

Climatic conditions have so great an influence upon the drying of peat that it is impossible to say how long peat must be laid out in order to dry sufficiently. I think that the season in the northern United States and Canada can hardly exceed six months and will in most places probably be less, unless the locality is one with slight rainfall. If the drying period is six months, then during that time the fuel for the whole year must be prepared. In the case of the plant of 1000 horsepower average output there would have to be prepared each day 72 tons of fuel, with 33 per cent. of moisture.

Most important is the time necessary to dry the peat. This varies with the extremes and middle of the season, set backs by rain, heavy dews and possible frosts; on the average, peat properly laid out may reach the necessary degree of dryness in from two to three weeks if it be not spread too thickly. As my object is not so much to show what has been done in specific localities, but to show the conditions attending a peat-power proposition, I will assume that the drying field, whether bog or upland, will be covered to an average depth of 4 inches and that the peat will be lifted every three weeks throughout the season of six months.

One acre of drying ground carried to a depth of 4 inches would contain 472 tons of fresh peat having 90 per cent. of moisture, or 71 tons of air-dried peat of 33 per cent. moisture. The acre of drying ground would be covered 9 times in the six months and the fuel produced per acre per season, that is, for the year, would be slightly over 600 tons.

The 1000-horsepower plant requires 36 tons of fuel per day or 13,140 tons per year. It would, therefore, require 22 acres of drying ground or not quite 1/30 of a square mile. I think that in most places this area would hardly be sufficient. Should the peat be laid out in cakes the space would be somewhat increased by the vacant spaces between the cakes, unless the cakes be made correspondingly larger. Peat in this form would dry quicker than in sheets and the efficiency of the ground would probably be much increased.

Experience has shown that the use of drying shelves is out of the question in handling such quantities of peat on account of the cost and deterioration of

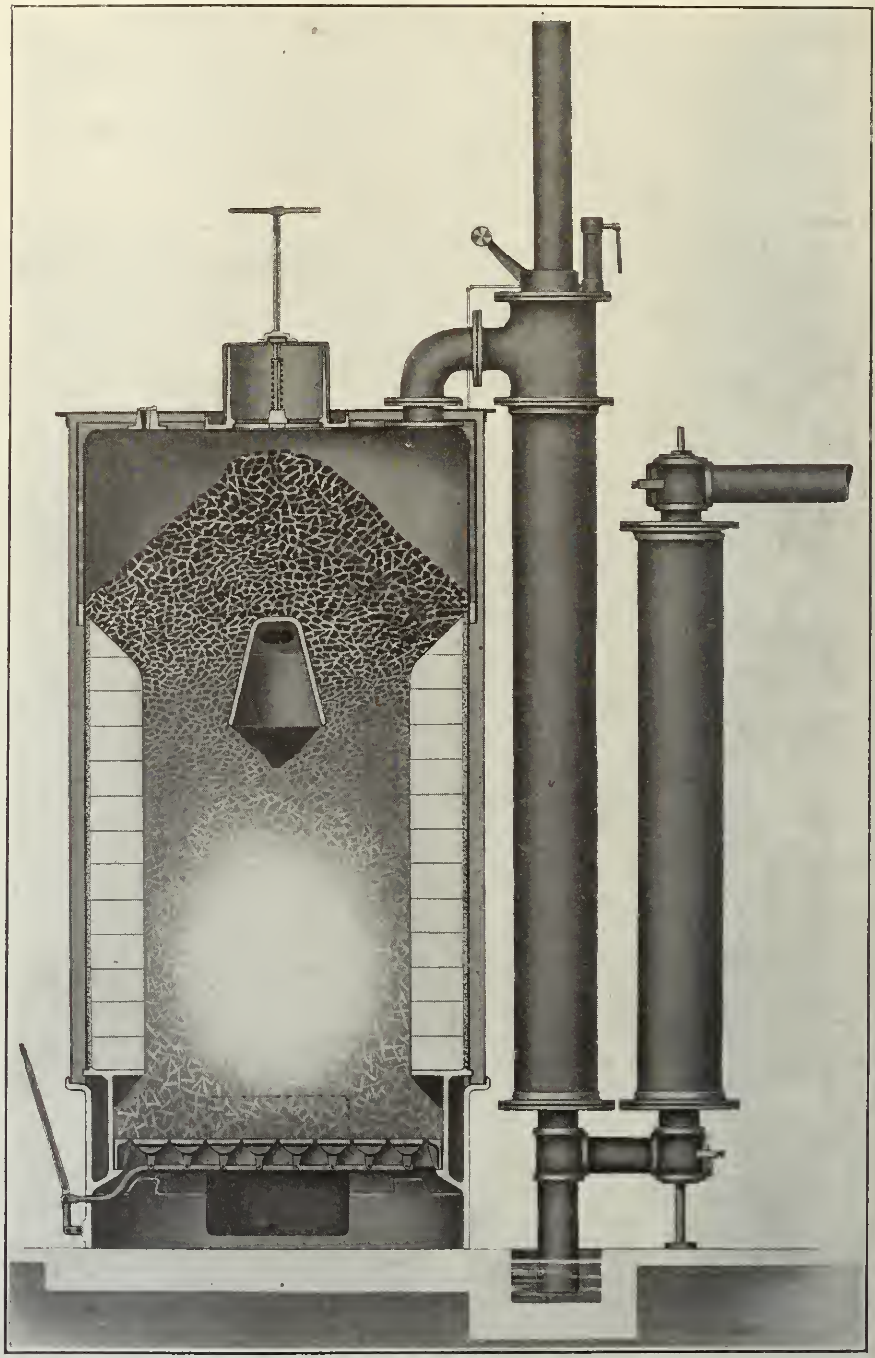


FIG. 1. THE SYRACUSE SUCTION GAS PRODUCER
(See page 1607)

the lumber. Further, it is not an easy matter to fill the shelves and unload them mechanically.

Two other ways of removing the water from peat are often suggested, pressing the water out and evaporating it by artificial heat. I would not say that peat presses are impracticable, but I do not think that the inherent difficulties have been overcome yet. It requires fine screens to hold the peat and great pressure to force the water out. Filter presses or machines of this type work too slowly to handle peat in large quantities. The requirements of a peat press are continuous and rapid operation, great strength, and screens that will not clog up and yet will hold the peat from passing through.

To remove a small percentage of the water would not be worth while using a press. However, if half of it could be removed by a press, it would be profitable to do this and subsequently air-dry the peat. The waste heat from the producer and the exhaust gases from the engines might be utilized in some way to finish the drying of the peat. The exhaust gases might be passed through the drying house, but the waste heat from the producers could not easily be utilized because it would require an enormous amount of piping, as the hot gas at this stage must be confined.

The peat should be removed from the drying field by mechanical gatherers. Strong and durable machinery is required. The gatherer must be designed to lift rather large pieces of peat and not clog or break. It must move on rails and be propelled by power. When loaded, it must carry the peat to the storage bins, which must be constructed so as to allow a circulation of air and at the same time protect the peat from rain.

If peat be dried to 50 per cent. of moisture or under it is uninjured by frost alone and can stand winter cold. It has been shown that peat should not dry too long in the hot sun because it becomes brittle. A better fuel can be obtained by partially drying on the field and finishing under cover with a good circulation of air.

As half a year's fuel must be stored for the winter it is evident that the storage plant must be quite large on account of the amount stored and its bulky nature. For the plant delivering an average of 1000 horsepower continuously, about 6500 tons must be stored, and as a cubic foot of peat fuel weighs about 25 pounds, the volume occupied by 6500 tons will be about 520,000 cubic feet.

Storage bins can be constructed so that even during the winter the peat will continue to dry.

By using machinery, peat fuel ought to be manufactured for $1 per ton or less. If so, it can compete with coal at $2 a ton.

The Syracuse Producer

The Syracuse gas producer, illustrated herewith, differs from the usual forms of updraft apparatus in several details, chief among which may be considered a hooded gas outlet below the top of the fuel bed in the generator and the use of a scrubber of a simple baffle type instead of the familiar coke- or lattice-filled tower.

The hooded gas outlet in the generator (see Fig. 1) does not carry the gases out of the generator but delivers them to the space in the upper end of the shell, above the fuel bed. Two pipes extend from the hood to points near the top head of the generator, the pipes being inclined at an angle of about 45 degrees. The opening in the hood which leads to one of these pipes is visible in the sectional view of the generator, Fig. 1.

The generator is built with a vaporizer in the upper end. This is not the usual deep "pan," but consists of a very shallow space between the two top walls and a very narrow annular space or "leg" extending down the side walls to a point about level with the gas-outlet hood. The body of water is, therefore, relatively small and thin; consequently the rate of evaporation is very flexible and can be changed suddenly to conform to sudden load changes.

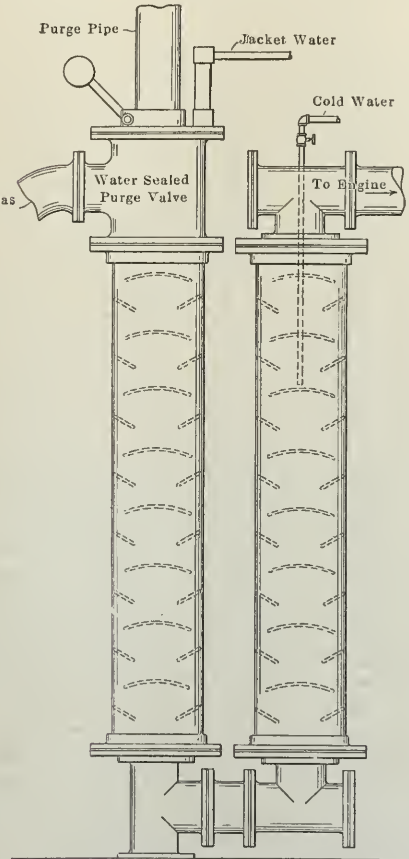


Fig. 2. Syracuse Scrubber

Besides the space between the generator walls which serves as a vaporizer, there is also a space between the inner and outer walls which serves as an air preheater. All the air passing to the ashpit goes through this space, which extends throughout the full hight of the generator shell. The air passes over the surface of the water in the vaporizer before entering the ashpit, and therefore carries steam along with it.

The annular space between the walls of the generator opens into a wider annular chamber in the cast-iron base which surrounds the grate and the bottom part of the fuel bed. This annular chamber constitutes a preliminary vaporizer, its function being to supply steam when the generator is being started up, before it has become hot enough at the top to evaporate the water there. The water supplied to the regular vaporizer in the top, not being evaporated, overflows the partition between the vaporizer "leg" and the air preheater, and runs down to the annular chamber around the grate, where it is evaporated by the heat from the freshly made fire. When the generator has got to making gas regularly, the ashes accumulated on the grate protect the wall of the "preliminary vaporizer" from the fire and prevent it from being damaged after all of the overflow water has been evaporated out of it. An automatic regulator then controls the supply of water to the top vaporizer and prevents any further overflow into the air-heating space between the shells.

The construction of the scrubber is indicated in Fig. 2. It consists of two large pipes connected at the bottom and containing a series of baffle plates over which the cooling water flows. The water discharged from the engine jackets is used in the first leg of the scrubber and cold water is used in the second leg. The idea underlying this method of scrubbing is that the warm water will remove most of the dirt and other solid impurities, as well as some of the sensible heat, leaving comparatively little work to be done by the cold water in the second leg of the scrubber.

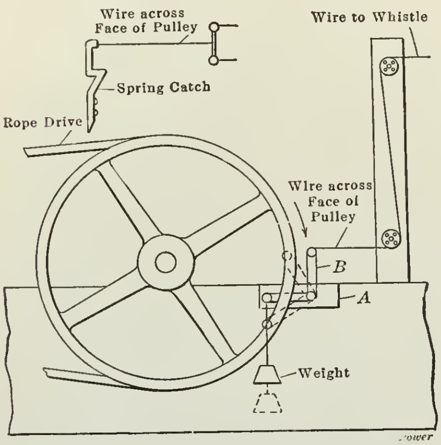
The grate in the generator is of the rocking type and is shaken by means of the hand lever shown at the left in Fig. 1. In order to prevent the formation of air passages through the fuel bed alongside the walls, which is apt to occur when the ashes are shaken off the edges of a grate, a stationary ledge is provided all the way around the wall at the grate level; this maintains a solid body of ash next to the wall when the grate is shaken. The generator is intended to gasify only anthracite coals, and is built by the Syracuse Industrial Gas Company, of Vernon, N. Y.

Readers with Something to Say

Safety Device for Rope Drive

The accompanying sketch represents an ingenious and handy safety device for rope transmission. It was used with much success at a plant in which I worked a few years ago. The plant was of 3000 horsepower and all the heavy machinery was run by rope drive. Before this device was installed considerable trouble and expense were caused by ropes breaking, but after its installation we were able to avert all serious trouble in this line.

The device is made in the following manner: A piece of iron plate *A*, about 5x8¼ inches, is drilled and countersunk at each corner, and an additional ¼-inch hole is drilled at the center. A second plate *B*, about ⅜x1x16 inches, is bent into a right angle, and a hole is drilled at each end and at the vertex of the angle. Then both plates are loosely riveted together at the center holes and plate *A* is screwed to the engine frame or foundation. Thus the plate *B* is free to swing about the rivet as a pivot. Next, a spring catch, shown at *C*, is riveted to plate *A* in such a manner as to engage and hold the right-angled piece *B*. A wire is stretched across the face of the pulley from the spring catch to a pine stick held by screw eyes. To the vertical arm of *B* is attached a wire leading over pulleys to a steam whistle, and attached to the other arm is a weight.



Safety Alarm in Position

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Should the driving rope begin to break, the first loose strand will strike the wire, disengaging the spring latch from the lever *B* and allowing the weight to pull the latter into the position shown by the dotted lines. This causes a pull on the wire connecting the whistle and the latter blows, giving the alarm. Should the strand of rope hit the wire very hard, the pine stick will break and save the spring latch from damage. After the alarm has sounded, the engine can be shut down before any serious damage results.

Edward Moran.

Columbus, O.

Taking Indicator Diagrams

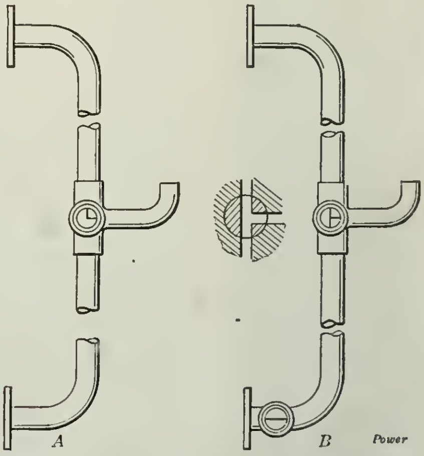
When taking indicator diagrams, two great hindrances to obtaining satisfactory results, assuming that the indicators are in good condition, are throttling of the steam and condensation in the indicator pipes. When an indicator is attached by a short length of pipe to the ends of each cylinder, these losses are reduced to a minimum; but two indicators per cylinder are somewhat of a luxury, not only because of the cost of an extra indicator, but if the engine be of long stroke, it means employing two men per cylinder in order to obtain a satisfactory set of diagrams.

As a compromise, it has become customary, especially in vertical engines, to have the indicator pipes arranged so that by turning a cock a single indicator may be connected with either end of the cylinder. The operator has a much better chance of obtaining satisfactory diagrams if he may stand still and merely turn a cock than when he has to move hurriedly from one instrument to another. The usual arrangement of the pipes is as shown at *A* in the accompanying sketch. The change-over cock is of the two-way type and usually has the two holes in the plug 90 degrees apart, although sometimes they are spaced at 120 degrees.

Though the pipes may be blown with steam before fitting the indicator, a certain amount of condensation takes place before everything is ready for taking the card; also during the intervals between taking successive cards, condensation will take place and the only outlet for the water is past the piston of the indicator; hence a large part of it remains in the pipes and instrument. The water in the pipes exerts a throttling effect upon the steam while the water in the indicator itself has an effect equivalent to that of adding a weight to the piston.

It is sometimes urged that a small amount of water will do no harm and I have heard it said by those who should have known better that since there is always a greater amount of steam than water in the leakage past the indicator piston, there should not be any fear of waterlogging the instrument. These men had evidently forgotten that much of the vapor which they saw coming from the relief holes of the indicator had probably leaked past the piston as water and had then been reëvaporated, owing to the drop in pressure.

I once had charge of a large triple-expansion engine which, for the maker's trial, had been fitted with an indicator at each end of each cylinder. The trial being satisfactory and the engine accepted, the indicator pipes were fitted as shown at *A*. Later, when another set of diagrams were taken, the engine developed apparently 5 per cent. less power under conditions practically identical with those during the maker's trial. The pipes were overhauled and found quite clear. They were replaced and carefully covered, but there was no perceptible improvement. Then an alteration was made in the pipes which proved very satisfactory. The two-way cock for the indicator was converted into a three-way cock by continuing one of the holes through the plug; a second straightway cock was fitted close to the lower end of the cylinder, for the first cock, as altered, had no point for shutting off. This lower cock could be left open while diagrams were taken, as all indicators have a three-way cock of their own, in order to obtain the atmospheric line. When the cock is in the position shown at *B*, in which it should never be left for more than an instant at a



Usual and Improved Methods of Piping for Indicator time, a quarter turn either way will connect it to either the top or the bottom of the cylinder. While changing from one end of the cylinder to the other, the cock is brought through this end position, and the momentary rush of steam will clear the pipes of any water that may be in them, also the greater part of the water from the instrument. This scheme is most useful in the case of the low-pressure cylinders of condensing engines, where the steam carries a large proportion of moisture.

J. A. Emerson.

Newcastle-on-Tyne, Eng.

Shrinking on a Pump Gear

The article by H. S. Brown on the proper fitting of engine cranks, in Power for June 14, brings to mind a bad accident, resulting in a large loss of time and money, which was caused by the attempt of an engineer unfamiliar with machine-shop methods to shrink a large gear on the crank disk of a hydraulic triplex pump. The pump was driven by a 225-horsepower induction motor and the gear was to be shrunk on over the crank disk, as shown in Fig. 1. The original gear had had six spokes, but owing to the frequent cracking of the arms the new gear was made webbed.

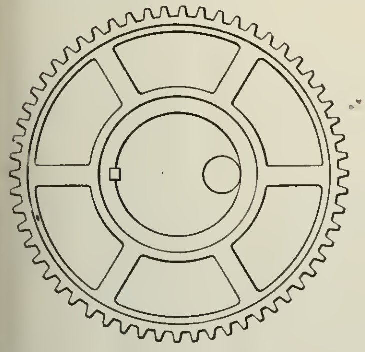


Fig. 1

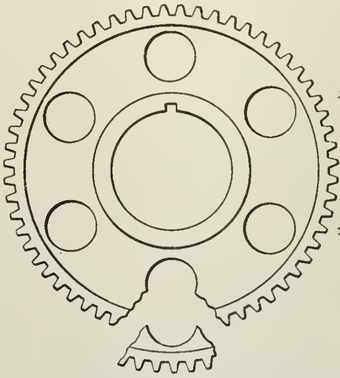


Fig. 2

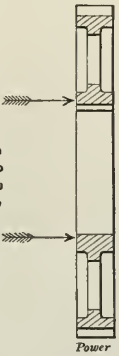
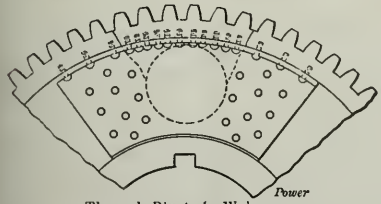


Fig. 3

Shrinking on a Pump Gear

The gear was finished complete in the shop and the proper shrinkage allowance was made by boring the gear 32 thousandths smaller than the diameter of the crank disk, which was 30 inches, and delivered to the engine room. As the shop was very busy the engineer's request that he be allowed to shrink the gear on was granted. The first move he made was to suspend the gear from a crane near the pump and set it at the proper hight to shove onto the disk. A turnbuckle was used for close adjustment. The engineer was then left to finish the job, and about an hour afterward a loud report startled those in the vicinity of the engine room. Upon investigation a large piece of the rim was found cracked out, as shown in Fig. 2, and lying on the floor. The flame from three crude-oil torches had been concentrated on the heavy metal in the hub as indicated by the arrows in Fig. 3, instead of applying the heat to the whole gear uniformly, and the expansion of the hub had been resisted by the rim until its ultimate strength was exceeded and the gear cracked along the line of least resistance.



Through Rivets in Web.
Tap Rivets in Rim as shown

Fig. 4. The Patch

As there was no time to make a new gear, and also because the gear was a very expensive one, being 7 feet in diameter, it was decided to patch it. This was greatly faciliated by the webbed construction, and patches like those shown in Fig. 4 were applied and riveted into place. The gear has run satisfactorily for several years. The bedplate of this same pump cracked around several of the foundation bolts as represented in Fig. 5 due to the heavy upward thrust of the driving pinion and owing to the difficulty of gaining access to the foundation bolts a repair was very difficult to make. The method indicated in Fig. 6 was finally hit upon and the repair made without the necessity of longer foundation bolts, which would have been necessary had a plate or angle-iron patch been used, because anything placed under the nut would have left only a few threads of the bolt to be engaged.

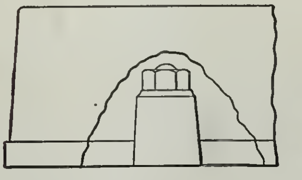
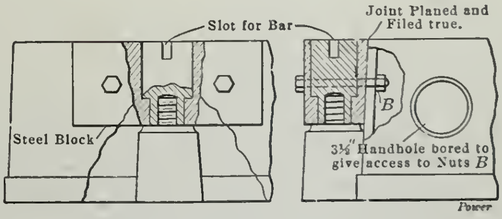


Fig. 5


Slot for Bar
Joint Planed and Filed true.
Steel Block
B
3½" Handhole bored to give access to Nuts B
Power
Fig. 6

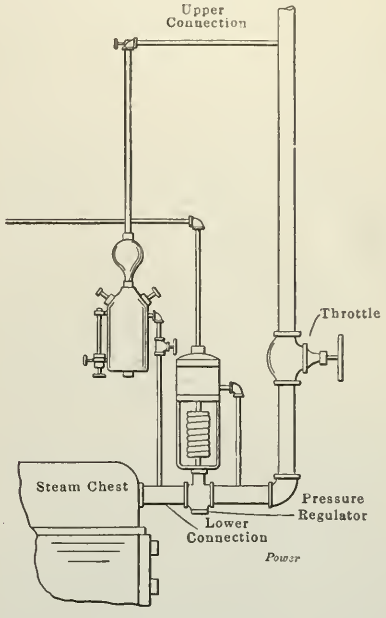
H. F. Rudolph.

San Francisco, Cal.

Defects in a New Plant

In a power plant with which I have recently been brought into contact, some rather surprising things are observed, especially as the equipment is new and apparently no attempt had been made toward the curtailment of expense in first cost.

The main steam line was found to leak at nearly every joint; occasioned, no doubt, by improper provision for expansion and contraction, or by inferior gaskets. This trouble was serious from the fact that the 10-inch line was near the ceiling and was surrounded with other pipe lines, all nicely covered and painted, but making the joints of the steam line almost inaccessible.



Wrong Way of Connecting Lubricator

Another result of poor judgment in designing the plant is the use of a motor-driven centrifugal pump for pumping out a drip well located under the floor. The boiler blowoff tank drains into this well, also all the drips from the engines and pumps; consequently the water is hot all the time. The head is about 10 feet and the outfit serves only as an ornament and an unprofitable investment. Fortunately, all electrical apparatus is duplicated with steam-driven machinery and, by using a cold-water primer, the well is pumped out by an ordinary duplex pump.

Further evidence of poor design lies in the fact that two large three-cylinder pumps, connected by link belts to 30-

horsepower motors, are provided to run the elevators. These pumps cannot be operated with the present electrical equipment, as the starting and stopping of the elevators causes an excessive variation in the voltage, consequently the steam pumps have to be used to run the elevators and the motor-driven pumps are merely to look at.

Another botch is the manner in which the lubricators are attached to the engines and pumps. They have double connections; the upper is attached to the steam pipe in the usual way, but the bottom one feeds directly into the steam chest or the line below the throttle. The lubricator cannot be operated until after the engine has been started, as the unbalanced pressure would empty the lubricator within a few minutes. This is particularly annoying in connection with the elevator pump which is controlled by pressure regulation.

The accompanying sketch shows that if the feed were properly adjusted to supply the pump while running, the lubricator would be emptied as soon as the pressure regulator cut off the steam supply.

Although this plant is new and supposedly modern in every particular, all coal has to be handled twice, and the ashes wheeled out in the old-time way.

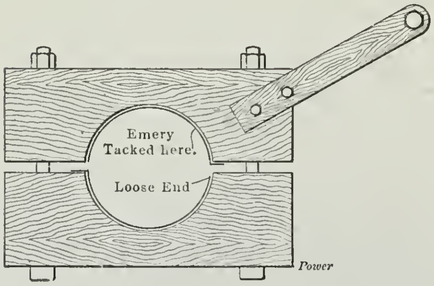
Edward T. Binns.

Philadelphia, Penn.

Truing a Crank Pin

An arrangement for turning a new surface on crank pins, as shown in the sketch, is one which I have found very valuable and handy on work that would have been impracticable and expensive to have taken to the shops.

Two blocks of wood of the required size are bolted together with liners between them and a hole is bored through the center of the joint, a little larger than the pin, which, of course, has previously been filed as nearly true as possible. The liners are then removed and around the inside of the hole coarse emery cloth is placed; this is fastened at one end by tacks in the open joint, and at the other end it is free. A strip of wood is fastened to the side of the block to form a crank for turning by hand. The apparatus is then clamped around the pin and revolved, using oil with the emery and changing to finer grades of emery cloth until the desired surface is obtained.



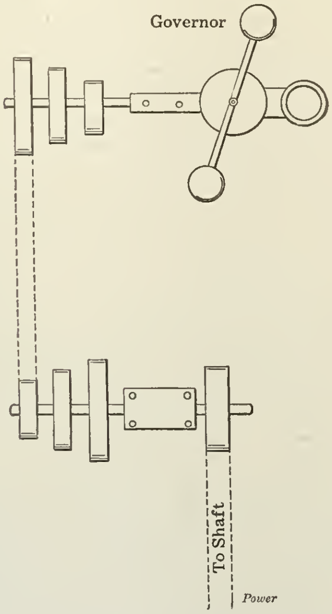
Block Used for Truing Crank Pin

F. C. Holly.

Yazoo City, Miss.

Speed Variation with Throttling Governor

The accompanying sketch shows a simple method of obtaining a variable speed with a throttling governor. It is accomplished by the use of a set of cone pulleys which permit of 50 per cent. variation in speed, yet leave the engine under perfect control.



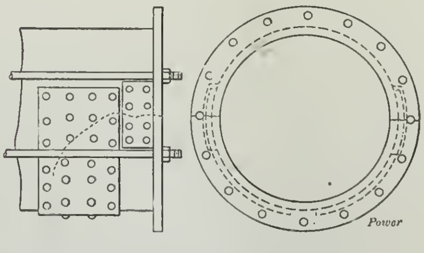
Pulleys for Varying Speed of Governor

C. R. McGahey.

Sheffield, Ala.

Patching a Broken Cylinder

The accompanying sketch shows the method of patching the cylinder of a Corliss engine which had been cracked by the breaking of a follower plate. The cracks extended from the center on each side at the head end down to a point 18 inches from the head end, almost meeting at the bottom. Upon starting the engine the cracks opened up about 1/16 inch.



Patched Cylinder

A strong clamping device, consisting of rods and bars was made and with the aid of a sledge hammer the crack was closed up until it was hardly visible. Then a 1-inch soft-steel plate was riveted on each side next to the flange and on the body of the cylinder a ¾-inch plate was riveted to each side. All the holes were countersunk and the rivets were driven cold with the heads left slightly high in the cylinder; these were afterward dressed off and polished. After all the rivets had been driven the clamping device was taken off and the crack remained closed. A groove 5/16 inch square was cut over the crack and calked full of soft copper. To further strengthen the cylinder, two 1⅛-inch rods were placed on each side, extending from head to head.

The cylinder has now been in service for some time, operating under a steam pressure of 150 pounds per square inch and is perfectly tight.

Frederick L. Ray.

Louisville, Ky.

Back Pressure

There seems to be much confusion regarding the term "back pressure" as applied to a heating system and to a refrigerating system.

In a steam-heating system the back pressure is obtained by introducing a back-pressure valve into the exhaust line of the engine; this allows all steam, in excess of the amount desired for heating, to escape. It has the effect of retarding the engine; that is, it is ahead of the engine and works against the piston.

Back pressure in a refrigerating system is the pressure of the ammonia, or other refrigerating agent, behind the compressor. This assists the compressor in doing its work. It is analogous to an ordinary water pump with the suction end connected to two tanks, one above the pump and the other below it. If the pump is started with the valve on the pipe leading to the lower tank open and the other tank cut out and the number of strokes counted; then if the valve on the line from the upper tank is opened and the lower tank cut out, the pump will be found to increase in speed. That which the water end is to a pump, the ammonia cylinder is to a refrigerating machine; which means that the back pressure on a refrigerating plant tends to assist the engine while the back pressure on a steam-heating plant retards the engine.

R. P. Wilcox.

Chicago, Ill.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Information on an Air Lift

Referring to William Beaton's inquiry on "Air Lifts" in the June 21 issue, in my opinion the arrangement which he shows would not be entirely satisfactory, as it would be difficult to proportion the size of the air jets so that too much air would not escape through the upper opening, where there is the least resistance.

There should be a valve in each connection to properly control the flow of air. While this system is occasionally used, it is of no real benefit in overcoming deficient submergence as it admits the extra air where it does the least good. The air bubbles expand as they ascend the discharge pipe, which increases their lifting power, as they approach the top; and too much air in the discharge pipe, especially near the top, breaks up the water into spray and reduces the efficiency of the lift.

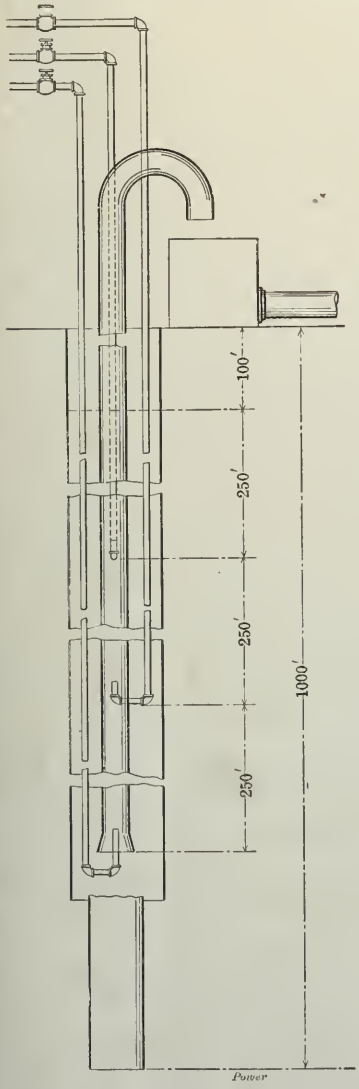


Fig. 1. Air Lift Used in Oilfields

Fig. 1 shows the proper arrangement for this style of air lift as used in the oilfields, and arranged for a 1000-foot well. It is not used to overcome deficient submergence, but to make it possible to start a deep well, in which the hight of the liquid varies greatly between the pumping and standing levels, with a compressor capable of furnishing air at ordinary pressures only. The amount of air pressure required depends entirely upon the submergence. A pressure of 110 pounds will start a well with 250 feet submergence, but it will take 330 pounds to start a well with 750 feet submergence. It is seldom that more than one air jet is used at one time, the upper ones being closed as the level of the liquid in the well falls.

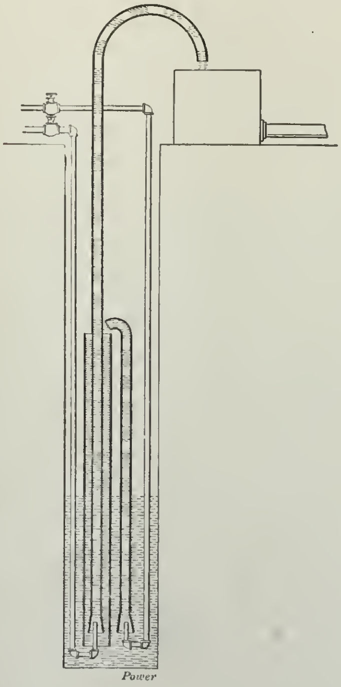


Fig. 2. Method of Improving Deficient Submergence

Fig. 2 shows the only way, known to me, of improving a deficient submergence, yet there are very few wells large enough to permit of this being properly installed.

With the correct proportioning and arrangement of pipes, the well under discussion can be pumped satisfactorily with the ordinary air lift, but not with as good economy as would be obtained with more submergence. From the figures given, the submergence would be about 31 per cent., while in the oilfields, wells are frequently pumped with a submergence of 25 per cent. or less. A submergence of 31 per cent. and a 110-foot lift with properly proportioned pipes would require approximately 1¼ cubic feet of free air actually delivered to the well (not compressor displacement) per gallon of water, and about 23 pounds air pressure.

G. A. Reichard.

Los Angeles, Cal.

Discharged without Cause

In the issue of August 9 under the above title Mr. Wilcox relates his experiences. It seems to me that he was a victim of circumstances to a certain extent, and possibly hasty action by the chief based on information from others.

From the number of machines which Mr. Wilcox was responsible for, as well as from the fact that he was required to help in the boiler room too, I draw the conclusion that there was an abundance of nerve displayed by the boss in expecting so much of one man, not to mention the act of discharging him without inquiring more fully into the trouble.

As to Mr. Wilcox's advice to make a thorough inspection on going on and off duty, this should invariably be the rule, and in some cases precaution should be extended a little further. If anything should be found out of order and likely to cause trouble and offer a chance for a dispute if possible, call someone's attention to it, who is not immediately interested. As long as two or more men are called to work on the same job, just that long will there be chances for disputes; and no innocent man should suffer for another's mistakes.

The chief's telling him that he was hoodooed recalls my own experiences in a small plant several years ago. The company required good references before employing me. I had been on duty but a few days when one of the quarter-box adjusting bolts of the main bearing of the engine broke, causing a shutdown. The bolt had been strained for some time, as the break plainly showed, and finally came apart, though I had not touched it with a wrench. The boss said nothing about this, but his looks did not belie his feelings.

The water supply was from a small river some 800 or 1000 feet from the mill. An antiquated pump placed in a pit was used to force the water into an overhead tank. In about two weeks after my advent the pump failed, due to the suction pipe rusting through. This caused another shutdown. This time the boss said a few things until I convinced him where the trouble lay, but it was evident that his opinion of me as an engineer was nothing to boast about.

In about four weeks, while starting up one Monday morning, the boiler-feed pump showed signs of distress. Apparently it was working against an enormous pressure. I made sure that the valves on the feed line were open and then inquired how long the boiler-feed line had been in. Several years, was the answer and I was sure that the pipe was closed with scale. At 9 o'clock we were compelled to shut down and what I was told would fill a book. The boss said that the other engineer before me had never had any such trouble with scale and he did not believe that I knew what I was talking about. After he had got through I quietly informed him that in case I failed to find the pipe choked with scale, he could retain all of my wages. When the boiler was sufficiently cool I reached through the top manhole with a three-wheel cutter and cut the feed pipe, taking it out in two pieces. It was a good load for one man, it was so full of scale.

Of course, if my predecessor had remained another month he would have had the same trouble. It was a case of the new engineer and trouble being due at about the same time, and I was the victim of the circumstance.

Joseph Stewart.

Hamilton, O.

Pumping Layout

In answer to the inquiry of H. B. Fillmen in the August 9 Power concerning the proper method of pumping several wells, I offer the following:

If it is possible, the use of a steam pump is preferable to an air lift. One pump is sufficient and will give better satisfaction than would an individual machine for each well. The steam consumption of the single unit will be less for equal work, as will also the friction losses and slippage. Further, the first cost of installation and subsequent maintenance will be lower.

Another thing to be observed is that probably these wells were bored one at a time and pumped individually. The yield then was 75 gallons per minute; but when all three are pumped simultaneously, it is highly improbable that the combined output will be anything like 225 gallons per minute. If 150 gallons are obtained from wells so close together the flow is unusually good.

In piping to the wells, a cross should be placed on the inlet of the pump and from it an individual pipe run to each well. A valve should be placed in each line, as it is unlikely that the quantity obtainable from the several wells will be equal. Each valve may be so set that the supply shall not be reduced in any one well so that air can come over and break the suction.

The original investment for a pumping system would be less than for an air compressor and its auxiliaries, and on the former the maintenance charges would be less.

The only occasion for using an air lift is when the water is so far below the surface that pumping is impracticable. Furthermore, water obtained in this manner cannot be used immediately but must be allowed time to free the entrained air.

In regard to the amount of air required to pump the well, the proposition with the data given is unanswerable. It is necessary to know the diameter of the well pipe and the distance of the water below the surface, as well as the quantity and the total lift.

Truman D. Hayes.

Cambridge, Mass.

There is no need of more than one pump to do the work in Mr. Fillmen's layout. Make the suctions as straight and short as possible, and connect a gate valve close to the pump in each line, as often one well will give a better flow than another, and each can be tested or cleaned without interference from or with the others. Before using this layout be sure that the water will not recede beyond the limit of suction, a very probable occurrence. The Pohlé system of pumping by compressed air is in certain cases much handier and, if properly installed, should be more economical and satisfactory in every way. Its initial cost is higher, however. Its advantages are; the suction will never come out of the water; no danger of freezing in cold weather; no lost efficiency on account of air leaks in suction or worn packing, and less cost of upkeep. Another advantage of this system for artesian wells is that the constant rush of water low down in the well tends to keep the well clean, and to some extent increase the flow.

In installing this system there are two important points to consider if the greatest efficiency is desired.

First, the lift pipe should be submerged at least as much as the pipe extends above the surface of the water in the well. However, there is no gain in extending it more than one and one-half times the hight.

Second, no higher air pressure than is enough for the lift should be used.

This "enough for the lift" cannot be accurately determined, owing to lack of reliable data, especially as Mr. Fillmen has not given the size or depth of his wells. Figuring from tests on wells of this type, with equal lift and submergence, and with the air pipe running down outside the lift pipe, 75 cubic feet of free air per minute compressed to 40 pounds should handle 225 gallons 75 feet and give an efficiency of 60 per cent. from the receiver, the best that can be expected of this small size.

I would advise Mr. Fillmen to correspond with some maker of this type of apparatus, from whom much more accurate information can be obtained.

O. Monthan.

Tucson, Ariz.

Rumbling in a Boiler

In the August 9 issue of Power there appeared an account of a "Rumbling in a Boiler," by M. H. Tarbell. Perhaps an experience which I had some years ago will help to solve the problem in this case.

The boiler was of 500 horsepower and had a 10-inch angle stop valve. It was reported to be rumbling badly under about the same conditions as are described by Mr. Tarbell. As this was considered dangerous, the fire was drawn and the boiler was allowed to cool. I then made a thorough internal inspection, looking especially for a loose feed pipe, loose stays or tools left in the boiler at the last cleaning, but found none and, moreover, everything appeared to be in good working order. Fortunately there was a gate valve between the main header and the stop valve. After bleeding the steam from the header the angle-valve bonnet was lifted off and the trouble was at once apparent. The disk had worked loose from the stem and the valve was working somewhat like a nonreturn valve. When the pressure in that particular boiler was low and no steam was being generated, the disk dropped onto the seat and remained there until the boiler again came up to pressure; then it should have lifted but as it fitted tightly into the body of the valve an excess pressure was necessary before the disk would lift. When it did lift the superheated water in the boiler generated steam so fast that it rumbled like the lid of a teakettle.

After the disk had been secured to the stem the rumbling ceased and no further trouble has been experienced.

Charles W. Long.

Wilkes-Barre, Penn.

M. H. Tarbell does not state in his letter in the August 9 issue what kind of a boiler it is which acts as described. I assume that it is either an internally-fired boiler without an underflue or a tubular boiler with a return underflue which, from some cause, is not effective. When getting up steam the water around the tubes is heated, while below them it remains comparatively cold. As soon as the safety valve blows, the water begins to circulate. This reduces the temperature of the steaming water and part of the steam is rapidly condensed causing a rumbling and chattering noise. The pressure will drop, of course, but is quickly restored by steam from the other boilers, if they are connected, and the heat from the furnace. The situation is not without danger as the reduction of pressure might cause an upheaval in the other boilers. I advise Mr. Tarbell to put the underflue in an efficient condition or to blow a little steam from the boiler intermittently while pressure is being raised in order to give the water a chance to circulate and acquire a more uniform temperature. I have had the same experience with portable boilers having round fireboxes. The steam pressure would drop from 75 to 25 pounds while the boiler was rumbling and shaking. Boilers suffer greatly under strains caused in this way and soon show defects, such as leaking tubes.

H. WIEGAND.

Indianapolis, Ind.

Accident to Blowoff Tank

In the August 9 POWER, R. E. Cooper shows a design of a blowoff tank and asks for criticism.

In my opinion the design is not half bad but it would be improved some if he would put an elbow on the blowoff pipe inside of the tank, and let it "look" in a horizontal direction either right or left so as to discharge the water around the inside of the tank. He shows a 4-inch vent pipe which is small enough for the purpose.

In regard to the tank which exploded, the wonder is that it did not explode sooner. It ought to be apparent to anyone that running the blowoff pipe to a point near the bottom of the tank or any distance below the water level is simply courting danger. In a tank of such design the water is lifted bodily from the bottom of the tank by the incoming water and vapor and projected against the top of the tank, especially if there is a leaky blowoff valve or if the water is thoroughly hot as it is likely to be after the first of a battery of boilers is blown down.

Another reason why the blowoff pipe should not be water sealed is that when through blowing down, the vapor in the pipe will condense and unless there is an air leak the pipe will fill with water which will become cool, and when blowdown time comes again a slug of hot water and vapor will shoot into a pipe full of cold water. Sometimes waterhammer does things.

A wonderful looking device is illustrated in the December 28, 1909, POWER. It is called a blowoff tank. If it possessed a few more trimmings such as a corn husker and a wheelbarrow it might be a useful article. A principal objection to it is that the blowoff pipe extends to a point near the bottom of the tank. The design must have been the result of a Welsh rarebit.

R. MCLAREN.

Berlin, Ont.

Leaky Floats

E. Granfield in the August 16 issue does not state whether the floats are for open-tank work, exposed to atmospheric pressure only, or for inclosed service and exposed to heavy pressure and possibly heat. In the first case solder will hold on any copper float if the float is properly cleaned and tinned before putting on the ordinary two parts tin and one part lead solder. This solder melts at a temperature of about 340 degrees Fahrenheit. A solder of one part tin and three parts lead melts at about 480 degrees.

If the solder is for floats of the second order the leak can be permanently stopped by using as a solder one part of hard spelter of zinc and two parts of copper. This solder melts at about 700 degrees. A blow torch or blow pipe will have to be used to apply it.

In either case the float should be heated considerably before attempting to solder, so that absolutely no moisture will be left on the inside. If ever so little is left, it is liable to blow through the solder when it is melted and leave a minute hole which may not cause much trouble at the start but will allow the float to fill gradually, and at the end of some days its usefulness will be gone. In all cases it is absolutely necessary to clean the copper around the leak before attempting to solder.

JAMES E. NOBLE.

Toronto, Can.

Packing a Feed Pump

In reply to George Kramer's inquiry in the August 9 issue as to the best method of packing a feed pump, I will say that I have been very successful in packing our feed pumps, which handle feed water at 210 degrees. The pump is 6x4x6 inches in size. It works against 135 pounds pressure.

I use ⅜-inch square hot-water packing. I let the packing lap about an inch or more by splitting it. It should not fit too tightly. The pump should be run fast after the packing is first put in until the water gets hot enough to make the packing expand. Some of this packing has lasted as long as six months under continuous operation.

I took the metal-expansion rings out of our pump, got a 3-inch brass sleeve, sawed it to the right length and used it to hold the packing in place.

T. P. THOMAS.

St. Paul, Minn.

Regarding George Kramer's letter on "Packing a Feed Pump," I submit the following:

All that I have used for years is square flax packing boiled in tallow. Any packing manufacturer can supply it. It comes to us on spools in sizes of ½, ¾, ⅞ and 1 inch.

Our feed pump takes the ½-inch size; the stuffing boxes hold four rings. The feed water is taken from a heater and is at a temperature of about 208 degrees. I have had packing of this kind last for six months. I use a liberal amount of graphite mixed to a stiff paste with cylinder oil applied to each ring. The water pistons are of brass and not always in the best of condition.

J. P. COLTON.

Ohio City, O.

That Mr. Kramer's packing lasted but two weeks seems to indicate that some condition is not right as neither the temperature nor pressure is excessive. The first two "guesses" would be that the cylinder is scored or that the water contains some substance that disintegrates ordinary materials.

For impure water I have used leather packing with success. This was made up of leather washers cut from an old belt, the outside diameter to fit the cylinder and the inside to fit the piston body. This material has the advantage of being easily shaped to accurately fit the cylinder in case it should be worn out of round.

For gritty water, I have used for packing, cotton-cloth buffing wheels that have been worn small in the shop. The grit works down into the mesh toward the center leaving the surface clean. At one time, using this material, I ran an old pump, with a cylinder so scored that ordinary packing would not last two days, continuously for several weeks without renewing the packing.

T. D. HAYES.

Cambridge, Mass.

Will the Pump Raise the Water?

In discussing W. F. Callister's problem, as to whether a pump will lift water 10 feet if its supply is the discharge line of an impeller situated 70 feet below, both Mr. Sperry and Mr. French in the August 16 issue seem to agree that it will, but neither gives the real reason why it will. If the impeller will raise the water 60 feet against atmospheric pressure, it will, when the pump removes this pressure, force it up the remaining 10 feet into the pump. Mr. French's statement that the impeller must not prevent the atmospheric pressure from being transmitted to the water is false as that has nothing to do with the matter.

E. S. FITZ.

Cleveland, O.

Engineers' Wages

I notice the fact in POWER that a great number of engineers are getting their minds to work upon the wage question. This question, if followed up as it should be, is bound to bring results for the general good of both employer and the engineer. Good, conscientious, straightforward and painstaking effort will be appreciated and a prosperous condition in general will be the result of the final amicable adjustment of the wage problem. The article on the front page of the July 26 issue contains the expression, "Wages mean power. Not only the necessities of life, but almost anything in the world, except contentment, can be obtained for money." Very true indeed. An engineer's power, importance or ability is most frequently judged by the wages he is known to receive. If he gets good wages he must be a good man, and in nine cases out of ten you can bet your bottom dollar that he is giving his employer value received.

Some fellows are content anywhere and under any conditions, while others are not. One's individual makeup has largely to do with this, although it must be admitted that the wage question is about the first, the middle and the last consideration, even though all other items pertaining to the deal are satisfactory. Another article in the same issue, by E. F. Henry, clearly demonstrates the injustice to the engineer practised by the employer. Ability counts for nothing, everything is considered purely from the wage point of view. What are we going to do about it? It appears to me that right at this point a properly conducted and thoroughly organized Union would be a very important adjunct in regulating what we may call the knifing of one engineer by another. The union should be, as Mr. Henry says, for education, publicity and an arrangement whereby and in conjunction with the educational features the members could be given a rating in accordance with their proved ability. The union could then say to the employer, A man such as you require is worth so much. No one having the required rating can be secured for any less money.

Remember the old adage, "In union there is strength." Bring into a union means for education and the true fraternal and brotherly love features which are made mention of in the ritual of the N. A. S. E. and it is my opinion that an ideal union will be evolved, one not only satisfactory and agreeable to its members but one which employers will readily and kindly give recognition and support.

After a union is once on foot, I think that a universal system of operating reports and requisitions could be agreed upon which would not only enable the employer but the engineer as well to know just what conditions are and what is purchased and used in the plant each day. The reports should be in duplicate. They would thus afford an indisputable record. Mr. Henry's idea is good. Employers have the right to know where every dollar goes, and it is up to us to be capable of showing them. We will not be able to do this if we do not educate ourselves.

Our wages will always be down if we do not hang together. Dissatisfaction will reign supreme if we do not jump in and push together for the general good of all and exert every effort to eliminate all antagonism among ourselves or between us and our employers. In my opinion there never was much gained by fighting. Fight, of course, if the cause is just and there is no other way out. A man without some nerve and temper is seldom any good as an engineer, but the man who is able to control his nerve and temper is, as a general thing, the fellow that gets along the best with other men. He commands the respect of his employer, quite frequently to the extent of drawing the most money.

Let us then be up and doing
  With a heart for any fate.
Still achieving, still pursuing,
  Learn to labor,—educate.

T. M. STERLING.

Middlebranch, Ohio.

Trouble with Barometric Condenser

I read with much interest W. P. Elston's article in the August 16 issue of POWER stating the trouble with the barometric condenser under his care. My experience with a siphon condenser was identical to his, but, through relentless effort and observation, I finally succeeded in mastering what seemed to be an insoluble mystery.

We have a 500-kilowatt vertical steam turbine to which is connected a barometric condenser. The distance from the overflow of the well to the exhaust nozzle is 34 feet, the same as in Mr. Elston's case. Needless to say that practically every time I started the engine large quantities of water would come over into the exhaust pipe, and cause a terrible water hammer, after which I would have to drain this water out through an 1¼-inch drain pipe provided for the purpose at the lower end. The pipe was so small that the process of draining was very slow.

Eventually the fact revealed itself that a compound condensing engine with a direct-connected air-pump and jet condenser, when started slowly will form a vacuum in the exhaust pipe and condenser which will extend back into the receiver, because, having hardly any load, the governor will allow the steam valves to admit but a small quantity of steam, sufficient only to maintain a good vacuum. Therefore, if the injection water valve is opened wide instead of just cracked, through ignorance or lack of judgment, disastrous results to the cylinder are liable to follow. The condensing water will flow into the condenser unresisted and, not being removed fast enough by the air-pump, it will accumulate and soon find its way into the cylinder, where it will cause trouble if no devices are provided to get rid of it, such as snifting valves, floats with vacuum breakers, etc.

On the other hand, if the load is starting to come, the governor will allow the steam valves to open enough at each stroke so that pressure will soon exist on the exhaust side of the piston in the low-pressure cylinder, thereby preventing the advance of the water to the cylinder, and by regulating the injection-water valve, a good vacuum can be maintained under reasonable conditions.

If these facts are borne in mind when operating an engine with a siphon condenser, not much trouble should be experienced.

The way in which I start my engine is as follows:

First, I crack the throttle to let the engine speed up slowly. I then start the water pump, after which, when I observe by the ammeters on the switchboard that the load is being thrown on gradually, I open the throttle five or six turns and cut the condenser in immediately by closing the drip valve at the bottom and allowing the cover of the relief valve on top of the condenser to close, and by opening the top and bottom seals of the turbine.

If the siphon condenser is cut in before the engine has any load, it is quite certain that the troubles which Mr. Elston and I experienced will occur because it requires such a small quantity of steam to bring a turbine up to speed that the vacuum will extend way back into the first stage. But, when the load comes on, more steam is required, and with a pressure above the atmosphere in the exhaust pipe of the condenser no water can possibly flow back.

If Mr. Elston will bear in mind never to cut in his condenser before the load starts to come, he will experience his troubles no more.

LUKE MARIER.

Fall River, Mass.

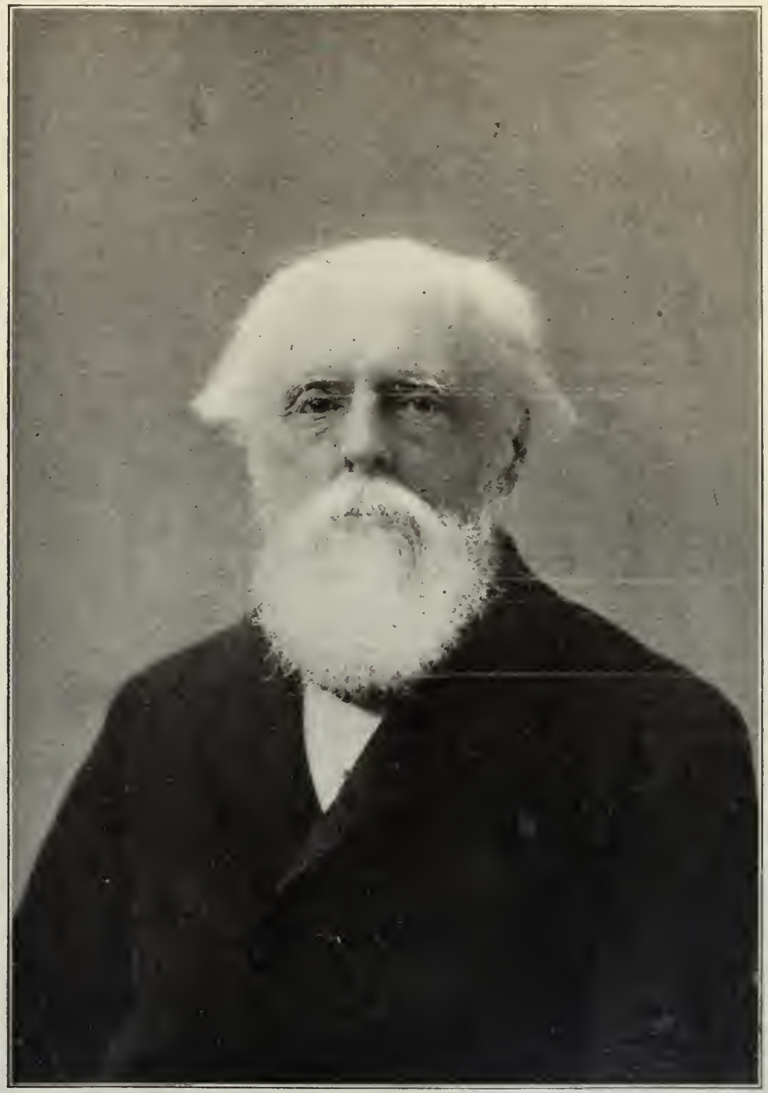
Safety valves are liable to stick on their seats; they should therefore be raised carefully at frequent intervals. Attach a small wire cable to the safety-valve lever, pass it over a pulley, and bring the end down to a convenient place for ready service in lifting the valve. See that the outlet to each and every safety valve is full size of the valve opening.

Charles Talbot Porter Dead At 84

On August 28, at his home in Montclair, N. J., Charles Talbot Porter passed away. He was born on January 18, 1826, at Auburn, N. Y. All that heredity could do for a man's brains was done for his. On his father's side he was a great-grandson of Jonathan Edwards, while through his mother he was descended from such New England families as the Phillipses and Winthrops. After graduating from Hamilton college, in 1845, he read law in the office of his father, Hon. John Porter, and thereafter practised for several years in Rochester and New York City. The college instruction of that day had taught him rather worse than nothing about mechanics, which proved to be his natural bent when he found it out. He soon acquired a disgust with the law, at which his clients seem to have been more proficient than himself. One of them beguiled him into a venture with a "centrifugal-force engine"—his first engineering experience—and another unloaded on him the patent of a stone-dressing machine. This machine, while impracticable as it stood, contained the germ of a good idea. Mr. Porter went to work to improve it, picking up by the way a knowledge of drafting and designing, and therein brought out his latent mechanical faculty so well that the machine proved a technical success and made its way commercially until the practice was introduced of sawing out, instead of dressing, the building stones for which it was intended and that fact destroyed its usefulness. The stone-dressing machine had been driven by an engine which governed faultily, to remedy which Mr. Porter was led to the design of his well known central-counterpoise governor (patented 1859). Into the manufacture of this he entered, equipping a shop in New York for the purpose. In 1861 Mr. Porter patented an isochronous centrifugal governor for marine work, which made its début on the German Lloyd steamship "New York," where it received high commendation. Previous to that time, the engineers had been obliged to throttle their engine by hand.

About this period, Mr. Porter called on a New York firm of hatters in relation to their steam plant and there fell in with their engineer, John F. Allen, who showed him the design of a variable-cut-off valve and gear that he had worked out. It was agreed that they should attempt to apply this valve mechanism in connection with Mr. Porter's governor to a high-speed engine, and thus the Porter-Allen engine had its inception. The indicator was then little used in America, and one adapted to high-speed work was not obtainable. At this juncture, and at the instance of Mr. Porter, Charles B. Richards designed his first indicator. Mr. Porter bought the patent. An instrument for testing the spring was devised by him. With the success of the engine elements thus far brought together, Mr. Porter set himself to making improvements in details of design—cranks, journal boxes, eccentrics, etc.—and he prepared an 8x24-inch engine for the London International Exhibition of 1862. This was made to run at 150 revolutions per minute, since he justly apprehended that a speed of 200 revolutions would frighten people. As Mr. Porter sailed for England to look after his exhibit, there occurred the Mason and Slidell incident which brought that country to the verge of war with the United States, and when he arrived he found that the space allotted to American exhibits had been confiscated. However, he was permitted to install his engine to drive some looms, in place of one that had failed, though with the proviso that he should not exceed a speed of 120 revolutions. But the engine ran so smoothly that the speed limit was removed. After some time, the engine was sold to drive a foundry blower at high speed. The indorsement implied by the purchase helped to convert the apathy with which it had been received in England, because it was noncondensing, into a wave of interest and appreciation.

British orders for governors now began to come in. Mr. Porter busied himself with the standardization of the Richards indicator and its manufacture in England, one application being to locomotive service, on which subject he read a paper at Newcastle in 1863, before the mechanical section of the British Association for the Advancement of Science. Improvements in the bed and other features of the engine also occupied him, and he applied the engine to condensing service, soon designing a high-speed air pump for this purpose. He had, in fact, entered upon quite a protracted sojourn



Charles Talbot Porter, Father of the High-speed Engine in Great Britain. Arrangements were made with Ormerod, Grierson & Co., of Manchester, for the manufacture of the engine.

From England he took one 14x24-inch engine, making 150 revolutions per minute, to an international exposition at Oporto, where he gathered some interesting experiences. One day when he could not get steam to run the engine, he was condoled with by the king of Portugal, which royal recognition proved useful to him later in getting out of the country without a passport.

Upon the insolvency of Ormerod, Grierson & Co., he was approached by the Whitworth company, which informally took up his engine and set about constructing one for the Paris exposition of 1867. Mr. Porter's acquaintance with Whitworth led to curious obscurations of personality and practice. A proposed written agreement with him failed of execution because of the demanded incorporation of a privilege for Mr. Whitworth to alter the engine at discretion. To this Mr. Porter dared not consent, fearing the result of Whitworth's aggressive individuality in a field new to him. Mr. Porter took five engines to the Paris exposition. An application of Ducommon & Co., of Alsace, for a concession to manufacture under the patents was accepted, for which reason a proposition later made by Farco & Co., the most eminent stationary-engine builders in France, had to be declined. Zerah Colburn, the editor of *Engineering*, procured of Mr. Porter materials for a series of articles in that paper. In 1868 Porter read before the Institution of Mechanical Engineers a paper on the Allen engine which led to an animated controversy over piston acceleration, the actual rate of which Mr. Porter was the first to point out.

Mr. Porter's English residence now drew to a close. Having received from his friend, George T. Hope, president of the Continental Fire Insurance Company, of New York, an encouraging offer of financial support if he would return to the United States and join Mr. Allen in the manufacture of the engine, he set about the plan of a model shop to be arranged substantially in accord with one designed by Smith & Coventry, near Manchester, Eng. It was a type with nave, two-story side bays and traveling crane. He returned to America in June, 1868. His little governor shop on West Thirtieth street, New York, had been kept running during his absence.

Upon reaching America, Mr. Porter found that the financial backer of the enterprise on whom Mr. Hope relied had leased a small building in Harlem and provided Mr. Allen very inadequately with funds for its equipment. Mr. Porter succeeded in obtaining from him some additional capital and started work. Incidental to engine building, some straight-edges and surface plates were constructed. Mr. Porter acquired at this time a valuable foreman in George Goodfellow, who came to him from the Whitworth works and who followed him in future business changes. Before the end of the first year, the unnamed capitalist before alluded to proposed the formation of a stock company. He was allowed a controlling interest, became president and is charged by Mr. Porter with serious mismanagement. Resistance to his unwise policy brought Mr. Porter to the verge of ouster for insubordination, though the situation was temporarily saved by Mr. Hope, who bought out the control. But the expiration of the lease on the shop, coupled with a lack of confidence on the part of capital anticipatory to the panic of Black Friday, 1873, forced a closing out of the business. Just at the end, an engine was built for the Clinton Wire Cloth Company, whose work peculiarly demanded high speed and uniformity.

Two engines had been exhibited at the American Institute fair, New York, in 1870, together with the Allen boiler, recently designed. This boiler had tubes of independent circulation inclined slightly from the vertical, superheated 23 degrees, and proved a great success. In their report, the judges, President Barnard, of Columbia, and Thomas J. Sloane, proposer of the gimlet-pointed screw, said: "The performance of the engine is without precedent."

Before the windup of the business, Mr. Allen had left to enter upon the manufacture of pneumatic riveters, invented by himself.

Manufacture of the engine was suspended from the spring of 1873 to that of 1876. This interval Mr. Porter improved by revising and standardizing its design, room for .the purpose being furnished him in the second story of Mr. Allen's little shop at Mott Haven. Resumption of construction work was effected by an arrangement with Mr. Phillips, surviving member of the firm of Hewes & Phillips, Newark, N. J. That establishment had of late been confining its attention to machine tools, in which business was so dull that a proposition from Mr. Porter was accepted to tide over the depression, even though he insisted in keeping the business in his own hands. In spite of a lack of ready money the construction of engines was carried on successfully.

The Edgar Thompson Steel Company, Braddock, Penn., at this time inquired for an engine to run a cold saw for rails, and Mr. Porter was able to furnish one which had been built to saw blocks for a wooden-pavement system that had been adopted for Washington, D. C., but abandoned. This led to sales of governors for other engines at those works and to the design of a new regulating valve.

At the Centennial Exposition, Philadelphia, 1876, Mr. Porter acted as a judge in coöperation with Professor Reuleaux, of Berlin, Horatio Allen, formerly president of the Novelty Iron Works, New York, and others. This furnished opportunity for some curious experiences and some caustic comments on human nature as well as on mechanical art. Mr. Porter took delight in describing to Professor Sweet the diminutive engine of his type that would equal in output the famous Corliss engine at the Centennial.

A 6x12-inch engine, to make 450 revolutions, was built for a bone-pulverizing mill at Newark, N. J., and on this engine was tried a new indicator invented by a young man who introduced himself as Harris Tabor. The diagrams proved free from such vibrations as the Richards indicator would produce at that speed, and with a certificate from Mr. Porter stating the conditions under which the cards had been taken, Mr. Tabor started for Boston to see Mr. Ashcroft.

An important share of Mr. Porter's attention was now given to building engines for rolling mills, and flywheel design came in for special consideration. An order from the Cambria Iron and Steel Company called for a 40x48-inch engine, larger than any that Mr. Porter had yet made, and most of the work, being too big for the Newark shop, was put out in Philadelphia. When installed, the engine increased the output of the roll train from 1200 to 3000 tons per week. The rapid action of Mr. Porter's engines was found incidentally to obviate the need of bloom chipping.

Before many years, the business had outgrown the Hewes & Phillips works and with Mr. Hope, who had an equal interest in it, Mr. Porter discussed the formation of a company with $800,000 capital, offering to assign his patents for $100,000 in stock. The Southwark works, Philadelphia, of the old engineering firm S. V. Merrick & Sons, were in the market for sale, and, impressed by their appearance, Mr. Porter advocated their purchase. The Southwark Foundry and Machine Company was now organized for this purpose though Mr. Porter had supposed that it would be called the Porter-Allen Steam Engine Manufacturing Company. From the start he began to meet disappointments premonitory of the obstacles later offered to his guidance of the work. A man connected with the earlier history of the establishment was elected president, while he was made vice-president with charge of manufacturing. At the start, an appropriation of $100,000 for new tools was refused Mr. Porter. The equipment proved to be in much worse shape than had appeared. A reissue of an important patent was needed at this time, and for some mysterious reason the patent solicitor opposed the idea of making application.

When made, it was at first rejected, but through Mr. Porter's personal activity was subsequently allowed. This saved Mr. Porter from inability to fulfil his agreement with the company but caused friction with the directors. Six months had passed without building an engine. Finally the desired appropriation for new tools was granted. Some were ordered from England; the modernizing of the equipment furnished Mr. Porter with quite a field for exercise of his ingenuity. Earlier in that year, 1880, he had participated in a meeting at the office of the *American Machinist*. at which it was determined to organize the American Society of Mechanical Engineers, and at its first session, held in Stevens Institute, he had read a paper on the strength of machine tools. A demand for engines for electric-lighting purposes was now created. Mr. Edison came along one day, wanting to buy a thousand engines but soon was convinced that he did not want more than twenty-four and finally concluded that six would suffice. Three Porter-Allen engines were actually installed for him. Overcharge, against Mr. Porter's advice, on an order for the Calumet & Hecla mine diverted business amounting to $100,000 in a year. A large engine was built by Mr. Porter for the Otis Steel Works, of Cleveland. Various engines were now being turned out, but difficulties of one sort or another met Mr. Porter at every turn, the chief obstacles being those within the works, where, in spite of the responsibility placed upon him for results, his ideas were disregarded both by those above and below him. As between him and the president, the directors chose to sustain the latter, and Mr. Porter's connection with the establishment ceased. The business—owing to bad management, the responsibility for which Mr. Porter has placed on others—was practically ruined, but was reorganized under management of James C. Brooks, then of the firm of William Sellers & Co., under whom it revived and has flourished.

Subsequent to retirement from connection with the manufacture of the engine that bears his name Mr. Porter lived to spend a long and honored portion of his career, practising the engineering profession in New York City, while he resided at Montclair, N. J. Ill health, resulting from an accident, forced him to abandon work at times, but generally he was active and alert in spite of his advanced years.

Mr. Porter was the author of "A Treatise on the Richards Steam Engine Indicator and the Development and Application of Force in the Steam Engine" (1874), and of a work of more general character, "Mechanics and Faith" (1885), in which he ably showed why he, a man of science, could none the less accept as true things which belong to a higher realm than that of the senses.

At the instance of the editor of POWER, Mr. Porter wrote a most interesting series of "Engineering Reminiscences," which ran in that paper and in the *American Machinist* during 1903 and 1906, suspension through the interval being occasioned by Mr. Porter's health. They were later published in book form.

As a writer, Mr. Porter was possessed of a graceful, accurate and attractive style, which apart from knowledge of his history suffices to stamp him as one of the men of old-school culture who are not now found so frequently in the engineering profession as at a time when education for mechanical pursuits was less specialized.

On April 13, 1909, Mr. Porter was awarded the John Fritz medal, which was established in 1902 by the profession of engineering as a meed of recognition for notable scientific or industrial achievements.

The story of his life constitutes a history of the rise of the high-speed engine.

Dwelshauvers-Dery's Laboratory

By F. R. Low

It was my recent privilege to be shown by Prof. V. Dwelshauvers-Dery, the distinguished professor emeritus of the University of Liège, through the laboratory of steam engineering which he proposed, when such a laboratory in connection with a university was unheard of, which he secured only after years of perseverance, and in which were conducted experiments which have made both the university and the experimenter famous in the field of steam engineering. In a corner of a little courtyard, with quiet paths and mouldy trees it stands, a modest little one-storied building with no suggestion of industrialism or mechanics in its appearance or surroundings, reminiscent of the attitude of the grave and learned dispensers of culture, of the date of its inception; that is, if such an innovation must be tolerated it must be subordinated to the classic atmosphere of the university. Passing through the anterooms and offices one reaches the laboratory proper in the center of which is the famous engine of Beer which, under the skilful manipulation of the professor and his corps of trained assistants, furnished the data for so much information as to the actual operation of steam in a cylinder and the relation of that action to the abstract laws which were supposed to govern it. This engine has so lately been described in POWER* that it is needless to repeat the description here in detail. The guides carry a scale divided into thousandth parts of the stroke (0.60 mm.) with a vernier which enables one to locate the position of the piston to the six-hundredth of a millimeter and the valves, which are riding cutoffs, have their separate scales and verniers, six in all, two for admission and one for exhaust upon both head and crank ends, all reading to the tenth of a millimeter. One original Richards indicator purchased at the exposition of 1862 is still upon the engine and we were shown the method of calibrating the indicators by dead-weighting their pistons. They are still so free from friction that the lines drawn when removing the weights coincided almost perfectly with those drawn when the weights were added. The reduced motion is taken from a rod sliding beside the indicators operated by an eccentric and rod having the same proportions as the main crank and connecting rod and the only cord is that, some 6 inches in length, used to connect the paper drum to the reciprocating rod, so that the movements of the piston of the engine and the paper of the indicator are strictly proportional.

Behind the engine is the measuring tank for the water, fitted with a weir and with an apparatus for measuring the hight of the water so delicate that the addition of four or five drops in a tank containing some 500 gallons will bring the water level into contact with a metallic point and produce a very apparent indication (a sort of inverted Hookes gage). The water level is measured with a scale which has to be read with a lens.

It is Dwelshauvers-Dery's pleasure and boast that his laboratory has turned out real scholars and experimenters; notably Georges Duschene, whose continuation of the work begun with the professor upon the interaction of the working fluid and the cylinder walls was brought to an end by his untimely decease, and Armand Duschene, his brother, who is carrying on the work in the same laboratory at present. With delicate thermocouples of silver and platinum wire exposed to the steam or embedded in the cylinder walls, he measures the varying temperature throughout the revolution and is able to plot diagrams of the temperatures of both metal and fluid for all sorts of conditions.

It is easy to determine the volume of the cylinder up to some point after the admission valve has closed, including that of the clearance. The pressure at this volume can be measured from the indicator diagram; the weight per cubic foot found from a steam table and this weight multiplied by the volume and the number of times it is filled per hour would be the amount of steam used per hour *if it were always steam.*

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*June 28, 1910, page 1159.

The steam consumption calculated from the diagram in this way is always less, usually by 20 per cent. or more, than the steam actually furnished, as was determined by weighing the feed water. This is because the steam parts with its heat to the cylinder walls and is present as water at the point in the stroke at which the pressure is taken from the diagram. This investigation of that action is therefore very interesting as throwing light upon this "missing quantity" the reduction of which is the main object of jacketing, compounding and superheating. Professor Duschene has promised to prepare for POWER in the early future a résumé of his work, including his latest and unpublished results.

Another of Professor Dwelshauvers-Dery's scholars, Professor Hanocq, is conducting in the same laboratory a series of interesting experiments upon centrifugal pumps. A pump is motor driven, the motor being also fitted with a prony brake. Connections are made to the induction passage and to various points in the diffusion vanes, and the vacua and pressure measured by mercury manometers. The water handled can be accurately measured, the efficiency of the motor under all loads is determinable, and the input of energy is shown by the electrical instruments, so that the efficiency of the pump, which may be changed from single- to two-stage, may be accurately measured under various conditions. The leakage is determined by blocking up the openings in the impeller and pumping water into the pump at various pressures, only that passing which the clearances will allow. The leakage may be thus measured either while the pump is standing or running. The investigation promises to be interesting, but has not yet progressed far enough to be discussed.

Professor Dwelshauvers-Dery is passing his period of well merited repose in a charming home upon the banks of the Meuse, across which he can look upon the scene of his earlier activities. As Dr. R. H. Thurston, accustomed to the much greater resources of Sibley and of Ithaca, once said of it to him, "It is a small equipment but it has turned out great work."

Improvements not Appreciated

By W. Osborne

The job was not a nice one but the man who came along with it was one who talked in a pleasing way. He had come a long way, more than twenty miles, across the country to reach this shop because he had heard that good work was done, that special efforts were put forth when a man was in a hurry and that a fair price was charged. Besides he had a job that he must take back with him.

This job proved to be the cylinder of a sawmill engine. It had been 12x16 inches originally, which was a long time ago, but had worn and been rebored until it was about 12¼ inches diameter and in need of reboring, but the place where it most needed repairs was in the steam chest.

The valve seat was worn down so that the ports had lengthened out and finally got down beyond anything provided in the original plans of the designer. The valve was a single D valve moved by a fixed eccentric. The steam distribution must have been very bad even with things in good shape.

After looking things over it was decided to make a false valve seat and new valve, and to rebore the cylinder and put in a bull ring with two snap rings.

This job had been brought to the shop shortly after seven o'clock in the morning. While a cut was being run through the cylinder a sketch was made and given to the patternmaker. That evening a valve and seat were cast in the foundry. The next day they were fitted and the seat secured to its place in the steam chest with brass screws. After fastening a cut was taken over it. The balance of the job had been done, but according to good practice the valve and seat should be scraped together, and there was another little point that had not been attended to. To get in a seat of reasonable thickness it had been necessary to leave very little clearance between the valve stem and the valve.

It had been explained that it would be necessary to watch things for a few days for the valve scraping had not been as well done as it would have been with more time available. Everything seemed to be understood and away went the man.

Two weeks later a letter was received. It calmly and judiciously set forth that the job done had been one to make even sawmill mechanics blush. Nothing had been right.

This letter came in the morning mail and after reading it the owner of the shop handed it to the young man who had been in charge of the job and who had done most of the work on the planing and fitting of the valve and false seat.

Without a word he handed it back and began to take off his overalls.

"What does this mean?" asked the owner in surprise.

"It means that I am going to see that job, if it is the last thing I ever do," was the reply.

Seeing his determination the owner told him where he would find the store which this man owned and where he could find the mill, even if he did not find the writer of the letter.

At the mill it was found that the engine was not in good shape. It was certainly working hard and making hard work of it. To help out the bolts a piece of timber was wedged from the steam-chest cover to a beam and another one from the cylinder head to another beam. Even with that help both joints were leaking. The steam gage on the boiler showed 100 pounds and the throttle was wide open. To furnish lubrication for the cylinder there was an old-style tallow cup that would hold about four ounces. It was tapped into the side of the governor body. The fireman explained that it was put there so that it could be filled. There was a valve made to be closed when the lubricator was being filled but this valve leaked so that it was necessary to close the throttle valve and shut the engine down to fill it. The worst sin that a man could commit around that mill was to stop it even for a minute. He also explained that the cylinder had arrived in the night and had been put on without any lining up. It had been found that it did not line with the crosshead and the piston bull ring had been filed so that things could be got together. The valve should have been filed to let the stem down a little farther, but instead of that it had been sprung enough to connect up. The eccentric had been rolled around so that the steam was admitted at the right time. It had taken until morning to get it going and it was allowed to run along without any load until the fireman and the men helping him had got breakfast. It then had been put to work pulling the mill and had run twelve hours a day until the Sunday following. On being examined the cylinder had been found somewhat cut and the valve and seat quite badly cut. The fireman could do very little except report their condition, and the letter resulted.

The millowner defended his position in the matter by referring to the practice on a little narrow-gage road that ran near. The master mechanic of this road was a friend of his and had posted him on all that was good practice.

Rather than have or make trouble the bill was paid, but so strong was this man's faith in his friend, the master mechanic, that he believed that all trouble would have been saved if the job had been planned and done by him.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for August, 1910, *was* 170,000.

September 6............ 35,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

Shop Inspections

When apparatus is built for a plant erected under the supervision of a consulting engineer, it is generally specified that it shall be inspected or tested at the shop to the satisfaction of the engineer before acceptance. Often in such cases the manufacturer is seriously embarrassed by the incompetence displayed in making the inspections. In most cases the consulting engineer is not an expert on the constructive details of all the apparatus necessary for even a small installation; and it is not a reflection on his ability, and should not hurt his pride, to acknowledge the fact when occasion may warrant it. Instead of taking this view of the subject, however, the consulting engineer often assumes that with his superior general knowledge of mechanics, he can make a bluff at the job and get away with it, without anyone being the wiser. He can generally fool his patron in such an attempt, but he never deceives the manufacturer regarding his ability.

Almost invariably the novice endeavors to impress the builder with his carefulness by picking flaws with details that could not possibly affect the usefulness or life of the apparatus, and if there should be faults to warrant a real kick, he usually fails entirely to notice them.

For example, in inspecting boiler work, great stress will often be laid on the exact spacing of the rivets, regardless of whether variations from the specifications noted affect the pressure required or not, while the laps at the longitudinal joints may be so short that the rivet heads come to the edge of the plate without causing comment. Undue calking of rivet heads, denoting poor riveting, will entirely escape his notice if the rivets have been made tight before pressure is applied; while a hair-line crack on the beading of a tube will call for his earnest remonstrance regarding the workmanship or the material. Whether the tightness displayed under the pressure test is produced by well fitted courses or undue calking is a matter that he does not understand the importance of determining, and no attention is paid to it. An inspector who is thoroughly posted on his work and has confidence in himself does not think it necessary to make a kick simply to show that he has performed his duty, but the novice generally assumes that the acceptance of apparatus without any fault finding is looked on as lack of care on his part rather than an indication of carefulness in manufacture.

If a consulting engineer has reason to doubt the honesty of a manufacturer, and is not himself an expert in the details of construction of the apparatus to be inspected, he should employ an inspector who is thoroughly qualified to perform the work; for only in this way can his client's interest be fully protected and justice be done the manufacturer.

As the Field Broadens

One of the salient features of the proposed Institute of Operating Engineers is the admission to full membership of only those who can demonstrate that they are operating engineers in the full sense of the term.

It is not expected that these men will be required to show that they can design boilers, pumps, engines and other power-plant machinery, but it is required that they shall have the mental attainments and possess the manual skill necessary for the operation of any power-plant equipment and the management of all classes of workers connected therewith.

As one after another of the callings, vocations and trades have by the application of labor-saving and labor-displacing machinery become comparatively simple craft operations and the skilled workers reduced to the status of "hewers of wood and drawers of water," the field of the operating engineer has broadened and the responsibilities have increased to a degree realized only by a few.

Not many years ago the generation and transmission of power concerned the engineer only to the extent of the physical processes occurring between the coal pile and the flywheel rim, and he covered that field only in a general way. Today his field embraces the composition of the coal before it is piled, that of the gases that go up the chimney and the transformed energy of the steam far beyond the switchboard. From being a general worker in one comparatively narrow field he has found it necessary, in order to keep abreast the times, to be an expert in several broad fields.

Anything that has to do with moving machinery is the operating engineer's field, and as one new form of power transformation after another has been developed—the high-power waterwheel, the electric generator and motor, the hydraulic and the electric elevator, the steam turbine and the internal-combustion engine, as well as the machinery they drive—they have all been placed in the care of the engineer. It is the purpose of the Institute through its subsidiary branches and correspondence system to prepare apprentice and junior members for full membership by systematic training.

Apprentices and junior members are the foundation on which the Institute will be built and the man who passes through the lower grades to full membership will demonstrate by his progress that both mechanically and mentally he has earned his standing in the organization. In this connection it may be interesting to know that hundreds of able and prominent operating engineers from all parts of the country are giving the matter of organization their serious attention, and universities, colleges, schools and extensive employers of skilled operatives have promised hearty coöperation in the effort to found and support the proposed Institute, in recognition of its object to furnish the industries of the country with a class of men whose skill and fitness for responsible positions will be practically assured.

Entropy

Time was when the practical engineer did not have any use for entropy; if he ran across the word he considered that it referred to some abstruse theoretical matter that was interesting to professors; if he had been harassed by the conception as a student he put it from his mind and turned his attention to real things.

But the steam turbine has changed all this and now the practical engineer is confronted by entropy diagrams and entropy tables. What is entropy any way? Various answers are given to this question, depending on the point of view, and all are more or less correct.

Now entropy was originally a mathematical device invented independently by Clausius and Rankine for solving problems in steam engineering; Rankine called it the thermodynamic function, and Clausius invented the name now adopted.

In order to understand the reason for such a device it will be necessary to look a little further back in the history of thermodynamics. The first experiments in thermodynamics were made on air because they were the easiest, and fortunately the properties can be expressed by comparatively simple equations. For example, at constant temperature the volume is inversely proportional to the pressure, and at constant pressure the volume is directly proportional to the absolute temperature. These laws can be combined into an equation with three variables,

$$pv = RT$$

Coming now to steam, the experiments were vastly more difficult and were dangerous, and it was only when the French government gave Regnault a subvention that anything like precise knowledge could be obtained. His results have remained standard partly because he was an exceptional experimenter and even more because of the great difficulty of making better experiments. It is, as it were, but yesterday that better experiments have been completed and even these have nowhere near the precision of experiments on air.

In order to find the influence of the externals on the behavior of a substance like air or steam it is necessary to determine first how they would behave without such interference and such investigations must be made by computations because the externals like the walls of a steam-engine cylinder are always with us. Such an investigation can be made easily enough for air, resulting in the well known adiabatic equation for air, and there is no reason for injecting the conception of entropy into this discussion, although it commonly gets there. As a matter of fact the equations were deduced before the function or device called entropy was invented.

But when it comes to saturated steam it will be found that all the properties, pressure, volume, total heat, etc., depend on the temperature and the derived equation with three variables that would be derivable is not forthcoming. The three variables are there and this becomes evident when investigation is made of the adiabatic action of steam, for some of the steam condenses and gives the so called dryness factor usually represented by the letter x. This represents the part of the steam which is not condensed, so that $1 - x$ is the water of condensation.

Now if there are three variables and it is desired to calculate a numerical result, it is very convenient to get rid of one of them. In the case of air there is a convenient example in isothermal expansion represented by the equation

$$pv = constant$$

All that is necessary is to put in a value of p and calculate v. The adiabatic equation for air is nearly as good, for with logarithms the same thing may be done with that equation.

But for steam there is no such royal way. There have been attempts to make approximate equations of that sort for steam. Rankine worked out an exponential equation but it is not general enough.

Turning to the isothermal for air once more, the trick is done by making the temperature constant; a mathematician who is used to thinking in abstractions has no difficulty at all in dealing with the adiabatic action in like manner. He says, Let us try making something constant; we do not know what its form will be and it has no name, but at any rate assume that there is such a thing (only he calls it a function) and see what happens. In the case of steam the result was inevitable, for Clausius and Rankine worked it out independently and no one has been able to work out anything else. This is no place for the analytics of the case, which can be found in any textbook on thermodynamics.

The whole matter is further complicated by the reason that the professor, being original, saw at once that if there was an entropy for steam there must be one for air and proceeded to determine it. And he has good reason for dealing with the entropy of air when teaching students, for the whole discussion is easier than steam. But nobody except the professor and the students who suffer under him have any practical use for entropy of air; and the only reason for computing numerical values is to show that it can be done, which is more interesting than useful.

There is one inconvenience about adiabatic computations for steam—they can be made forward but not backward; a pressure can be assumed and the corresponding volume computed, but if a given volume is desired there is nothing to do but guess and try. It is easier to make several computations and draw a curve. Hence the reason for temperature-entropy diagrams for steam. Such diagrams must be on a large scale, but they are otherwise convenient. A table is even more convenient and more precise. Anyway, the engineer has only to learn how to work the diagram or the table and he gets his results and can leave others to wrangle over meanings and philosophies.

Now the engineer is the man who does things; if he can invent a way for himself, or if someone will do the trick for him, and if he gets results, he is satisfied. But the physicist is a man who finds out things, and whenever he has a device which will do one thing, he tries to see if he cannot do something else with it. So he does not rest content with the idea that entropy is a device for solving problems in steam engineering; he enlarges and develops the conception and uses it for other purposes. From this come the various conceptions concerning entropy.

Leaky valves are as bad as a hole in your pocket.

A monkey wrench makes a poor hammer; don't use it for one.

Putting up a line of steam piping means more than cutting threads and screwing lengths of pipe together.

The fellow who is making the most of a small job is far ahead of the fellow who is making a botch of a big one.

Inquiries of General Interest

Proper Heat for Babbitt

How can I tell when babbitt is hot enough to pour when babbitting shaft boxes?

D. J. C.

When the melted metal will char a pine stick thrust into it, it is hot enough to pour satisfactorily.

Efficiency of Joint and Change of Pitch of Rivets

In a boiler, if the pitch of the rivets is 2½ inches and the efficiency of the joint is 72 per cent., what would be the efficiency of the joint if the pitch were changed to 3 inches?

C. H. F.

If the joint with a pitch of 2½ inches was correctly designed the plate and the rivets would be of equal strength and increasing the pitch without increasing the diameter of the rivets would weaken the joint in proportion to the increase in pitch.

Babbitt Sticking to Shaft

Will babbitt stick to an ordinary shaft if the latter is not oiled?

D. J. C.

In some cases it will, especially in the case of bearing shells, such as are used on many high-speed engines. It is better to coat the shaft with plumbago (black lead) or chalk before pouring. When babbitting bearings for shafting it is better to employ a mandrel the size of the shaft. The heat from the babbitt may spring a shaft of small diameter, if it is of steel.

Variation in Gage Readings

In a battery of boilers is it possible for the steam gages to indicate a difference in pressure? Is so, what is the cause?

S. R. B.

It is not only possible, but quite probable that when a number of gages have been used for a considerable length of time the pressure indicated by them will vary to a slight degree, say, from 1 to 2 pounds.

When the gages are connected up exactly alike and subjected to practically the same temperature, the difference is due to the effect of the alternate expansion and contraction of the spring, or to the fact that the springs have become "set" at different pressures.

The difference in pressure may be due to different vertical distances measured between the bottoms of the gages and the points at which the gage pipes are connected to the water columns or boilers. The gage pipes being of different lengths contain varying "heads" of water, which produce corresponding variations of pressures in the gages.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Composition of Coal

What is the composition of bituminous coal?

B. B. L.

A good quality of bituminous coal contains about 61 per cent. fixed carbon, about 31 per cent. of volatile matter, known as hydrocarbon, 7 per cent. of ash and 1 per cent. of sulphur. The exact composition depends on the quality of the coal, etc.

Air Supply in Combustion

Why is it necessary to provide a supply of air for combustion in the furnace?

J. W. B.

As combustion can only take place through the union of oxygen with combustible matter and as the only source of oxygen is the air, it is necessary that air pass through the fuel in order to become thoroughly mixed with the volatile matter and gases being distilled from the fuel. If the air is allowed to pass over the fire, very little of the oxygen can combine with the combustible matter of the fuel.

Duplex Pump Cushion Valve

Please describe the operation of the cushion valve on duplex pumps.

L. A. M.

A duplex steam pump is generally provided with four ports connecting the steam chest and cylinder, the two outer ports admitting steam only, the two inner ports being used for exhaust steam only. A small port or bypass connects the steam port and the exhaust port at each end of the cylinder, the bypass being opened and closed by means of a valve called the cushion valve.

When the cushion valve is closed the steam in the cylinder escapes through the exhaust port into the cavity under the valve and thence into the exhaust pipe. When the piston covers the exhaust port the escape of the steam is shut off and what remains in the cylinder is compressed in the clearance space, arresting the movement of the piston before the end of the stroke is reached.

When the cushion valve is open the exhaust steam flows through the bypass from the steam port into the exhaust port, reducing the volume of steam compressed in the clearance space, and the stroke of the piston is lengthened. By manipulating the cushion valve according to the speed of the piston, the length of the stroke may be regulated and the piston prevented from striking the cylinder head.

Necessity of Air in Combustion

How can you prove that it is necessary to supply air through the fire for combustion?

W. R. O.

By shutting off the air supply, either by closing the damper or allowing the fire to become so thick or clinkered that the air cannot pass through it.

Combustion

Why must air be heated before combining with the coal?

J. A. D.

The air does not combine with the coal, but the oxygen in the air combines with the combustible matter in the coal. It is the combustible matter that must be heated before oxygen will combine with it.

Drilling Boxes for Babbitting

Is it necessary to drill holes in a box to keep the babbitt from coming out or working loose?

A. W. B.

For bearings for line and countershafts it is not necessary to drill holes for the purpose of keeping the babbitt in place, but it is quite necessary to do so in the case of connecting-rod brasses, main bearing boxes and all bearings that are subjected to shocks and the reversal of the direction of rotation. In the latter cases it is better to pene the babbitt thoroughly and then bore the brasses or other bearings to the size of the shaft or pin.

Operation of Dampers

What effect on combustion has the closing and opening of the damper?

H. M.

Closing of the damper cuts off the admission of air and prevents the fire from getting a proper supply of air; opening the damper permits air to pass through the grates, and has the effect of promoting combustion.

New Power House Equipment

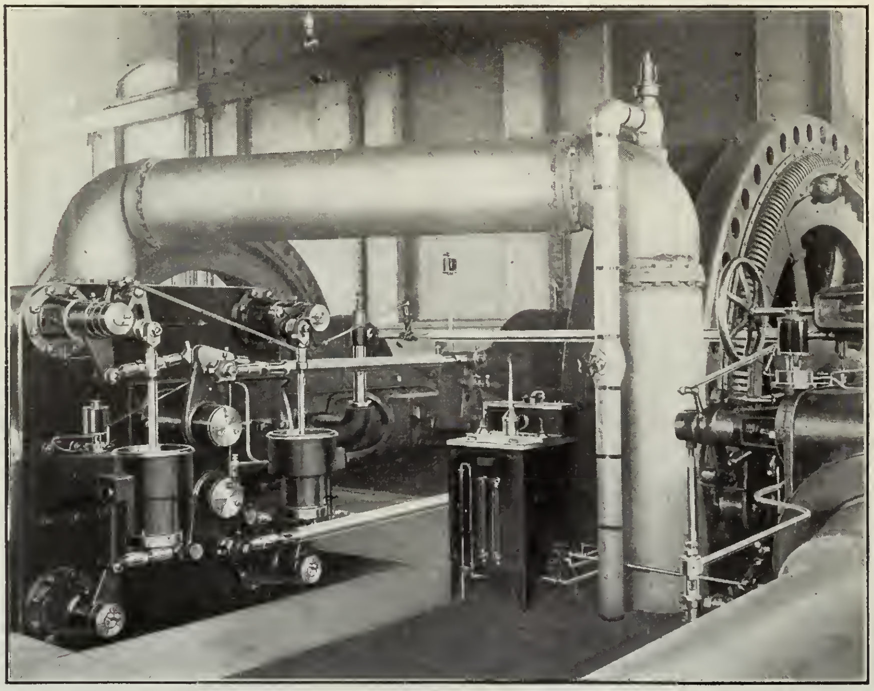
What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Richardson Automatic Oiling and Filtering System

The Richardson-Phenix Company, Milwaukee, Wis., is putting out a line of unit-oiling systems, which obviate the necessity of connecting two units to a common pipe line, or having any storage reservoirs or drainage tanks in the basement or overhead.

The filter is intended to be placed on the engine-room floor. A double-ended pump is attached to the filter and arranged to be driven from some part of the valve motion on the engine. A coupling is provided for the driving connection of this pump, which may be easily disconnected. A handle is provided so that the pump may be operated by hand when desired.

This pump handles the dirty oil at one end and clean oil at the other. One end of the pump is connected to a drainage-receiving well, to which all the drains from the engine are piped. This well is of cast iron, and is intended to be placed in or under the floor. It is provided with a substantial cap. A large, removable strainer catches the greater part of foreign matter draining away from the engine bed and bearings.



Arrangement of Filter and Oiling System

The pump takes the dirty oil from the receiving well and forces it into the upper back chamber of the filter, where a heating coil is provided for warming the oil to assist in the separation of foreign matter and water. The overflow from this chamber is gradual, although the incoming oil may come in irregular quantities. This chamber is provided with a settling basin in the base for retaining such foreign matter as will settle at this point.

Flowing from this chamber, the oil enters a central separating chamber and passes down through a central tube, rising up through water, where the separation of dirt is carried on to a still further extent. This chamber is provided with a gage glass, so that the amount of water and oil may be observed. It is also provided with an automatic overflow so the amount of water will remain constant.

Overflowing through adjustable fittings, the oil from the upper chamber enters the filtering cylinders, which are wrapped with filtering cloths, the oil finally coming to rest in the main storage body of the filter, where a cooling coil may be provided if oil passing through the filter is large enough in quantity to demand it; or if the temperatures are such as to make it necessary to reduce the temperature of the oil.

The filtered oil is pumped up into a glass tank supported on a pipe stand, in which is an inner pipe serving as an overflow, carrying the oil down and out of the side opening in the base of the stand, the excess oil returning back to the pump.

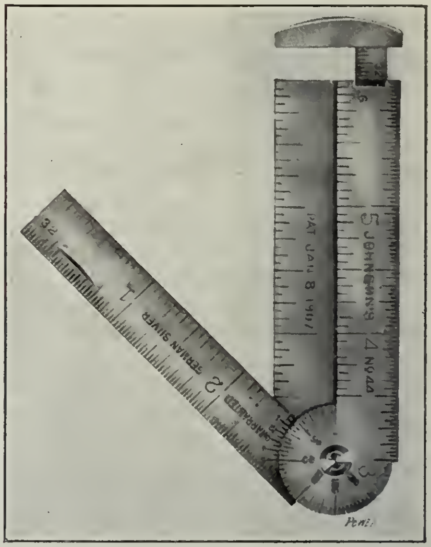
The glass tank on the overflow stand indicates the general operation of the system and offers a means of determining the efficiency of the action of the filter.

From the base of the overflow stand, a single pipe with its branches supplies the several sight feeds about the engine. A valve is provided in this line near the overflow stand, which may be closed when the engine is shut down. The opening and closing of this valve are all that is necessary in starting and stopping the entire system.

Johnson's Patent Combination Rule

This handy rule is 6 inches long, but doubles up on a center joint which is so constructed that the rule remains fixed wherever set.

The center joint is designed as a vernier, divided into ten parts, which correspond with nine divisions on the protractor scale. Each division on the protractor represents five degrees, so that each division on the vernier is smaller by one-half degree than the protractor spacing.



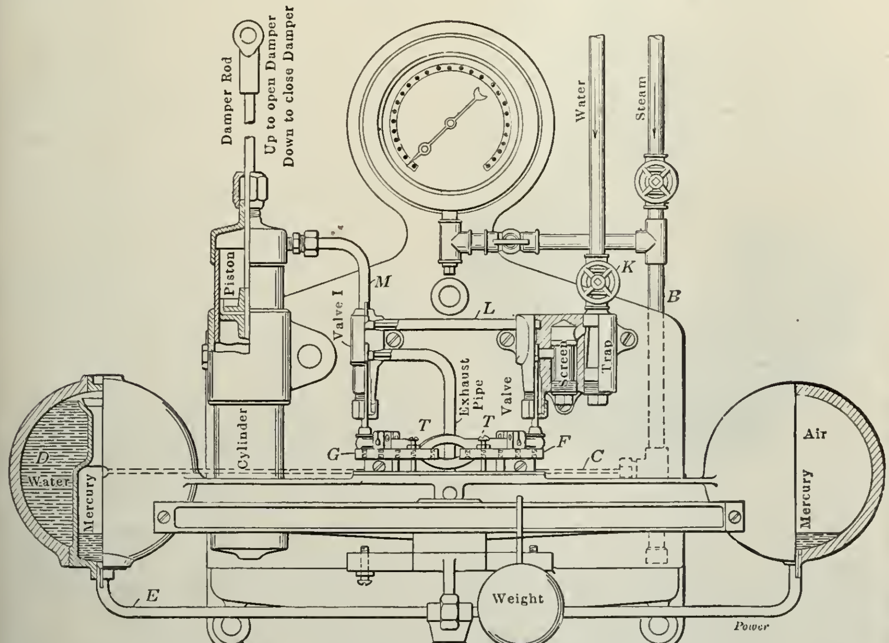
Johnson's Patent Combination Rule

When determining the angle to which the rule is open, it is necessary to read the protractor scale, count on the vernier the number of spaces up to the division line coinciding with one of the rulings on the protractor, and then add to the angle read off on the protractor the number of half degrees indicated by the vernier. Distinctive features of the rule may be readily seen upon examination.

The rule is made of german silver and is manufactured by the E. P. Johnson Rule Manufacturing Company, 553 Monroe street, Chicago, Ill.

The Diamond Furnace Regulator

The accompanying illustration shows the design of the Diamond damper regulator, which has been placed on the market by the Diamond Power Specialty Company, of Detroit, Mich. The novel feature in the construction of this machine is that of using mercury as a balancing power to control the inlet and outlet water valves.



Showing Details of Diamond Furnace Regulator

This mercury is used in connection with two hollow globes which are supported by a connecting arm, which rests on a knife edge in the center. These hollow globes are also joined by a bypass, through which the mercury moves back and forth as the steam pressure increases or decreases. The movement of the mercury from one globe to the other brings an increase in weight on one side or the other of the knife edge, as the case may be, which causes the connecting arm to tip. The tipping of this arm opens or closes the water valves which control the damper.

Another feature is the simple method of regulating the steam-pressure point at which the damper may be closed, which provides for changes between day and night pressure of steam or for a change at any time desired. This is accomplished by simply sliding the weight on the bar to the right or left.

An important point in the construction is the ease of access to the water valves, which are so constructed that the stems may be reground into the valve seats without the removal of the stem from the valve body or the removal of any other parts of the regulator.

The regulator is connected to a water or air supply. This supply should have a pressure of at least 20 pounds. It is also connected with a steam line carrying boiler pressure.

The pipe *B* carries the steam to a flexible-drawn copper tube *C*, which connects with the globe *D*. This globe is filled with water due to condensed steam. As the globe fills, air is forced through the inner chamber and the bypass *E* and the mercury is driven before it into the hollow globe on the right. As the pressure in both of the globes becomes equalized, the mercury drops back into the bypass, where it remains to be shifted from one side to the other, according to the increase or decrease of the steam pressure. When the steam reaches the required pressure, the counterweight is placed on the scale in such position as to exactly balance the two hollow globes.

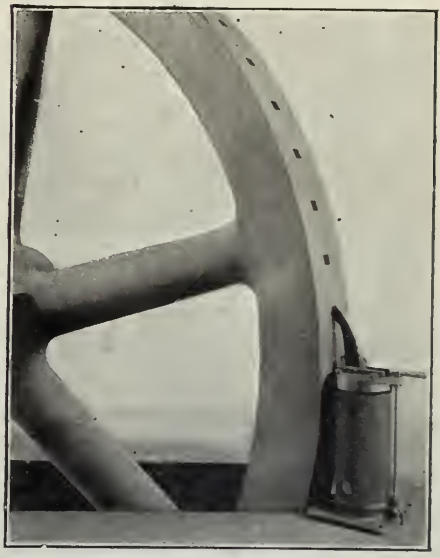
As the steam increases above the required pressure, the mercury is driven toward the hollow globe on the right, which increases the weight on that side and tips the arm. The tipping of the arm opens the inlet water valve by means of a lever connected at *F* with the valve stem and extending across the knife-edge rest and touching the balance arm by the use of an adjustable set screw *T*. The water, before reaching the inlet valve, passes through a trap or settling chamber and a screen. After passing the inlet valve the water flows through the pipes *L* and *M* to the cylinder, driving the piston down and closing the damper.

As the steam pressure decreases, the pressure in the globe *D* on the left also decreases and the compressed air in the hollow globe on the right assists in driving the mercury back to the left, causing the left end to become heavier. This movement of the mercury tips the arm down to the left side and closes the inlet valve which shuts off the water supply. At the same time it opens the outlet valve *I*, which allows the water to flow from the cylinder back through the pipe *M* into the exhaust pipe; the piston is drawn up by the weight of the damper as it opens.

The Badger Engine Jack

The Wisconsin Engine Company, Corliss, Wis., have designed a new type of engine jack which operates with either steam or compressed air and is designed to turn the flywheel quite rapidly, or ⅛ of an inch at a time, as desired.

The illustrations show the jack in position and the method of application. On large engines, and particularly engines getting hard service, where frequent inspection and quick adjustments are necessary, this jack is a saver of time and labor.



Badger Engine Jack

If there is more than one engine and it is not desirable to equip each with a permanent jack, a base can be set in front of each wheel, and permanent piping laid from the source of steam or air supply to each base. An air or steam hose can be substituted for the permanent piping by connecting from a convenient plug in either the air or steam line.

This jack is built for any diameter or weight of wheel, and can be used for engines of all types and sizes, having square rim wheels with the ordinary barring holes.

NEW PUBLICATIONS

Fuel and Refractory Materials. By A. Humboldt Sexton. Published by D. Van Nostrand Company, New York, 1910. Cloth; 360 pages, 5x7 inches; 104 illustrations; numerous tables. Price, $2.50.

This is the second edition of Professor Sexton's well known textbook, and the title page bears the legend: "New and revised edition"; the preface also informs the prospective reader that "the whole book has been carefully revised." It is unfortunate that the author used the adverb "carefully," in view of many of the statements and expressions found in the body of the book. A few of these are: "air which contains about 4 per cent. of CO_2, the oxygen being reduced by a like amount," on page 13; "the absolute zero for the Fahrenheit scale is —461 and that for the Centigrade scale —273," on page 32; the heat lost in flue gases "can be obtained by multiplying the products of combustion by their specific heats and the temperature at which they escape," on page 34; "39,600 B.t. units of heat," page 37; "the heat of combustion of benzene is found to be 1,418,310 B.t.u.," page 43; "the hydrogen in solid fuels is in the solid condition, and therefore heat must be absorbed in melting it," page 45; "the Priestman engine, which is the best known" oil engine, page 317; "1 B.t.u. = 772 foot-pounds," pages 311 and 352.

Besides the many blemishes of the kind just mentioned, there are careless inconsistencies and errors in copying or proof-reading; for example, on page 40 the heat value of carbon burned to CO = 4350, but on page 354 it has risen to 4370 B.t.u. On the latter page, the heat value of CO burned to CO_2 is given as 4370 instead of 10,130; on page 68, short flame fat coal is called "caking" coal when "coking" is meant.

The discussion of liquid fuels is perfunctory and skimpy, covering less than six pages, while forty-five are devoted to charcoal and coke. In discussing the heat value of fuels the author follows throughout the confusing and highly unpractical method of considering the atomic weights as pounds; a molecule of carbon being regarded as weighing 12 pounds, one of hydrogen two pounds, and so on.

In many places the tone is too dogmatic to be scientific; for example, in the statements on page 27 regarding dissociation. On page 15 the author puts forward the ancient and discarded argument (apparently in an effort to be scientifically precise) to the effect that when oxygen and hydrogen combine in what is termed "combustion," it is not necessarily the hydrogen that is burned —it may be the oxygen. And yet "the whole book has been carefully revised." Of what value is revision if it does not modernize and rectify?

The section on charcoal and coke and their production is far and away the best in the book All through, the purely descriptive parts are much superior to the analytical and scientific discussions.

OBITUARY

Joseph Garbett, for many years prominently identified with the mechanical industry of Minneapolis, died in his home in that city on Monday, August 22, of heart trouble, at the age of 58 years. He had been in poor health for some time, but although of late unable to discharge his duties as chief engineer of the Minneapolis Steel and Machinery Company, he had seemed to be gradually mending, and had expected soon to resume his position in an advisory capacity at the works. A sudden change took place, however, and he passed away.

Joseph Garbett

Mr. Garbett was born at Shropshire, England, in 1852. His early education was obtained under the greatest difficulties and only by exercising that high degree of determination which was characteristic of him and which led to his later success in life. He was a self-made man in every respect, having worked himself up from the lowest rounds of the ladder to positions of great responsibility in the mechanical world. The first years of his commercial life were spent in the Coalbrookdale Iron Works, England, finally becoming a superintendent in these works when about 23 years of age. In 1879, he left England, coming direct to Minneapolis and entering the employ of O. A. Pray & Co., a manufacturer of flour and saw-mill machinery, as foreman patternmaker. He steadily developed into a thorough and able mechanic, and when the Twin City Iron Works was established, he and O. P. Briggs constituted the firm, Mr. Garbett serving as chief engineer. It was here that he designed the Twin City Corliss engine which is so favorably known throughout the country today.

When the Twin City Iron Works was absorbed by the Minneapolis Steel and Machinery Company, Mr. Garbett was prominent in its affairs, becoming mechanical engineer of that company and a director as well.

In addition to his other interests he was also senior partner in the firm of Garbett & Schneider, which does a large business in roll corrugating and kindred lines.

Mr. Garbett, although a forceful, able man of affairs, was of a retiring disposition and, though genial and always a true and loyal friend, he was chiefly devoted to his home, his business and his church. He was a mechanic of high standing and recognized ability, was a member of the American Society of Mechanical Engineers and an honorary member of Minneapolis No. 2, of the National Association of Stationary Engineers.

PERSONAL

Edward J. Kunze has been appointed assistant professor of mechanical engineering at the Michigan Argicultural College.

BOOKS RECEIVED

Chemist Pocket Manual. By Richard K. Meade. The Chemical Publishing Company, Easton, Penn. Leather, 443 pages, 4x6¾ inches, 39 illustrations, indexed. Price, $3.

High-Speed Steel. By O. M. Becker. McGraw-Hill Book Company, 239 West Thirty-ninth street, New York. Cloth, 360 pages, 6¼x9⅜ inches, 273 illustrations, indexed. Price, $4.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE OF THE TWO-CYCLE TYPE. Alfred George Scholes, Ilford, England. 967,250.

TURBO - GENERATOR CONSTRUCTION. Charles H. Smoot, Chicago, Ill., assignor, by mesne assignments, to General Electric Company, a Corporation of New York. 967,254.

PNEUMATIC MOTOR. Eugene T. Turney, Rock Island, Ill., assignor to Artista Piano Player Company, Milan, Ill., a Corporation of Illinois. 967,271.

WIND-WHEEL ELECTRIC GENERATOR. Alfred Carlson, Klamath Falls, Oreg. 967,325.

WATER WHEEL. Thomas A. MacDonald, Clifton, N. J., assignor of sixteen and one-third one-hundredths to Isaiah E. Zimmerman, Paterson, N. J., sixteen and one-third one-hundredths to William H. Castles, Kingsland, N. J., and sixteen and one-third oneundredths to George Lentz, North Arlington, . J. 967,402.

WAVE MOTOR. Arzeno W. Reynolds, erris, Cal. 967,437.

TURBINE. Julius Brown, Peekskill, N. Y. 67,494.

STEAM TURBINE. Alexander Albert lolle, Olst, Netherlands. 967,639.

REVERSIBLE INTERNAL COMBUSTION NGINE. Frederick T. Flinchbaugh, York, enn. 967,749.

INTERNAL COMBUSTION MOTOR EN- INE TURBINE OR REACTION WHEEL. ohn Hutchings, London, England. 967,778.

COMPOUND INTERNAL COMBUSTION NGINE. Robert M. Pierson, New York, . Y., assignor, by mesne assignments, to . P. Power Company, Newark, N. J., a Corporation of New Jersey. 967,828.

BOILERS, FURNACES AND GAS PRODUCERS

GAS GENERATOR. Nelson Goodyear, New ork, N. Y., assignor to J. B. Colt Company, ew York, N. Y., a Corporation of New Jersey. 967,183.

HYDROCARBON BURNER. Frederick R. Blount and Nelson Blount, New York, N. Y. 67,315.

ACETYLENE-GAS GENERATOR. John M. Flitcraft, Elkhart, Iowa, assignor of one-half to J. W. Fryer, Osceola, Iowa. 967,357.

AUTOMATIC FORCE - DRAFT SMOKE CONSUMER. James R. Harrell, Cincinnati, Ohio. 967,375.

SUCTION GAS PRODUCER. Constantine Lee Straub, Milwaukee, Wis., assignor to Loomis Pettibone Company, a Corporation of Connecticut. 967,459.

VAPOR BURNER. Harry V. Dresbach and Warren R. Morey, Iola, Kan. 967,622.

OIL BURNER. George Franklin Moody, Altoona, Penn. 967,653.

WATER - TUBE STEAM GENERATOR. August Gottlob Burkhardt, Düsseldorf, Germany. 967,718.

STEAM BOILER. Edward P. Hanahan, Springfield, Ohio. 967,763.

POWER PLANT AUXILIARIES AND APPLIANCES

INJECTOR. Robert Grundy Brooke, Macclesfield, England. 967,717.

THREE-WAY VALVE. Constantine Lee Straub, Perth Amboy, N. J., assignor to Marine Producer Gas Power Company, a Corporation of New York. 967,460.

ROTARY STEAM VALVE. Scott W. Perry, Versailles, Ill. 967,827.

AUTOMATIC AIR PUMP. James A. Kenworthy, Oakland, Cal. 967,199.

GAS CLEANER. Leon P. Lowe, San Francisco, Cal. 967,212.

WATER-LEVEL CONTROLLER. Edward P. Noyes, Winchester, Mass., assignor to C. P. Power Company, Newark, N. J., a Corporation of New Jersey. 967,228.

PACKING. Hakon Wilfred Ramberg, Brooklyn, N. Y. 967,235.

VALVE REGULATOR IN CORLISS ENGINES. Soffus Strommerson, Sioux City, Iowa, assignor of one-half to Edgas Beaumont, Sioux City, Iowa. 967,261.

BALANCED STEAM FITTING OR JOINT FOR JOURNALS OF ROTARY STEAM CYLINDERS. Alonzo Aldrich, Beloit, Wis. 967,294.

BELTING. John W. Hilton, Bellevue, Ky. 967,379.

PACKING. John C. Kingsbury, Dudley, Mass. 967,392.

CARBURETER FOR EXPLOSIVE ENGINES. Godfried J. Mayer, Buffalo, N. Y., assignor of five-twentieths to Donald R. Clark and four-twentieths to Herman Mayer, Buffalo, N. Y. 967,407.

CONTROLLING MEANS FOR TURBINES. Henning G. Sahlin, Malden, Mass., assignor of forty-five one-hundredths to David Rice, Boston, Mass. 967,444.

UNION CHECK VALVE. Harry H. Groh, Monocacy, Penn., assignor of one-half to James R. Kline, Monocacy, Penn. 967,514.

VALVE-RESEATING DEVICE. Andrew M. Lockett, New Orleans, La. 967,532.

WATER-COOLED PISTON FOR GAS ENGINES. Gustav B. Petsche, Philadelphia, Penn., assignor to Southwark Foundry and Machine Company, Philadelphia, Penn., a Corporation of Pennsylvania. 967,551.

STARTING AND REVERSING MECHANISM FOR GAS ENGINES. John V. Rice, Jr., Bordentown, N. J., assignor to Rice Gas Engine Company, Bordentown, N. J., a Corporation of New Jersey. 967,559.

FURNACE-DOOR OPENER. William J. Rohan, Racine, Wis. 967,565.

PIPE JOINT. Wesley Christy, Globe, Ariz. 967,611.

WATER GAGE. George H. Goodwin, Los Angeles, Cal. 967,634.

VALVE. Cassius Clay Palmer, New York, N. Y., assignor, by mesne assignments, to the Railway and Stationary Refrigerating Company, New York, N. Y., a Corporation of Maine. 967,659.

HOSE COUPLING. William H. Shepheard, Portsmouth, Va. 967,679.

MOTOR FOR DRIVING BOILER-TUBE CLEANERS. Henry F. Weinland, Springfield, Ohio, assignor to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 967,693.

VALVE MECHANISM. Edward V. Anderson, Monessen, Penn., assignor of one-half to Charles E. Golden, Crafton, Penn. 967,702.

STEAM-ENGINE AIR PUMP. Thomas Thompson, Blackheath, England. 967,860.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC-WELDING APPARATUS. Alvin E. Buchenberg, Toledo, Ohio, assignor, by mesne assignments, to the Toledo Electric Welder Company, Cincinnati, Ohio, a Corporation of Ohio. 966,382.

TIME SWITCH. Willeston C. Pugh, Seattle, Wash., assignor of one-half to August Gamblee, Seattle, Wash. 966,118.

INDUCTION LAMP. Peter Cooper Hewitt, New York, N. Y., assignor to Cooper Hewitt Electric Company, New York, N. Y., a Corporation of New York. 966.204.

ELECTRIC DRILL. Frederick W. King, Cleveland, Ohio. 966,217.

DYNAMO ELECTRIC MACHINE FOR THE PRODUCTION OF CONTINUOUS AND ALTERNATING CURRENTS. Jakob Schurch, Bruchsal, Germany. 966.246.

ELECTRIC SWITCH. John F. Smiley, Louisville, Ky. 966,252.

CIRCUIT BREAKER. William E. Richards, Portage, Penn. 966,360.

TEST COUPLING FOR ELECTRIC CONDUCTORS. Patrick J. McDonald, Rochester, N. Y., assignor of one-half to George H. Smith, Rochester, N. Y. 966.400.

INDICATOR FOR ARC LAMPS. Joseph H. Allen, Chicago, Ill., assignor, by mesne assignments, to General Electric Company, a Corporation of New York. 967,151.

LOW - SPEED CIRCUIT CONTROLLER. Fred B. Corey, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 967,171.

ALTERNATING-CURRENT MOTOR. Valere Alfred Fynn, London, England. 967,362.

INSULATION BAG FOR CONDUITS. Anton Funke, Westig, Germany, assignor to the Papyrus Artificial Paper Manufacturing Company, Hoboken, N. J. 967,360.

METHOD OF ELECTRIC WELDING AND REPAIRING. Heinrich L. J. Siemund, New York. 967,579.

ELECTRIC TERMINAL. Eduard Gerhardt, Jr., Mauer, near Vienna, Austria-Hungary. 967,631.

ARC LAMP. Charles H. Sprague, Newton, Mass., assignor to Starbuck Sprague, Boston, Mass. 967,684.

ELECTRIC ARC LAMP. Thomas Joseph Digby, London, England. 967,738.

MOTOR - CONTROLLING APPARATUS. Samuel H. Keefer, Plainfield, N. J., assignor to Niles-Bement-Pond Company, Jersey City, N. J., a Corporation of New Jersey. 967,782.

POWER PLANT TOOLS

PIPE WRENCH. Peter E. Erickson, Port Chester, N. Y. 966,182.

PIPE WRENCH. William T. Bennett, Mason City, Ill. 966.300.

LIFTING JACK. Alfonzo F. Wilson, Worthington, Ind. 966,374.

EXPANSION BOLT. James H. Baker, Springfield, Ill. 966,515.

RATCHET DRILL. Joseph M. Reams, New York, N. Y. 966,597.

WRENCH. Homer J. Munday, Carterville, Mo. 966,698.

WRENCH. Adolphus C. Clements, Villanow, Ga. 966,888.

WRENCH. Hugh McDonald, New Castle, Penn. 966,943.

ELECTRICAL-ALARM SYSTEM. Albert Goldstein, New York, N. Y., assignor to International Electric Protection Company, a Corporation of New York. 966,903.

WRENCH. Dwight Moody Russell, Bulger, Penn. 967.062.

LIFTING JACK. Charles E. Hulander, Pittsburg, Penn., assignor to the Duff Manufacturing Company, Pittsburg, Penn., a Corporation of Pennsylvania. 967,126.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Preparing Boiler for Inspection

The inspection of a steam boiler is a matter of vital importance. When notified to prepare a boiler for internal inspection, proceed to cool down and clean the boiler. Remove all scale, clean out the ashes and soot on the grates and in the combustion chambers, and sweep off the plates, heads and tubes, so that every part of the boiler may be examined and its exact condition determined. Have the boiler cold for the inspector when he arrives. Have the steam gage ready for testing, water-column connections open for inspection, and everything in readiness to facilitate a thoroughgoing inspection of the boiler and all its connections. —The Fidelity and Casualty Company of New York.

BUSINESS ITEMS

The Vilter Manufacturing Company, Milwaukee, Wis., has recently received contracts for six refrigerating plants from 23 to 80 tons; also orders for five Corliss engines, three of which will be used for driving refrigerating machines.

The Syracuse Industrial Gas Company, Vernon New York, is to install shortly a 35-kilowatt complete gas-power lighting outfit for the Cycle Hatcher Company, Elmira, N. Y. This will be a suction producer and generator direct connected to gas engine. This is a repeat order.

The Magnolia Metal Company, 112 Bank street, New York, is in receipt of a letter from H. C. Martell, of Balboa, Canal Zone, Panama, in which he says: "I suppose there is more Magnolia metal being used on the machinery bearings on this 'big ditch' than on any other large job in the world, and it is without doubt giving all us engineers entire satisfaction."

The Wheeler Condenser and Engineering Company, of Carteret, N. J., builders of Wheeler dry-tube condensers, Wheeler rectangular jet condensers, Wheeler admiralty type condensers, Wheeler-Edwards air pumps, etc., has recently acquired the drawings and patterns of the Conover condenser and are prepared to furnish condensers of this type and repair parts for those now in use.

The Ohio Brass Company, of Mansfield, Ohio, will occupy booth No. 80, with its exhibit at the Rochester convention, and will show a complete set of samples of its entire line, especially featuring its regulating valve, gage cocks, water gage, etc., with section samples. The company will be represented by W. T. Jameson, general sales agent and C. N. Allerding, of the engineering department.

The Sterling Lubricator Company, of Rochester, N. Y., is erecting a new factory for the manufacture of its Sterling force-feed lubricators. The new plant is expected to be in operation by the first of the new year. The demand for Sterling lubricators has increased to such a large number that the company has found it necessary to triple its output, and the new factory has been designed with this end in view.

Bulletin 1051, on "Revolving Field Alternators," recently issued by the Ideal Electric and Manufacturing Company, of Mansfield, Ohio, not only gives the details of construction of a new line of alternators which they have recently perfected, but also gives test curves and data showing the performance of these machines under various conditions. It also contains discussion and curves showing the influence of power factor on efficiency, heating characteritics, exciting current and power necessary to drive the machine. These generators are of the revolving-field type and are built in sizes up to 250 k.v.a. and in all standard voltages. The bulletin contains numerous illustrations and will be sent to interested persons upon request. This is the first of a series of engineering pamphlets to be published and the Ideal Electric and Manufacturing Company will be pleased to place the names of engineers who are interested upon their mailing list.

In a 96-page book entitled, "High Efficiency of Centrifugal Pumps," the De Laval Steam Turbine Comapany, of Trenton, N. J., has brought together some 75 charts, diagrams, photographs and a vast amount of engineering information relating to such subjects, as ways and means for testing centrifugal pumps, charts of the results of such tests, the interpretation of these charts for the purposes of the engineers, etc. Valuable suggestions are given for the drawing up of specifications and for testing pumps, to ascertain whether or not specifications have been made. In this connection it is stated that the De Laval Steam Turbine Company guarantees beforehand the characteristics of the pumps built by it and subjects each pump to a thorough test at the works. The purchaser is invited to witness this test and a full report is supplied to him. This book will undoubtedly prove of interest to anyone having to do with pumping machinery in any way.

A new firm of belting manufacturers has been organized, known as the Olmsted-Flint Company. This firm will maintain its principal offices at 136 Liberty street, New York City, and a branch house at 1026 Arch street, Philadelphia. It is also announced that branch houses will also be established in the near future in Chicago and Boston. The incorporators of the new company are: Willard I. Olmsted, Stanley B. Flint and George S. Baker. The Olmsted-Flint Company will manufacture a complete line of leather belting, balata belting, canvas stitched belting, rubber belting and solid woven belting. It has taken over a large plant in Philadelphia for the manufacture of its leather belting. The firm will also represent exclusively in the United States several large European manufacturers of belting and transmission specialties. The members of the new concern are well known in the belting field, having been connected for some years past with one of the largest of the belt manufacturers. One of the features of the new organization will be the establishment of an engineering and experimental department, which will make a study of belt conditions and methods of reducing costs in both the transmission and conveying field. F. M. Schiffmacher, a well known transmission engineer, will head this department

NEW EQUIPMENT

Prince Albert, Sask., will install a new tiltration plant.

Winnipeg, Man., will expend $60,000 on a new gas plant.

Souris, Man., will install a complete new waterworks system.

The town of Walworth, Wis., voted to issue $18,000 bonds for waterworks.

The town of Rainy River, Ont., will build a large new pumping station.

The Harvey Shoe Company, Gardner, Me., will install a new water wheel.

The city of Higginsville, Mo., contemplates enlarging its electric-light plant.

The Spring Lake (N. J.) Water Works will build a new pumping station.

The Pipper Light and Power Company will equip a plant at Montreal, Canada.

A new ice plant will be erected by the Pottstown (Penn.) Brewing Company.

The Ottawa (Ont.) Light, Heat and Power Company will equip a large new substation.

New Glasgow, N. S., will spend $25,000 in equipping a complete new waterworks system.

The Peoples Coal Company, Taunton, Mass., is taking prices on hoisting engines and boilers.

C. E. Giddings, Hudson, Mass., is in the market for a 12- to 15-horsepower gasolene engine.

Peterboro, Ont., is calling for tenders for electrically operated air compressors and ejectors.

E. H. Lundy has asked the council at Eldera, Ia., for a franchise to install an electric-light plant.

The people of Osyka, Miss., voted to issue $20,000 bonds for waterworks and electric-light plant.

The town of Akron, Iowa, is advertising for bids on the installation of an electric-lighting plant.

The citizens of Bellefourche, S. D., voted to issue $50,000 bonds for the construction of waterworks.

The Central Georgia Packing and Provision Company, Macon, Ga., will erect an ice and cold-storage plant.

Daniels & Fisher, Denver, Colo., are building a large addition and expect to install some new power equipment.

The South Shore Traction Company has purchased site for a power plant on Oakwood avenue, Bayport, L. I.

The city of Terrell, Tex., voted to issue $10,000 bonds for improving its electric-light plant and waterworks.

The citizens of Cambridge City, Ind., voted to issue $8000 bonds for improving its water and electric-light plant.

The Narragansett Electric Lighting Company, Providence, R. I., will install a new steam turbine and generator.

The city of Fairview, Okla., will extend its electric-light system, install power equipment, etc. J. N. Voorhees, city clerk.

The city of Shelbina, Mo., contemplates installing a boiler in its electric-light plant. E. P. Weaver is superintendent.

W. W. Womack, Jackson, Miss., contemplates developing water power to generate electricity for lighting and power.

The Peoples Railway, with headquarters at Stratford, Ont., will be in the market for large quantities of electrical equipment.

The city of Boston, L. R. Rourke, superintendent of streets, is requesting bids for two steam turbine driven centrifugal pumps.

The citizens of Albertville, Ala., voted to issue $25,000 bonds for waterworks. J. B. McCravy Company, Atlanta, Ga., engineers.

The Hocking Valley Creamery Company, Athens, Ohio, recently organized, will erect a plant for a general creamery and ice-cream business.

The Suburban Gas Company will erect a plant at Houston Heights, Tex. The company was recently incorporated and granted a franchise.

The city of Thibodaux, La., will open bids September 23, for the construction and installation of an electric-light plant. R. J. Maquin, clerk.

The East Liverpool (Ohio) Traction and Light Company has purchased a building at Yellow Creek, which will be remodeled and equipped as a power station.

Power and The Engineer

Vol. 32 NEW YORK, SEPTEMBER 13, 1910 No. 37

IN Europe conservation long ago ceased to be a fad. On the Continent conservation is a vital issue; the people are in dead earnest,—they have to be. Farms are cultivated to the very edges of the highways and railroad rights of way. The unending struggle is to make the ground yield more and to make that which is yielded go farther. The people there practise economy to a degree which is as yet undreamed of in America.

What applies to the food products applies also to fuel. Industrial prosperity demands a fuel of reasonable cost.

Some countries, as for example, Sweden, Norway, Denmark, Finland and Holland, have practically no coal deposits and are forced to import what they need. During war times a country would be seriously handicapped if its fuel supply were suddenly interrupted. For this reason the governments of those countries which have little coal are keenly interested in the development of a suitable substitute which can be produced within their own borders.

Peat, which is found in great abundance in all parts of Europe, has given satisfaction as a fuel. In Holland it has been used for centuries. The production of peat at the present time is over a million tons per year. In Russia the annual output amounts to over four million tons. The peat-fuel resources in other parts of Europe are proportionately well developed. America also has vast peat deposits. The United States, exclusive of Alaska, has over eleven thousand square miles of peat bogs. Canada has more than thirty-seven thousand square miles of bog land.

In many places, especially in Canada, peat fuel would find a very ready sale if it were offered at a price which would compare favorably with that of coal.

It is easy to burn and is comparatively free from smoke. Its heat value is about 40 to 60 per cent. of that of coal, and it would appear that the most efficient way in which to use peat for power generation is in a gas-producer plant.

The peat-fuel industry has not been a success in America due to the comparatively low cost of coal and the imperfect machinery which has been used for handling the peat.

The Canadian government is doing a great work at its experimental stations in demonstrating that peat can be used profitably for industrial purposes. These stations are described in this issue.

The United States Government also is making considerable effort to encourage the development of the possibilities which peat fuel possesses. Several large industrial and engineering concerns are making investigations to determine for their own benefit the availability of peat for power generation.

These facts indicate that the time is not far distant when peat will play an active part in the conservation of the American coal supply just as it is doing at the present time in the less richly endowed countries of Europe.

The methods of handling and using this product are not as yet highly perfected. However, the fact that so much interest is being taken in peat, is a token that great progress may be expected within a few years.

Canada's Experimental Peat Plants

By A. R. Maujer

> *At the government bog lands peat fuel is dug by hand, pulped in a mixing machine and spread out on the ground to dry. The total labor cost of all of the operations, including the loading of the peat into freight cars, is $1.63 per ton. At the fuel-testing station gas is made from the peat and is used in a Koerting gas engine which drives a 50-kilowatt generator.*

The coal deposits of Canada are all located in the western and eastern portions of the country; practically no coal is found in the central parts. The anthracite which is used in the central provinces is obtained from the Pennsylvania district. Due to the cost of transporting it such a great distance the cost is high. In Ontario and Quebec anthracite of very ordinary grade costs $7.50 per ton; in Manitoba the cost is as high as $10. For these reasons a satisfactory substitute for coal which could be made to compete with it commercially would find a ready market.

Peat, which is found in great abundance through Canada and the United States, is just such a substitute. Doctor Chalmers, of the Geological Survey of Canada, has estimated that there are considerably more than 37,000 square miles of workable peat-bog land in Canada. The depth of the bogs varies from 5 to 10 feet. This estimate is based on investigations which have approached thoroughness only in the territory east of Lake Superior. Throughout the western section of the country vast areas exist which have not been investigated but which undoubtedly have immense peat deposits.

Numerous attempts to manufacture fuel from peat have been made from time to time in various parts of Canada. None has proved entirely successful. The long succession of failures has served to discourage further efforts. The Canadian Government, recognizing the economic possibilities in a thriving peat-fuel industry, is endeavoring to stimulate the interest and enterprise of bog owners and manufacturers by demonstrating that peat fuel can be produced cheaply and that power for industrial purposes as well as heat for domestic use can be obtained therefrom. To accomplish this a portion, comprising about 300 acres, of what is known as the Alfred bog has been acquired and fuel is being manufactured. The peat is used at the fuel-testing station in a peat producer-gas plant of 60 horsepower capacity. The bog is situated about 40 miles east of Ottawa in Prescott county, province of Ontario. It covers an area of approximately 6800 acres. The peat varies in depth from 3 to 17 feet.

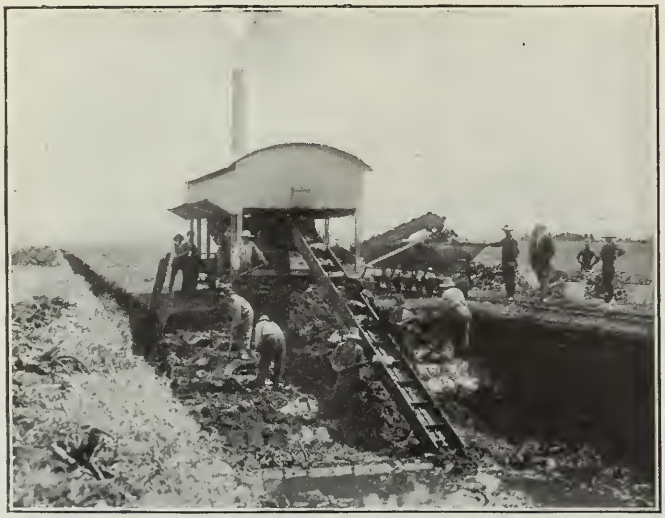


Fig. 1. The Mixing Machine and Method of Digging the Peat

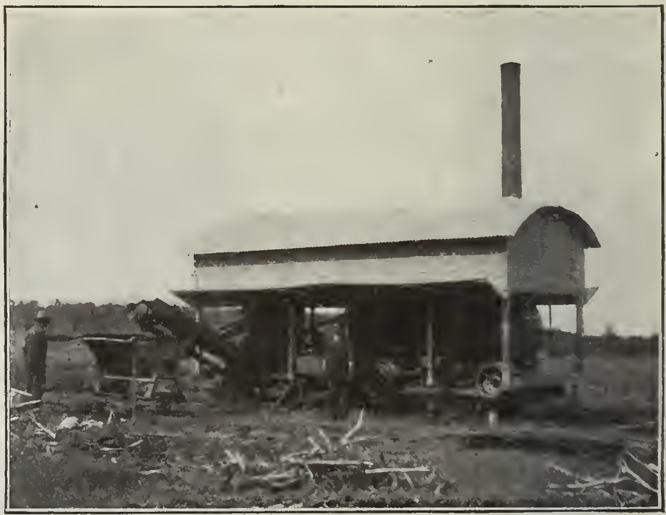


Fig. 2. Another View of the Mixing Machine

That the peat is suitable for fuel is shown by the following analysis which is given in Bulletin No. 4 of the Canadian Department of Mines.

ANALYSIS OF PEAT.
Absolutely Dry.

	Per Cent.
Volatile matter	68.23
Fixed carbon	26.00
Ash	5.77
Nitrogen	1.76
Sulphur	0.218
Phosphorus	0.033
Calorific value in B.t.u.	9005.000

Before actual manufacturing operations were started at the Government bog about 24,000 linear feet of drainage ditches were run late in the summer of 1909. This required the excavation of about 9300 cubic yards. The excavating was done entirely by hand, and cost approximately 8½ cents per cubic yard.

While the ditches were being dug an Anrep peat machine, together with other machinery and equipment, was purchased. brought to the Government bog and made ready for operations which were started in the summer of the present year. The machine has a capacity of about 30 tons per day.

The manner in which the peat is dug and handled is shown in Fig. 1. The men work at different levels so as to secure uniformity in the composition of the peat after it has passed through the machine. The peat is dug out with large sharp-edged spades and thrown onto the elevator. At the upper end the peat falls off into the hopper of the mixing machine in which all roots and stems are cut into small pieces and the peat is so thoroughly mixed that it resembles a pulp when it emerges. The cutting is accomplished by a number of knives mounted on a shaft which revolves at a speed of about 260 revolutions per minute. These knives pass sets of stationary knives which are fixed to the interior surface of the casing. A number of broad helix-shape blades mounted on the same shaft, do the mixing and force the peat out of the mouth of the machine onto a belt conveyer. The peat falls from the end of this conveyer into small steel dump cars which are hauled to and from the drying field by an endless cable. The cars travel on light portable steel tracks which are laid down on the roughly leveled surface of the bog.

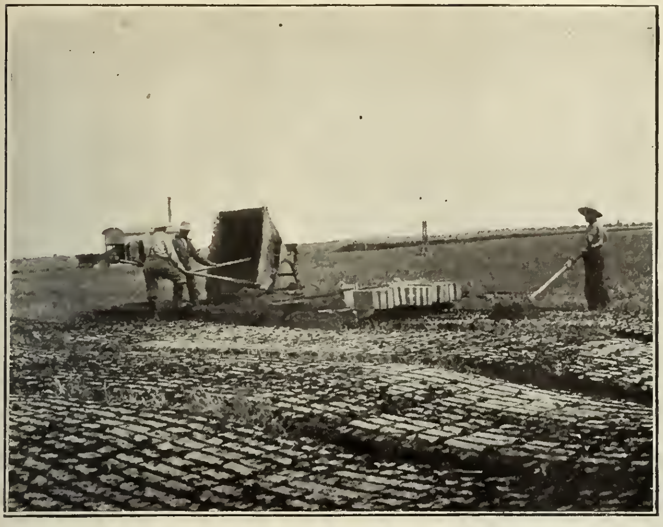


Fig. 3. The Jakobson Field Press

Fig. 4. Peat-storage Shed, Blacksmith Shop and Office

The drums which move the cable may be seen in Fig. 2. The manner in which the cable is gripped by the car is also shown in this figure. The upper of a pair of jaws attached to the truck of the car is raised and lowered by a cam and weighted lever. When the upper jaw is raised the cable rides on a grooved roller mounted on a spindle in the middle of the lower jaw and does not come in contact with either jaw. When the lever is thrown over the weight holds the movable jaw down and the cable is firmly gripped. The boiler and engine are built integral with each other and are made as light and compact as is consistent with durability. The rated capacity of the engine is 35 horsepower. The boiler is fitted with a grate suitable for burning either peat or brushwood. The entire outfit is mounted on a steel frame which is carried on car wheels. When it is necessary to move the apparatus, it is warped along over ordinary steel rails by means of a cable which runs through a pulley attached to an anchor and is wound over a slowly moving geared drum on the machine.

When a car full of peat arrives at the drying field, one man disconnects the grip and dumps the peat from the car into a Jakobson field press which travels along at the side of the tracks. The press is shown in Fig. 3. It consists of a wooden frame the front part of which travels on a wide wooden roller. The peat is roughly spread out on the ground within this frame by two men. The rear part of the frame is covered. Under the cover there is another wide wooden roller. This roller is set about 5 inches above the surface of the ground and is rotated by the front roller to which it is connected by chains located at the sides of the frame. The roller and frame press the peat out into a smooth flat strip about 6 feet wide and 5 inches deep. The back of the frame carries a set of 14 wooden knife blades which are held down by as many weights. The knives divide the strip of peat into 15 ribbons of equal width. These are cut crosswise into lengths of about 12 inches with suitable circular knives by the man who empties the cars. The object of cutting the peat up in this way is to hasten the drying by exposing more surface to the air. The press is slowly dragged along at the desired rate by a cable wound over a drum at the engine.

Fig. 5 is a diagrammatic plan of the

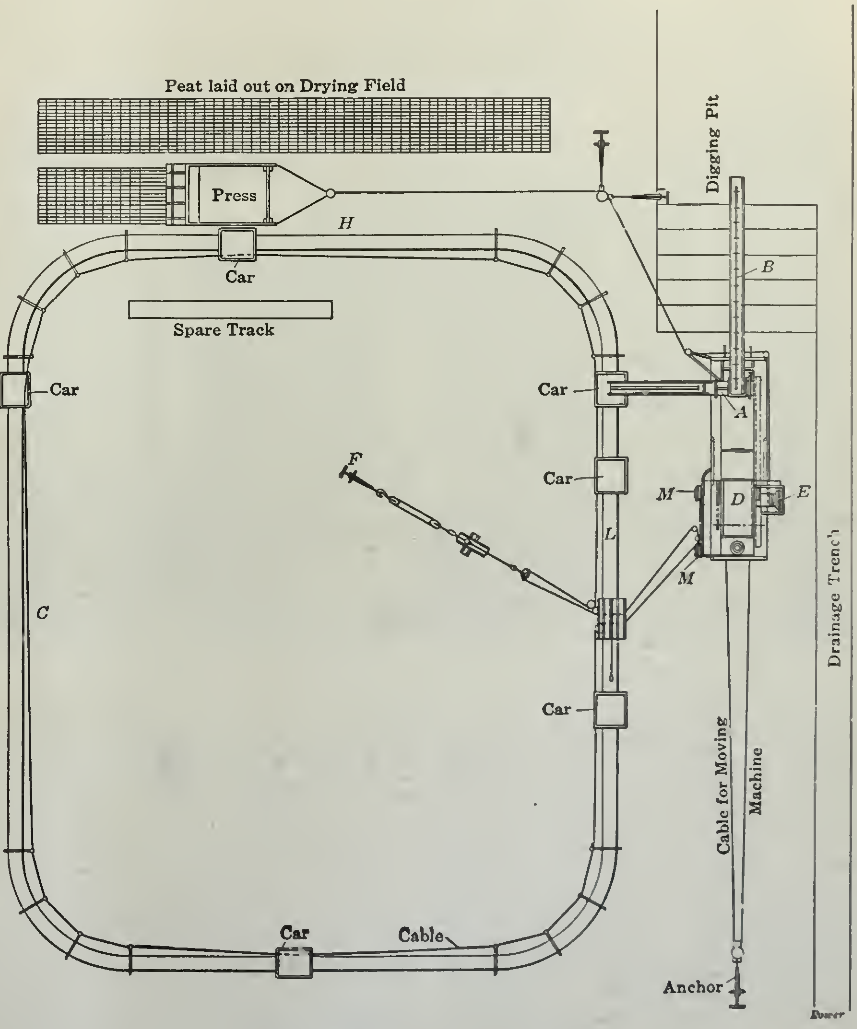


Fig. 5. Diagram of Peat Field

field. The engine and boiler are indicated at *D*, the mixing machine is at *A* and the elevator at *B*. The cable for hauling the press is operated by the drum *E* and the car cable by drums *M*. When it is necessary to move the tracks *H* to start a new strip of peat, a section is taken out of the sides *C* and *L*. The slack in the car cable is taken up by shifting the anchor and sheave *F* the required distance.

The peat is allowed to lie undisturbed for seven to ten days or longer, depending upon climatic conditions, until it is firm enough to be turned. After it is turned the peat is allowed to remain on the drying field for about ten days longer. It is then stacked. Each slab is spaced an inch or two away from its neighbors so that air may circulate through the stack and the drying process continue. The peat which is to be carried in stock for any length of time is removed from the stacks after about 30 days and placed in the storage shed which is shown in Fig. 4. The storage shed is of cheap, wood construction, 160 feet long, 22 feet wide and 18 feet high. The first small house beyond the shed is the blacksmith's shop; the second is the office.

Thirteen men and three boys are required to dig, machine and spread the peat. The men are paid $1.75 per day and the boys $1. This makes a total cost of $25.75 per day to lay the peat down on the drying field. At the rate of 30 tons per day the cost per ton is about 86 cents. The cost of turning the peat is 7 cents per ton and the cost of placing it in stacks is 25 cents. It costs 20 cents per ton to place the peat in the storage shed and 25 cents to load it from here into freight cars. This makes a total cost of $1.63 per ton for digging, spreading, handling and loading the peat for transportation. To this must be added the freight charges, the interest on the investment and the amortization. These items cannot be estimated from the data at present available. It is believed, however, that these will not cause the cost of the peat to become so great as to eliminate it from competition with coal at $7 or more per ton.

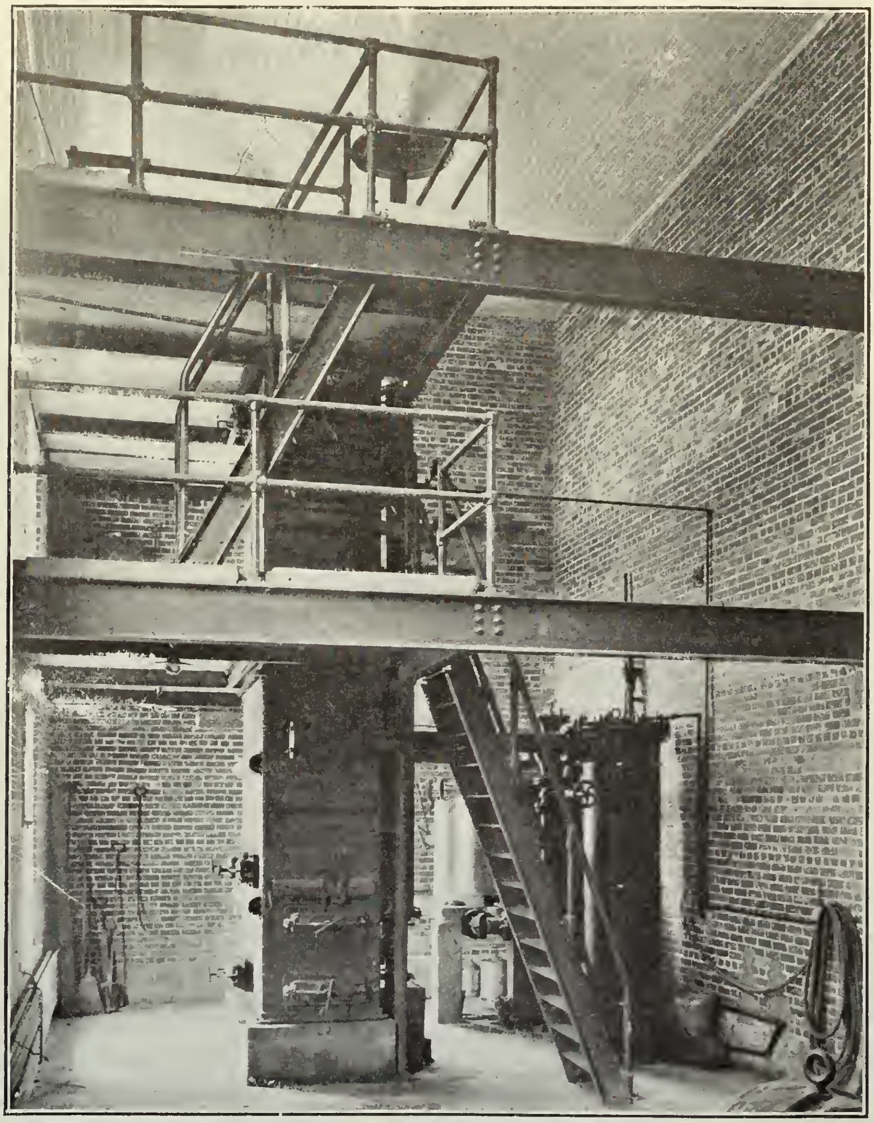


FIG. 6. THE KORTING GAS PRODUCER AND GAS-CLEANING APPARATUS

The Canadian Government will keep all of the peat that it requires for testing purposes and sell the rest at a suitable price.

FUEL-TESTING STATION

The Government fuel-testing station is located in Ottawa on Dolly Varden and Division streets. It was erected during the summer of 1909. At present the plant is equipped for testing peat fuel only. The apparatus consists of a Körting double-zone suction gas producer, the necessary purifying apparatus, a Körting gas engine and a 50-kilowatt Westinghouse direct-current generator. The producer room is large enough to hold additional apparatus and producers of other types are to be installed in the near future. These producers will be especially designed for bituminous coals and lignites. Part of the producer room and the Körting producer are shown in Fig. 6. The producer is 15 feet high, 2 feet 9 inches wide and 5 feet long.

The gas-cleansing apparatus is shown at the right of the producer. This consists of a wet tar separator and a coke scrubber.

The engine and generator are shown in Fig. 7. The engine is a single-cylinder, four-stroke-cycle machine, fitted with a throttling governor. The diameter of the cylinder is 15 inches, the stroke is 24 inches. The speed is 200 revolutions per minute. The air supply for the engine is obtained from the interior of the engine room through a brick duct built against the wall. This duct may be seen in Fig. 7 to the right of the engine. With this arrangement the temperature of the air supply remains practically constant throughout the year. During tests the current generated by the dynamo is absorbed by either the bank of five hundred 16-candlepower incandescent lamps, which is located on the end wall of the room, or by the iron grid which may be seen against the wall at the left of the view.

When the producer is started up, suction is furnished by a small centrifugal exhauster belted to an alternating-current motor. Current for the motor is obtained from the city-lighting company's mains. The motor and the exhauster may be seen at the extreme right of the view. The engine is started with compressed air.

The gas-testing apparatus is located on the right wall of the engine room. A detail view of this is given in Fig. 8. Gas is drawn from the system by a small Root blower and passed through a water-pressure regulator to a Junker gas meter. From the meter the gas goes to the calorimeter which is also a Junker instrument. The average effective heat value of the gas is about 126 B.t.u. per cubic foot. The producer requires from two and one-half to three pounds of peat per brake horsepower per hour. The plant has been in actual operation for such a short time that no specific data in regard to the cost of operation and the performance of the apparatus are available at this writing.

We acknowledge our thanks to Dr. Eugene Haanel, Canadian Director of Mines, for permission to obtain the material for this article. Much of the data relative to the peat-fuel manufacturing plant of Alfred was supplied by Aleph Anrep, peat expert for the Canadian government. Information relative to the fuel-testing station was secured through the courtesy of B. F. Haanel, chief engineer of the fuel-testing division of the Department of Mines.

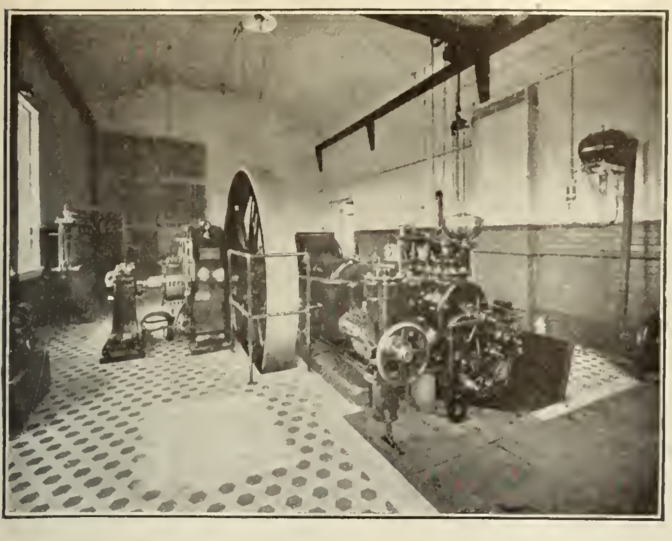


FIG. 7. A VIEW OF THE ENGINE ROOM

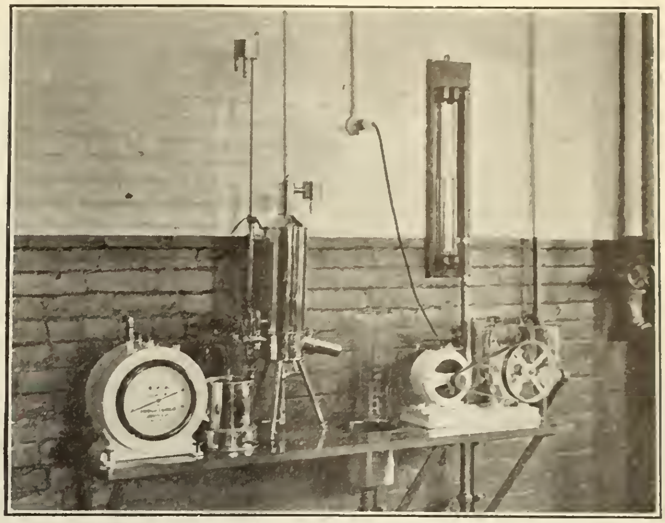


FIG. 8. GAS-TESTING APPARATUS

Notes on Economizer Practice

By George H. Gibson

A discussion of the relations between boiler and economizer heating surfaces and flue-gas temperatures, supplemented by economic data upon economizer practice with various operating conditions and different costs of coal.

Ordinarily 20 to 50 per cent. of the heat value of the fuel burned under a steam boiler escapes to the atmosphere in the form of sensible heat of the gases passing up the chimney. The percentage of the total heat of the fuel carried away by the flue gases at various temperatures, assuming that 26 pounds of air are used to burn a pound of coal containing 14,500 B.t.u., is shown in Table 1.

Each heat unit utilized in a boiler represents a certain cost in fuel and boiler upkeep. This affords a basis upon which can be estimated the value of the heat recovered by any heat-saving device, such as an economizer. The cost of obtaining heat in a boiler is made up of the following items:

Cost of fuel.
Cost of handling fuel and ashes.
Repairs and upkeep of the plant.
Fixed charges against the boiler.

The average first cost of the boiler and setting may be taken as $15 per boiler horsepower, and of the stoker, grate and draft apparatus as $5 per horsepower. The annual fixed charge is represented by the interest on this investment, plus the amount that must be laid aside yearly to provide for the indefinite continuance of the plant; that is, sinking fund and maintenance. Placing the interest at 5 per cent., and the sinking fund and maintenance charges at 12 per cent. of the first cost represents a total annual expense, exclusive of fuel, of $3.40 per boiler horsepower. The annual cost of fuel varies according to the amount of hours the plant is operated daily and to the number of days per year. The costs for incidental labor and power around the boiler plant are small, compared with the other factors, hence they may be assumed as constant without introducing any serious error.

With coal at $3 per ton, containing 14,500 heat units per pound, and the boiler operated at full load 10 hours per day and 310 days per year, at an average efficiency of 65 per cent., the annual cost for each boiler horsepower will be $16.28 for fuel and $3.40 for fixed charges, making a total of $19.68 for 3100 horsepower-hours. With coal at $3 per ton, and operating 24 hours per day and 365 days per year, the other factors remaining the same, the annual cost of 8760 boiler horsepower-hours becomes $49.40. Both figures, especially the latter, are below the actual costs incurred in practice as boilers are seldom operated with a load factor of 100 per cent. on all-day service, and it is, in general, necessary to have surplus boiler capacity in order to provide for cleaning and repairs; this would increase the fixed charges per boiler horsepower developed.

TABLE I. PERCENTAGE OF HEAT ESCAPING IH CHIMNEY GASES AT VARIOUS FLUE TEMPERATURES, ASSUMING 26 POUNDS OF AIR PER POUND OF COAL AND 14,500 B.t.u. PER POUND OF COAL.

Chimney Temperature, Degrees Fahrenheit.	B.t.u. in Gases per Pound of Coal.	Percentage of Total Heat in Gases.	Ratio of Heat in Flue Gases above 60 Degrees to Heat Absorbed by Boiler, Assuming 15 per Cent. Loss Through Grates and Radiation, per cent.
200	905	6.25	8
300	1555	10.7	14.4
400	2200	15.2	21.8
500	2850	19.68	30.2
600	3400	23.5	38.3
700	4150	28.6	50.6
800	4790	33.0	63.4
900	5440	37.5	79.0
1000	6090	42.0	97.5

The transmission of heat by convection from hot gases to iron surfaces in contact with water has been carefully investigated, and although affected by such factors as the velocity of the gases and the condition of the surfaces, it may be stated that for such temperatures as are found at the stack end of a boiler and in an economizer, the coefficient of transmission varies from 2½ to 4½ B.t.u. per square feet of surface per hour, per degree difference of temperature between the hot gases and the water.

TABLE 2. MINIMUM ECONOMIC TEMPERATURE OF GASES LEAVING THE BOILER, OPERATING 3100 HOURS PER YEAR.

Price of Coal per Ton.	Cost of Year.	Critical Temperature Difference, Degrees Fahrenheit.	Economical temperature of gases leaving boiler, pressure = 150 pounds gage. Degrees Fahrenheit.
$2.00	$14.25	319	685
2.50	16.97	268	634
3.00	19.68	231	597
3.50	22.40	202	568
4.00	25.10	181	547
4.50	27.80	163	529
5.00	30.50	149	515

OPERATING 8760 HOURS PER YEAR.

Price of Coal per Ton.	Cost of Brake Horsepower, Year.	Critical Temperature Difference, Degrees Fahrenheit.	Minimum economical temperature of gases. Boiler pressure = 150 pounds gage. Degrees Fahrenbeit.
$2.00	$34.10	133	499
2.50	41.70	109	475
3.00	49.40	92	458
3.50	57.10	80	446
4.00	64.70	70	436
4.50	72.40	63	429
5.00	80.00	57	423

One boiler horsepower-hour is equivalent to 33,300 B.t.u. Taking the lower value 2½ B.t.u. as the coefficient of transmission and letting D represent the difference in temperature between the gases and the water at the given point, each square foot of additional heating surface would recover

$$\frac{D \times 2\frac{1}{2}}{33,300} = 0.000075\,D$$

horsepower. Assuming that the boilers are rated on the basis of 10 square feet of heating surface per horsepower, the annual cost of maintenance for a square foot of heating surface would be $0.34; and the annual cost of a boiler horsepower recovered by the additional surface would be

$$\frac{0.34}{0.000075\,D} = \frac{4530}{D} \text{ dollars}$$

Equating this to the original cost of a boiler horsepower-year of 310 days at 10 hours each, namely, $19.68, D is found to be 231 degrees Fahrenheit; that is, if the difference in temperature between the gases and the water is more than 231 degrees Fahrenheit, the additional surface is more than paying for itself by the heat that it is recovering from the gases; if the difference in temperature is less, the additional surface is not paying for itself. This determines at once how far boiler surface can be extended profitably. Supposing the steam pressure to be 150 pounds gage, corresponding to a temperature throughout the boiler of 366 degrees Fahrenheit, the lowest temperature to which it will pay to reduce the gases by means of boiler surface is

$$366 + 231 = 597$$

degrees Fahrenheit.

In a similar manner the temperature to which it pays to reduce flue temperature may be determined. The lowest temperature to which gases should be reduced by boiler surface for various prices of coal are shown in Table 2.

The same theory can be applied to an economizer fed with cold water. Assume that the economizer, installed, costs $1 per square foot, and that the annual charges for interest, depreciation, maintenance, power and attendance amount to 12 per cent.; from which the lowest economical temperature difference with coal costing $3 per ton and the plant running at full load 24 hours per day and 365 days per year, is found to be 32.4 degrees Fahrenheit. With the cold end of the economizer receiving water at 60 degrees Fahrenheit the final flue-gas temperature could be brought as low as 92.4 degrees Fahrenheit. In Table 3 the values of D, the economical temperature difference, for the economizer, are given for different prices of coal ranging from $2 to $5 per ton, it being assumed that the coal is high grade and contains 14,500 B.t.u. per pound.

TABLE 3. FINAL TEMPERATURE OF FLUE GASES FOR DIFFERENT PRICES OF COAL FOR AN ECONOMIZER PLANT OPERATING 10 HOURS PER DAY AND 310 DAYS PER YEAR.

Price of Coal.	Cost of Brake Horsepower, Year.	Critical Temperature D, Degrees Fah.	Temperature Flue Gases Leaving Economizer for Various Temperatures of Feed Water, Degrees.			
			60 Deg.	100 Deg.	150 Deg.	210 Deg.
$2.00	$14.25	112.2	172.2	212.2	262.2	322.2
2.50	16.98	94.5	154.5	194.5	244.5	304.5
3.00	19.68	81.4	141.4	181.4	231.4	291.4
3.50	22.40	71.5	131.5	171.5	221.5	281.5
4.00	25.10	63.8	123.8	163.8	213.8	273.8
4.50	27.80	57.5	117.5	157.5	207.5	267.5
5.00	30.55	52.5	112.5	152.5	202.5	262.5

FOR A PLANT OPERATING 24 HOURS PER DAY AND 365 DAYS PER YEAR.

$2.00	$34.10	47.0	107.0	147.0	197.0	257.0
2.50	41.75	38.3	98.3	138.3	188.3	248.3
3.00	49.40	32.4	92.4	132.4	182.4	242.4
3.50	57.10	28.0	88.0	128.0	178.0	238.0
4.00	64.70	24.7	84.7	124.7	174.7	234.7
4.50	72.40	22.10	82.1	122.1	172.1	232.1
5.00	80.00	20.0	80.0	120.0	170.0	230.0

The value of the least profitable temperature difference can be expressed algebraically as follows:

$$D = \frac{E}{U \times Y_e \left[\frac{B \times S}{33,300 \times Y_b} + \frac{C}{2000 \times N_s \times H} \right]}$$

B = Annual charge in dollars on 1 square foot of boiler surface, including interest, maintenance, depreciation and labor,

E = Annual charge in dollars on 1 square foot of economizer surface, including interest, maintenance, depreciation and labor,

Y_b = Hours per year that the boilers are operated,

Y_e = Hours per year that the economizers are operated,

C = Cost of coal per ton in dollars, including handling of the coal and ashes,

H = Heating value of the coal in B.t.u. per pound,

S = Square feet of boiler-heating surface per boiler horsepower developed,

N_s = Efficiency of the boiler when operated at S square feet per boiler horsepower developed,

U = Coefficient of transmission per square foot per hour per degree difference of temperature between the gases and the water.

In applying these figures practical considerations should not be forgotten; that is, it will not pay to reduce the flue-gas temperature below 250 degrees Fahrenheit, as the resulting condensation and deposits of the water vapor upon the tubes, together with the sulphurous gases, will produce rapid corrosion.

Operation of the Steam Loop

By F. L. Johnson

A simple explanation of the principles upon which the action of the steam loop depends, showing how steam may be made to circulate rapidly in a closed system of piping and by this circulation return all condensation to the boiler from which the steam comes.

If a closed vessel or receiver be placed above a boiler and connected to it by two pipes, as shown in Fig. 1, it is evident that the pressure in the receiver will be the same as that in the boiler and the water level in the pipe leading from the water space in the boiler will coincide with the water level in the boiler.

If the valve on the outlet of the receiver be opened, the pressure in the receiver will be reduced by the outflow of steam and there will be a flow of steam from the boiler to the receiver. The reduction of pressure in the receiver will extend downward decreasingly in the pipes toward the boiler. In one of the pipes the water will rise to a hight depending upon the difference in the pressure acting on the surface of the water in the boiler and that acting on the surface of the water in the pipe leading to the receiver.

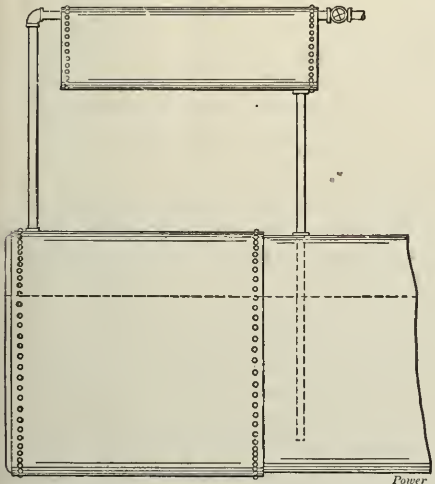


FIG. 1. SHOWING PRINCIPLE OF OPERATION

Through the other pipe steam will rise in an effort to make up the same difference in pressure, but the steam being so much lighter than the water, it will flow to the receiver while the water will only rise in the pipe until it reaches a hight where the weight of that portion above the boiler water level will be equivalent to the difference in pressure betweeen the boiler and the receiver.

So long as the pressure in the receiver is below that in the boiler the flow of steam to the receiver will continue and the water in the other pipe will stand at a hight above the water level in the boiler that will balance the difference in pressures in the boiler and in the receiver.

If the steam which flows to the receiver is wet or holds water in suspension, this water will be swept along in the current of steam until it reaches the receiver, where, owing to the greater volume of the receiver and the consequent slower travel toward the opening, the water will fall to the bottom of the receiver and run into the pipe connected to the water space in the boiler, adding its volume to the column of water already in the pipe, because there is no upward current of steam in this pipe to oppose the downward flow of water. If instead of the pressure in the receiver being reduced by the escape of steam from the outlet it is reduced only by the condensation which takes place, the same flow of steam through one pipe and rise of water in the other will take place, only in a lesser degree because the pressure difference is smaller.

From the action of the steam and water in the boiler and in the receiver, the principles upon which the operation of the steam loop is based may be readily deduced. They are as follows: A difference of pressure may be balanced by a water column; liquids and vapors tend to flow to the region of least pressure; rate of flow is proportional to mass and difference of pressure between two points; decrease of pressure in a steam pipe or vessel is proportional to the condensation; a current of steam will carry water with it by friction.

Like the injector, the steam loop is sometimes called a paradox, while as a matter of fact its operation and the results produced are just what should be expected from the conditions under which it is installed and operated. It is a system of piping so arranged that water of condensation collected by separators or otherwise is returned to the boiler whence it came without the interposition of traps or pumps. In its ordinary form it is simply a line of pipe consisting of three parts or sections which for convenient reference are called the riser, the horizontal and the drop.

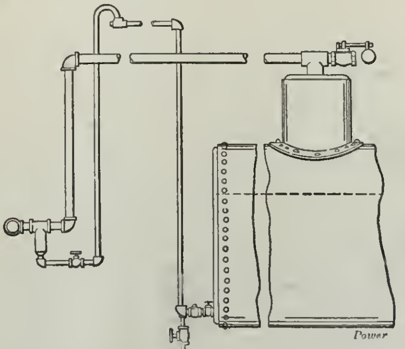


FIG. 2. MAIN PIPE AT BOILER LEVEL

Where the loop is used to return entrainment and condensation from the steam pipe of an engine, a separator is generally placed at or near the throttle valve and the "riser" is connected to the drain pipe of the separator. The riser is carried vertically to a hight depending on the steam pressure carried and the rate of flow desired, and is connected to the "horizontal," which, notwithstanding its name, pitches toward the "drop," into which it discharges. The "drop" leads to the boiler below the water line and the water returns through it to the boiler by gravity as fast as its head permits.

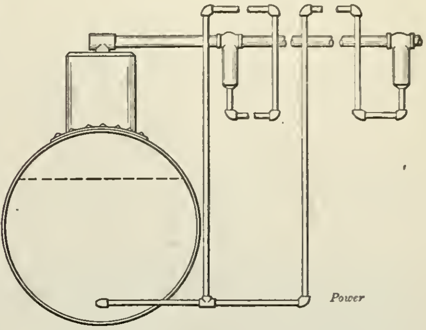


FIG. 3. DOUBLE LOOP ABOVE BOILER

If in a system there is a pressure drop of eight pounds from the boiler to the engine, this reduction of pressure will extend through the riser and the horizontal to the "drop," where the difference between the pressure in the horizontal and that in the boiler will be balanced by a water column approximately 20 feet higher than the water line in the boiler.

Now, if condensation in the horizontal reduces the pressure one pound more, the column of water will tend to rise 30 or more inches to balance the difference in pressure between the boiler and the horizontal. But the mixture of steam and spray in the riser will also travel toward this same point and being so much lighter, will travel faster, and as the entrained water reaches the horizontal it runs along to the drop and down to the column rising from the boiler, adding to its hight until it overbalances the difference in pressure, and water flows into the boiler from the bottom end of the drop.

It is the condensation in the horizontal that furnishes the motive power which operates the loop, and when this condensation is maintained at a rate sufficient to give the necessary difference in pressure, the hight of the column in the drop will be that corresponding to the difference in pressure and no greater.

The limits of pressure within which the loop will operate are wide, for the principles of operation are applicable equally to great as well as small differences in pressure and large as well as small quantities of water are readily moved.

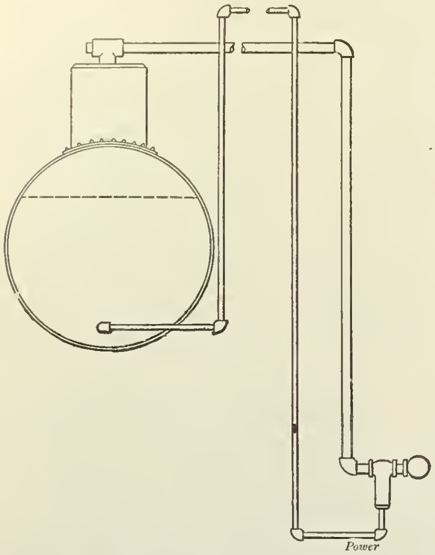


Fig. 4. Main below Boiler Level

Water may be returned from a level far below that of the boiler, and the action will continue through long pipes, whether run overhead or laid underground. The use to which the steam is put after the separator has removed the water and the riser conveyed it away has no effect upon the operation of the loop itself, and it may be used to return condensation from all sources, such as heating and drying systems, steam jackets, boiling kettles, etc.

Lack of head room may limit the application of the loop where there is a great difference between the boiler pressure and that at the point of delivery. If the pressure in a return is ten pounds below that in the boiler and a loop 30 feet high is necessary, it would take a loop 150 feet high to return the condensation where there is a difference of 50 pounds.

While the absolute pressure of the steam has nothing whatever to do with the operation of the loop (it will work equally well under low or high pressure), it might fail where the difference in pressure is great as compared with the lowest pressure in the system.

Suppose the boiler pressure is 25 pounds and in the return there is a pressure of only one pound; if the loop is 100 feet high and the condensation in the horizontal is sufficient to produce a vacuum of 14.7 pounds, the water in the drop will rise to about 80 feet, and it is doubtful if the one-pound pressure at the foot of the riser, aided by the vacuum, will force the mixed contents of the riser to the hight of 100 feet. It is certain that it will not if a large percentage of water is present.

Practically, a pressure difference of 15 pounds with ordinary boiler pressure is the condition under which the loop is expected to operate.

As the pressure difference in the system is balanced by the column of water in the drop it makes no difference how far below the boiler the riser starts so long as the pressure at the bottom of the riser is sufficient to lift the mixture of steam and water to the hight of the horizontal.

Fig. 2 is a diagrammatic sketch of the loop as commonly installed for taking care of the condensation and entrainment in a simple installation consisting of a single engine and one or more boilers, with the engine at or near the level of the boiler water line. Fig. 3 represents an installation where the engine is considerably below the boilers.

Two or more separators may be attached to risers which discharge into the same horizontal or separate drops may be connected to the same return pipe, as shown in Figs. 4 and 5.

Fig. 6 illustrates a method of double connection to radiators, drying coils and kettles, by which only dry steam is turned into the system and the condensation goes to the boilers through a second loop.

It may be readily seen from the foregoing description that the steam loop is a simple arrangement of piping which when properly proportioned and connected performs the duties of receiver and return pump. And that once working should continue indefinitely without attention. This it would do if only steam and water were handled, but as all boiler water contains some air which is set free by heat and as not all of it goes through the engine, some will accumulate in the horizontal pipe of the loop and being noncondensible will, if not drawn out of the system, stop its operation.

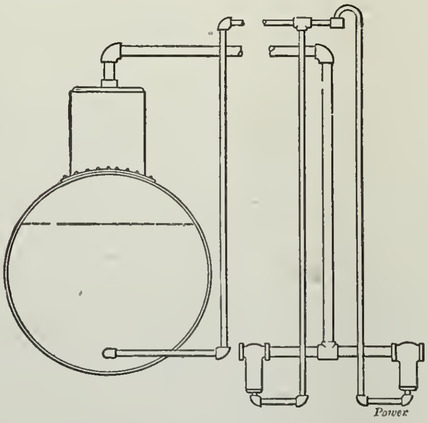


Fig. 5. Double Loop below Boiler

Air is a common source of trouble and the probable cause of all failures in operation, so that provision should be made by which the air may be blown out at any time. Sometimes an air cock is placed on the highest point of the piping and sometimes a small pipe is led from the horizontal pipe to a condenser in the plant.

But if the air is removed from the system at frequent intervals by a hand-operated cock or continuously by an open pipe or one leading to a condenser the loop will operate as long as steam for condensation in the horizontal pipe is furnished by the boiler.

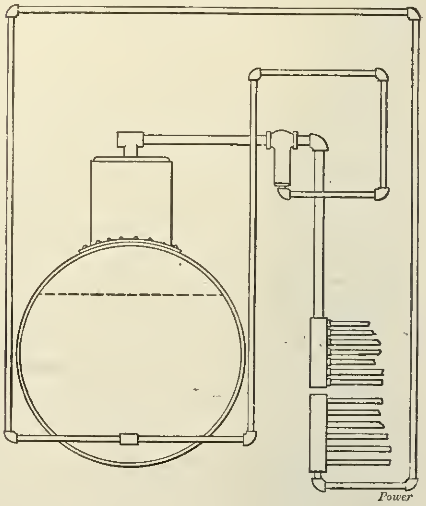


Fig. 6. Two Loops with Dry Kiln

It is reported that a second power station of the Shawinigan Water and Power Company will be situated near the present generating plant at Shawinigan falls, on the St. Maurice river, about 85 miles from Montreal, and will be designed for a capacity of 75,000 horsepower, as against 55,000 horsepower in the case of the existing station. It is stated that the company's transmission lines, which are of aluminum carried on wooden poles, are at present the most extensive in Canada, exceeding a total of 400 miles. For transmission to Montreal the current, which is generated at 2200 volts, is transformed up to 50,000 volts; 3000 horsepower is supplied to the Montreal street railways, and 15,000 horsepower to the Montreal Light, Heat and Power Company. It is of interest to mention that the Shawinigan company supply 25,000 horsepower of water power—undeveloped—to the Northern Aluminium Company, and 14,000 horsepower of water power to the Belgo-Canadian Pulp and Paper Company. The greater part of the above power is obtained at a second dam about a mile up the river.

Do not close the damper entirely when there is fire on the grates, as gas may collect and result in an explosion as destructive, as if caused by steam.

A Comparison of Efficiencies

By G. A. Glick

The various types of machines require different methods of computing their efficiencies. The economy of a water wheel and that of a steam engine are figured on a different basis. The author explains how the various efficiencies are obtained.

One hears sometimes that the efficiency of a waterwheel is 80 per cent. and the efficiency of a steam engine is 10 per cent. The person who hears these per cents. may not have them given to him as a comparison of the poor efficiency of the steam engine and the high efficiency of the waterwheel, but if he is not acquainted with the nature of these efficiencies such might be his impression. Such is not the case. Each efficiency is computed on a different basis.

Definition of Efficiency

Efficiency is the ratio, usually expressed as a per cent., of the energy utilized to the energy applied in any system, machine or combination of machines. A general expression for efficiency is represented by the equation

$$\text{Efficiency} = \frac{E - E'}{E}$$

where E equals energy in the working substance as it enters the machine, and E' equals energy in the working substance as it leaves the machine.

The efficiency as defined above is not for a practical machine, however, and the equation gives the efficiency that is the highest that can be hoped for under the conditions. Some energy is utilized in overcoming losses in the machine itself, and a more practical definition of the term is as follows: Efficiency is the ratio of the energy obtainable from a machine to the actual energy supplied and is represented by the equation

$$\text{Efficiency} = \frac{E - L}{E}$$

in which E equals the energy applied to the machine, or the energy in the working substance as it enters the machine, and L equals the term that includes all of the losses.

Mechanical Efficiency

The term mechanical efficiency is applied to a machine to indicate the loss in that particular machine due to mechanical losses as friction. This efficiency is really a simple one and is expressed by this equation

$$\text{Mechanical efficiency} = \frac{\text{Input} - \text{friction loss}}{\text{Input}}$$

In a steam engine the brake horsepower divided by the indicated horsepower, expressed as a per cent., gives the mechanical efficiency. The difference between the two is taken as the friction loss. This friction loss is assumed to be constant although experiments have shown that it varies with the load. In determining the mechanical efficiency of any other machine some means must be provided to drive the machine idle, and measure the input at the same time. This input is taken as the friction loss and when taken as a constant it is easy to see that the larger the load on the machine the higher will be the mechanical efficiency.

Efficiency of Heat Engines

Carnot's cycle is one of a purely theoretical nature and is of great use in the study of the theory of heat. It is the cycle of an ideal engine and mathematics will show that the efficiency as calculated by Carnot is the limit toward which the practical heat engines may tend. The efficiency as given by this cycle is expressed by the following equation.

$$\text{Ideal or Carnot's efficiency} = \frac{T - T'}{T}$$

in which T equals the upper limit of absolute-temperature range and T' equals the lower limit of absolute-temperature range. Degrees Fahrenheit are changed to degrees absolute by adding 461.

To apply the above efficiency equation assume a perfect heat engine to be operating on Carnot's cycle and receiving steam at a pressure of 200.3 pounds, gage, which it exhausts at atmospheric pressure. The absolute temperature due to 200.3 pounds gage pressure is

$$387.9 + 461 = 848.9.$$

The absolute temperature of the steam at atmospheric pressure is

$$212 + 461 = 673.$$

Substituting these values in the equation, we have

$$\text{Ideal efficiency} = \frac{848.9 - 673}{848.9} = 20.7$$

per cent. which is the highest that can be hoped for under these conditions.

The high efficiency of the gas engine is due to the high explosion temperature and relatively low exhaust temperature.

This efficiency of Carnot's is known by several names, such as Carnot's Efficiency, Ideal Efficiency and Thermodynamic Efficiency. Carnot's cycle is a reversible one; that is, heat may be changed to work or work may be changed to heat. The heat motors of today are not reversible engines so the application of this efficiency to the practical heat engines merely indicates the upper limit of efficiency which the designer may hope to approach.

Thermal Efficiency

The efficiency of the heat motor is expressed as a thermal efficiency. This efficiency is the ratio of the useful work done to the work supplied. If the work of the machine and the heat required to do this work are known, the efficiency is

$$\text{Thermal efficiency} = \frac{\text{Work}}{778\ (\text{B.t.u.})}$$

in which *Work* equals the foot-pounds of energy given up by the machine, *B.t.u.* equals the total heat units supplied to the machine, and 778 equals the constant to change B.t.u. to foot-pounds. The above quantities can be calculated from brake tests applied to a machine and the heat contents of the steam per pound. When the total pounds of water used by the engine is known, the total heat used can be calculated.

This efficiency can also be expressed as follows:

$$\text{Thermal efficiency} = \frac{2545}{\left(\begin{array}{c}\text{Pounds steam per H.P.-hr.} \times \text{heat content}\\ \text{per pound}\end{array}\right)}$$

Expressed in words this is 2545 B.t.u., the work equivalent of a horsepower-hour, divided by the pounds of steam per horsepower-hour, and also divided by the heat content of one pound of steam equals thermal efficiency.

This efficiency is very low because the heat content of steam is computed from a point known as absolute zero, which is 461 degrees below Fahrenheit zero. A great deal of the heat content of the steam is not available in the engine. It is just as if a waterfall 100 feet high, but 1000 feet above sea level, had its efficiency computed with regard to sea level as the basis for energy content. Because of this, the logical efficiency applied to heat motors is an efficiency ratio described in the following paragraph.

Efficiency Ratio

The efficiency ratio is the ratio of the B.t.u. available to the B.t.u. utilized. That is, it is the ratio of the heat equivalent of the work done and the heat required to do that work.

$$\text{Efficiency ratio} = \frac{\text{Heat equivalent of work done}}{\text{Heat utilized}}$$

Expressed in other words this efficiency ratio is really the ratio between the thermal efficiency and the ideal efficiency.

Efficiency of a Refrigerating Machine

It may be surprising to some to hear of an efficiency greater than 100 per cent., but such is the case with a refrigerating machine. This machine works on a reversed cycle and its efficiency may be expressed as follows. This equation is the same as that of the thermal efficiency of a heat motor except that it is reversed.

$$\text{Thermal efficiency} = \frac{778\ (B.t.u.\ supplied)}{Work}$$

Now the heat taken up by a refrigerating machine may be several times the heat equivalent of the work done in compressing the working substance, so the efficiency may be several times 100 per cent.

Efficiency of a Waterwheel

The efficiency of the waterwheel is really an efficiency ratio, but strictly speaking it should not be called such. The efficiency ratio of the steam engine and heat motors gives a fair way of comparing the different motors, but with a waterwheel its location at different hights above sea level makes any efficiency corresponding to the thermal efficiency out of the question. The efficiency below gives a fair means of comparison for all waterwheels. Theoretically 8.8 cubic feet of water per second over one-foot head will develop one horsepower. The expression for waterwheel efficiency is as follows:

$$\text{Efficiency} = \frac{P \times 8.8}{q \times head}$$

in which P equals the horsepower developed by the wheel; q equals the water taken by the wheel in cubic feet per second, the *head* equals head, or fall, in feet.

Efficiency of Dynamo

Electrical efficiency, in general, is also represented by the ratio of output to input. However, the equations for a generator and a motor are as follows:

$$\text{Efficiency of generator} = \frac{\text{Electrical output}}{\text{Electrical output} + \text{losses}}$$

$$\text{Efficiency of motor} = \frac{\text{Mechanical output}}{\text{Mechanical output} + \text{losses}}$$

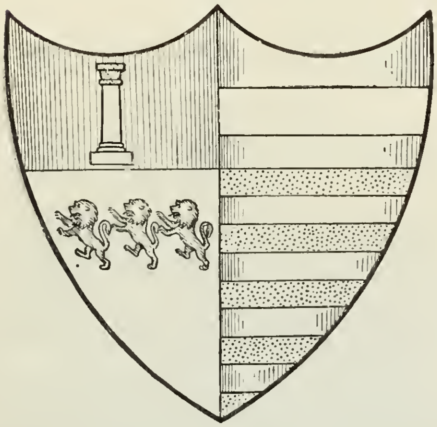
or

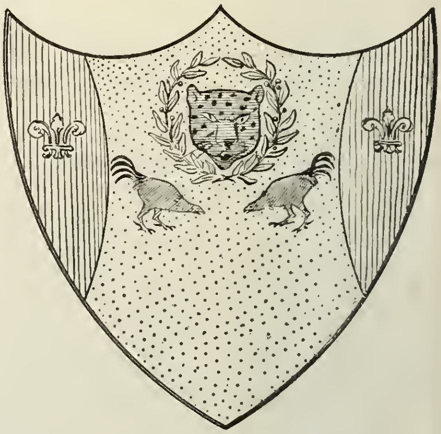
$$\text{Efficiency of motor} = \frac{\text{Electrical input} - \text{losses}}{\text{Electrical input}}$$

The losses in electrical machines of these classes are: Copper loss or heating loss in armature and fields; iron losses or hysteresis and eddy currents in armature core, and mechanical losses or bearing friction, brush friction and air resistance to rotating armature.

The John Cockerill Works at Seraing

By F. R. Low





When the use of blast-furnace gas in engines is mentioned one naturally thinks of the great works at Seraing, near Liège, Belgium, which bear the name of John Cockerill. It is generally known that the first large work of this kind was done at Seraing, but how comes it that this large Belgian establishment bears a name so conspicuously English and what has been its history?

John Cockerill was an English emigrant who attracted attention when the mechanical industries which have meant so much for Belgium were in their beginning. He began by manufacturing textile machinery and received in 1817 from the king of Holland for the price of $9000 the château of Seraing and its dependencies with the commission to install there new works and industries. In the works so established the steam engine was applied to the working of iron to an extent before unheard of. The construction of steamboats was undertaken and John Ericsson was here from 1825 to 1827, working out a steamer which must have been provided with high-pressure boilers with internal furnaces, forced draft and with engines having surface condensers, innovations too daring for the epoch. In 1835 the Cockerill works turned out the first locomotive and the first railroad rails used on the continent of Europe. In 1827 they made a centrifugal pump for the Charbonnages des Artistes, at Jemeppe, which machine, entirely forgotten, was still working in 1870. In 1863 they introduced the bessemer process upon the Continent.

The first engine to run upon blast-furnace gas was a small trial machine of eight horsepower, the patents of which the Cockerills purchased from Delamare Deboutteville in 1895. Two engines of 250 horsepower each were then built, and are now running in one of the several power stations or "centrals," as they call them, of the works. At the Paris Exposition of 1900, they exhibited an engine capable of developing a thousand horsepower. At the time, its capacity was given as 1000 horsepower with city gas and 700 with blast-furnace gas. Development has shown that the richness of the gas does not make as much difference as was formerly supposed.

Having in mind some recent experiences in the States, I asked the engineer who was conducting me about, how much trouble they had with cylinders breaking. He denied that they had any, and called my attention to their method of cylinder construction. The cylinder and jacket are cast in one piece, no bushing being used, and the metal in the barrel is not over 2 inches in thickness for the largest size, but they are tied lengthwise by enormous bolts passing through the jacket space and tying the back head to the frame in such a way that all the stress is taken by the bolts. They had been free, he said, from piston-rod breakage, which he considered more a question of material than design. The Cockerills have now commenced to build Diesel engines.

The entrance to the works is through the grand court of the old palace, which is maintained as a beautiful garden in the center of which is a fountain, around the base of which are four heroic figures of workmen, said to be portrait sculptures of workmen who emigrated to Belgium with Cockerill.

After the death of Cockerill in 1840 his establishments met with several reverses, but in 1842 they were incorporated with Gustave Pastor, nephew of Cockerill, at the head as the Societé John Cockerill, and with a capital of $2,500,000. In 1872 the capital was increased to $3,000,000 and in 1895 cut down to $1,500,000 by the reimbursement to the stockholders of the half of their value. In 1904 it was again raised to $2,500,000. The "Société" now owns its own coal and iron mines and had spent, June 30, 1909, over $20,000,000 for increases and betterments without calling upon its stockholders. The establishments employ between 10,000 and 11,000 hands, the average pay between 1902 and 1909 being about $250 per year.

The Efficiency of Compressed Air

By Snowden B. Redfield

> *"What is the ratio of the work represented in the compressed air to that done by the compressor?" This is a question often asked, and in the present article is answered by the author.*

It is fairly well known that the energy transformation in the use of compressed air is not high but the fact that it "does things," some of them better than by any other means and some which are not done by any other means at all, makes the matter of fuel economy of secondary importance. No apology is needed for the use of compressed air today.

That this matter of the fuel economy is not well understood is evidenced by the repeated appearance of propositions to develop the power of a water fall in the form of compressed air and transmit this through pipes to some distant point where its energy, like that of electricity, is supposed to be turned back again into work.

Such power-transmission propositions as this are utterly absurd because the losses in the compression and expansion of the air reduce the power efficiency to too low a point. These losses are not losses of pressure by friction, nor air by leakage, but simply losses due to heating and cooling.

It is not a difficult matter to prove that all of the work of compression goes into heat and is lost in pipe-line radiation. All of the work performed at the far end comes from the intrinsic heat originally in the air and the economy obtained depends entirely upon how the air is used. The actual cold compressed air traveling in the pipe does not at all represent the energy put into it by the compressor, as this has all been lost in radiation and the only answer to the question first expounded, is: "Nothing."

However, it is known that the air does work and this may be compared to the compressor work (although they have no direct relation) and in this way an "efficiency" may be stated.

As examples of machines using air with little or no expansion, rock drills and pneumatic tools may be cited and some interesting figures as to the efficiency of the power transformation, are given by the accompanying diagrams.

While actually indicating the drill or tool cylinder would be difficult, some rough idea of the indicated horsepower developed within it may be obtained by a little figuring, together with some judicious guessing.

Indicator diagrams of such machines would theoretically be rectangles, but wire drawing and cushioning effects of the valve mechanism would considerably modify this. It may be assumed then, reasoning from such a thing as a steam-pump cylinder, without cutoff, that the diagram factor will be about 80 per cent. In other words the actual mean effective pressure will be about 80 per cent. of what the theoretical rectangular diagram would give.

On this basis it is determined that a standard rock drill having a 3-inch diameter cylinder will develop about 6.2 indicated horsepower with 100 pounds at the throttle, this decreasing with the pressure supplied, down to about 3.7 indicated horsepower, with only 60 pounds pressure.

A 3-inch rock drill will require about 138 cubic feet of free air per minute with 100 pounds pressure at the throttle; this decreasing to 90 cubic feet, with only 60 pounds pressure.

Knowing the quantity of air and the pressure, the compressor horsepower is easily calculated.

Thus, allowing 10 pounds pressure drop in the pipe, a 3-inch rock drill will require 29.8 indicated horsepower in the steam cylinders of the compressor with 100 pounds pressure and single-stage compression, or 25.2 indicated horsepower with compound compression. These figures reduce as the pressure used is reduced, but this, of course, reduces the work done by the tool.

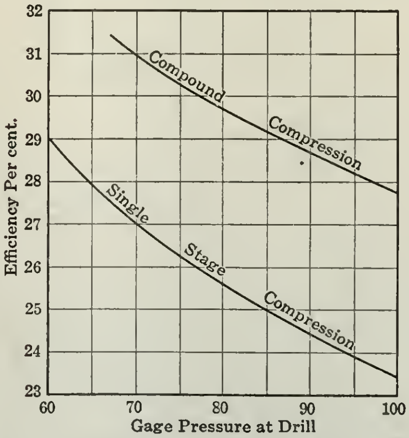


Fig. 1. Probable Efficiency Referred to Air End of Compressor

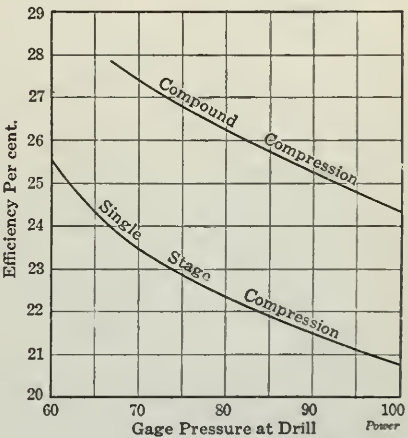


Fig. 2. Probable Efficiency Referred to Steam End of Compressor

Comparing the probable indicated horsepower developed inside the drill cylinder with the actual compressor power required to furnish the air, gives the probable efficiencies shown by the chart. These efficiencies are referred to both the air and steam cylinders of the compressor, so as to give a basis for calculations for various methods of driving the compressor. They include 10 pounds pressure drop in the pipe line.

Referred to the air end of the compressor, it may be seen that with single-stage compression and 100 pounds pressure, about 23.5 per cent. efficiency is obtained, increasing to about 29 per cent. with the low pressure of 60 pounds. Compound air compression brings these figures up to 27.8 per cent. with 100 pounds and 31 per cent. with 70 pounds.

Referred to the steam end, allowing 88 per cent. mechanical efficiency between the steam and air ends of the compressor, single-stage compression gives a little less than 21 per cent. efficiency with 100 pounds and about 25.5 per cent. with 60 pounds air pressure. Compounding the air cylinders of the compressor increases these figures to about 24.5 per cent. with 100 pounds and almost 27½ per cent. with 70 pounds air pressure.

While these figures for efficiency have been determined for rock drills in particular, they apply equally well to almost any machine using compressed air without expansion. It must, however, be remembered that the figures are based upon indicated horsepower only, both in the drill and the compressor. This is because of the impracticability of measuring the "brake horsepower" of the drill. If, however, brake horsepower efficiency is required, these figures for efficiencies of indicated horsepowers can be multiplied by the mechanical efficiency of the device using the air, say 90 per cent. or 80 per cent., as the case may be.

It is to be noted that the higher efficiencies are obtained with the lower pressures. This is because there is less loss by heating the air during compression and therefore it is advisable to use pressures as low as is consistent with the size and weight of the machine required to do a given amount of work.

Modern Releasing Valve Gears

By C. A. Tupper*

The concluding discussion upon the various representative types of releasing valve gear. The earlier developments leading up to present-day practice were discussed in the September 6 issue.

The first three gears shown in the following discussion should be credited to George Brown, who proceeded along somewhat different lines from those taken up in the first instalment of this article. This inventor is also responsible for the Brown-Harris gear shown in Fig. 29, which he originated when in the employ of the W. A. Harris Steam Engine Company.

In Fig. 17 is shown the Filer & Stowell gear, for which the principal claims are simplicity, accessibility and noiseless operation. Stress on the valve stem is relieved by the arrangement of the drop lever, the hub of which is so designed as to fit into a recess in the bonnet. The use of double ports in the cylinder reduces the throw of the wristplates and as a whole the gear is well designed for high speeds.

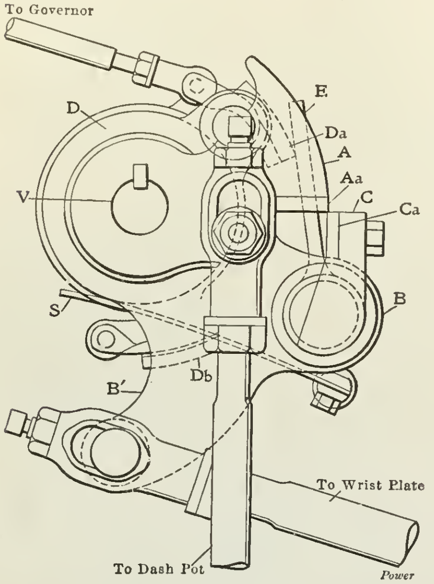


FIG. 17. FILER & STOWELL GEAR

A = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on *A*.
B B' = Bell crank which carries the hook *C* on one end of a short shaft journaled in its smaller arm; the other end of the shaft carries a tripping lever *E*. The bell crank receives its motion from the wristplate.
C = Hook.
Ca = Block carried by *C*.
D = Knock-off cam mounted on the valve stem, the position of this cam being determined by the governor.
Da = Projection for tripping.
Db = Projection for safety stop.
E = Trip lever.
S = Spring.
V = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *E* comes in contact with *Da*. This forces *E* and *C* away, and releases *A*, which is immediately drawn downward by the dashpot and closes the valve.

Fig. 18 is a sketch of the Wisconsin Engine Company's gear, following lines almost identical to those of Fig. 17, with, however, important features of its own. This was fully described in the October 27, 1908, issue of POWER, an abstract of which, for convenience in making comparisons, is given as follows:

In the design of the gear all parts subjected to the greatest strain have been placed as near the cylinder as possible, in order to get the benefit of rigid support and to reduce the leverage of the gear on the bonnet. The drop lever is inside the bonnet, rising and falling through the inside opening in the casting. Besides bringing the valve gear nearer to the cylinder, this construction, which has also been used by Allis-Chalmers Company and other builders in the design of engines for heavy service, offers the advantage of giving a substantial bearing to the valve stem on each side of the point where the turning movement is applied. The steam lever has a long bearing on the bonnet just outside of the drop lever, and carries the trip pin and trip steels by which the drop lever is engaged and the steam valve opened. An annoyance sometimes met with in Corliss engine operation is the wearing away or sudden breaking off of the edges of these steels and the consequent refusal of the drop lever to pick up and open the steam valve; this is obviated in the Wisconsin Engine Company's gear by an arrangement for controlling the lap of these steels while the engine is in motion. The trip pin contains a V-shaped recess, one side of which is held against the adjusting screw by a flat steel spring. By changing the set of this screw the relative positions

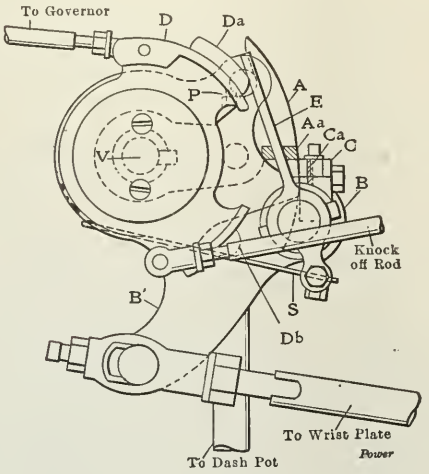


FIG. 18. WISCONSIN ENGINE COMPANY'S GEAR

A = Drop lever or crank arm, keyed to the valve stem inside the bonnet opening and connected to the dashpot rod. This has but one-half the ordinary leverage on the bonnet.
Aa = Catchblock on *A*.
B B' = Bell crank or steam lever, which carries the trip pin *C* on one end of a short shaft journaled in its smaller arm; the other end of the shaft carries a tripping lever *E*. The bell crank receives its motion from the wristplate.
C = Trip pin (behind nut and boss).
Ca = Block carried by hook.
D = Knock-off cam loosely mounted on the bonnet collar, the position of this cam being determined by the governor.
Da = Projection for tripping.
Db = Projection for safety stop (safety cam).
E = Trip lever.
P = Closing pin by which the drop lever is forced down should the dashpot fail to work.
S = Spring, adjustable while gear is in motion.
V = Valve stem supported on both sides of actuating point.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *E* comes in contact with *Da*. This forces *E* away, thus forcing *C* outward and releasing *A*, which is immediately drawn downward by the dashpot and closes the valve.

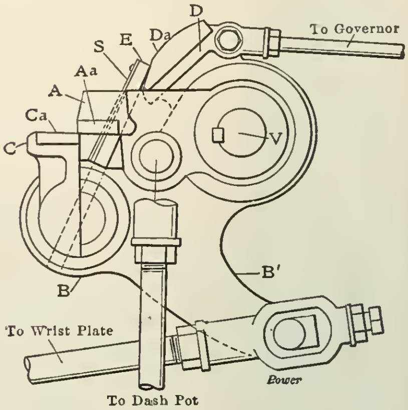


FIG. 19. STANDARD NORDBERG GEAR

A = Drop lever or crank arm, keyed on outer end of valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on *A*.
B B' = Curved double arm loosely mounted on valve stem and receiving its motion from the wristplate; *B* carries hook *C*.
C = Hook in form of L-shaped lever carrying die block at upper end.
Ca = Block carried by hook.
D = Knock-off cam, the position of which is determined by the governor.
Da = Projection for tripping.
E = Trip lever or arm.
S = Spring.
V = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *C* engages *A*, thereby raising *A* until *E* is forced outward by the projection *Da*, carrying with it the hook *C* and releasing *A*, which is immediately pulled down by the dashpot and closes the valve.

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of the two steels are varied, so that their laps may be increased when required.

Referring to Fig. 18, the knock-off lever, through which the governor controls the cutoff, is placed at the outer end of the bonnet, where it is practically free from disturbance by the other working parts. There is but one light rod connecting the governor lever with the knock-off lever on the bonnet at the crank end of the cylinder and a light rod connecting the crank-end knock-off lever with that on the head-end bonnet. This not only dispenses with the long rod and its lever between the governor and the head-end bonnet but greatly reduces the weight and consequently the inertia of the parts, which must be moved by the governor to control the cutoff. The governor and gear consequently work in harmony to secure close regulation. In the standard high-speed design no wristplate is used on the steam valves, the motion being transmitted through straight rods direct from the eccentric. Other points are included in the general list of features common to nearly all modern releasing gears, as mentioned near the end of this article.

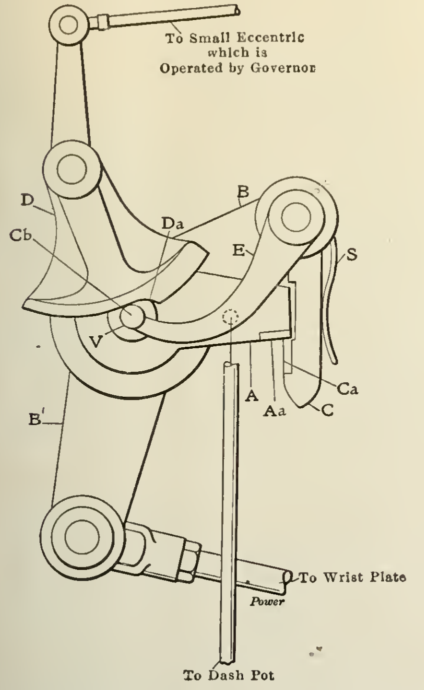

FIG. 20. SEDERHOLM LONG RANGE CUTOFF GEAR

A = Drop lever or valve-closing arm, keyed to the valve stem which it actuates, and connected to dashpot rod.
Aa = Catchblock on A.
$B\ B'$ = Two-arm lever B carrying C and E, and receiving its motion from the wristplate.
C = Hook journaled in B.
Ca = Catchblock on C.
Cb = Trip roller on lever E.
D = Knockoff cam, pivoted on the bonnet and positively operated by means of an eccentric.
Da = Tripping incline or offset, joining two concentric surfaces on D.
E = Curved trip arm on C.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa, raising A and opening the valve. Independently of the motion of B', the eccentrically operated knock-off cam D releases A when Cb comes in contact with Da. Drop lever A is then pulled down by the dashpot, closing the valve. The governor determines the lead of the knock-off cam D, thereby changing the admission from no steam to 0.8 stroke.

The gear under discussion is essentially the same as that formerly used by the Brown Corliss Engine Company, which the Wisconsin Engine Company succeeded, and early reference to it will be found under that name.

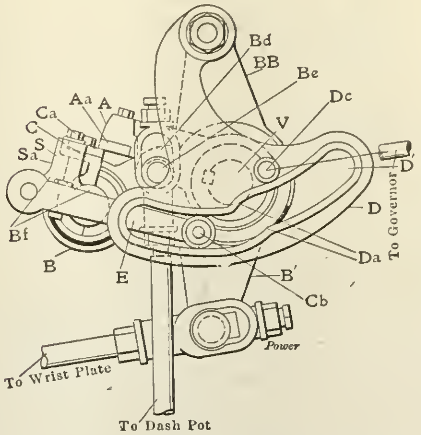

FIG. 21. FULL-STROKE NORDBERG GEAR

A = Drop lever or valve-closing arm keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on A.
$B, B', B\ B$ = Three-arm lever formed with a sleeve and turning on the bonnet through which the valve stem passes. B carries hook C, spring S (incased in Sa) and arm Bf which carries E on the stud projection. B' is connected to the wristplate. $B\ B$ holds D suspended from the stud.
Bd = Pivot pin on A extending into the path of Be.
Ba = Projection on the outer sleeve face of triple lever ($B, B', B\ B$).
Bf = Arm connected to stud on B, outside of C and overhanging E, which it holds suspended by the stud on its end and by S.
C = Hook in form of L-shaped arm, carrying catchblock Ca at the upper end and fitted to turn in the sleeve on stud of arm B.
Ca = Catchblock on C.
Cb = Trip roller on E, which projects into a slot D' in cam D.
D = Knock-off cam in shape of two arcs, each concentric with pivot connection on arm $B\ B$, and forming their junction, a short incline or offset at Da by which the trip arm E is turned sufficiently to disengage the latch block Ca.
D' = Slot in D into which Cb projects.
Da = Offset or tripping point in D.
Dc = Pivot pin on cam, which connects to cutoff eccentric.
E = Trip arm pivoted at one end on the stud which is journaled in Bf and carries the roller Cb at the other end.
S = Spring and bolt (incased) for holding Cb engaged to the upper surface of the slot in cam D.
Sa = Vertical socket formed on the upper side of Bf and containing S.
V = Valve.

OPERATION

When B' is pulled toward the wristplate, Ca engages Aa and carries A with it, opening the valve. Independently of the motion of B' the eccentric-operated knockoff cam D releases A where Da rides over Cb. This occurs exactly when Dc passes the center of the valve stem, no matter in what position lever $B\ B$ happens to be. A is then pulled down by the dashpot, thus keeping the valve closed. During the return motion Ca is kept clear of Aa and A until ready to engage, thus minimizing noise and wear. The action of the governor is such that it alters the lead of the knock-off cam D, thereby changing admission from no steam to 0.8 of the stroke. In case the dashpot should for any reason fail to close the valve, then the return movement of the levers B, B' and $B\ B$ causes Bd to engage Be, thus closing the valve.

The Vilter Manufacturing Company, which for many years used the standard hook mechanism, has recently brought out a new gear presenting many features similar to that shown in Fig. 18.

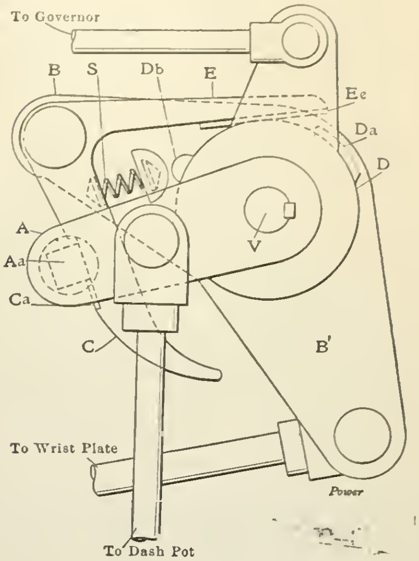

FIG. 22. NEW YORK ENGINE COMPANY'S GEAR

A = Drop lever, keyed on outer end of the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on A.
$B\ B'$ = Bell crank loosely mounted on outer end of valve-stem bonnet, carrying hook C. It receives its motion from the wristplate.
C = Hook carried by B.
Ca = Block carried by C.
D = Knock-off cam, actuated by the governor.
Da = Trip block carried by D.
Db = Safety block.
E = Trip lever.
Ee = Block carried by E.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled toward the wristplate, Ca engages Aa and raises A until Ee engages Da. This forces the trip lever E upward and releases A, which is quickly drawn down by the dashpot and closes the valve.

Fig. 19 shows the standard Nordberg gear. Its peculiar feature lies in the form of the hook, which is an L-shaped lever carrying at its upper end a block that engages with the block on the drop lever. The design of this hook, it is claimed, enables a comparatively short crank arm to be used, thus reducing the travel of the hook, and the size of the eccentric to a minimum. The gear, as now built, was described very fully in an article by Osborne Monnett which appeared in the February 22, 1910, issue of POWER.

Fig. 20 illustrates another of the inventions of E. T. Sederholm, now assistant chief engineer of the Nordberg Manufacturing Company. This is a long-range cutoff gear used for many years on the more powerful engines built by Fraser & Chalmers, of Chicago. From the drawing it will be seen that, in one respect, the steam hook is the same as the well known Reynolds type. It is a drag hook, but instead of having its knock-off portion in one piece with the hook, like the Reynolds gear, it has only the lifting portion of the hook keyed to the inner end of the small rocker shaft mounted on the steam lever. The outer end of this same rocker shaft carries a small curved lever ending with a pin provided with a roller. This pin and roller are in the exact center of the bonnet. From this it follows that, no matter in what position the steam lever may be, this pin remains stationary except so far as it turns around on its own axis. The knock-off cam, however, is not stationary but is mounted on a projection of the bonnet and is operated by means of a small separate eccentric. In this manner the cutoff can take place whether the steam valve is opening or closing; in other words, in whatever position of valve gear and piston that may be desired. A cutoff up to 0.8 of the stroke can readily be obtained. The variation in the point of cutoff is obtained by means of changing the lead of the knock-off cam.

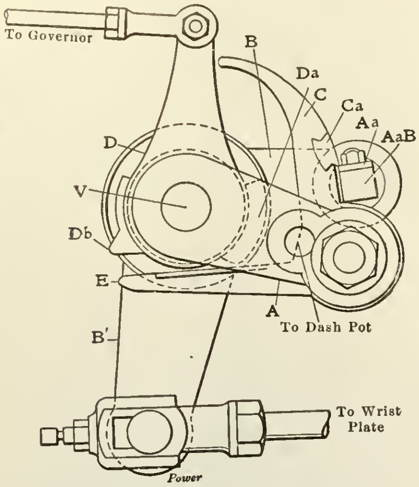

FIG. 23. MINNEAPOLIS GEAR

A = Drop lever or steam crank keyed to the valve stem, which it actuates, and connected to the dashpot rod. It also carries claw *C*.
Aa = Block on *AaB*.
AaB = Catchblock carried by *B*.
B B = Bell crank which carries the catchblock *Aa* and rod to wristplate on arm *B'*. This bell crank receives its motion from the wristplate and swings loosely on the valve bracket.
C = Claw.
Ca = Block on claw.
D = Knock-off lever or toe crank which swings loosely on the valve bracket.
Da = Steel toe fastened to toe crank (projection for tripping).
Db = Safety toe fastened to crank (projection for safety stop).
E = Part of claw *C*.
V = Valve stem.

OPERATION

When crank *B'* is pulled in the direction of the wristplate, *Aa* engages *Ca*, thereby raising *A* until *E* comes in contact with *Da*. This forces *Ca* away from *Aa* and causes the release of *A*, which is immediately drawn downward by the vacuum pot and closes the valve.

The Nordberg Manufacturing Company's full-stroke gear shown in Fig. 21 is somewhat similar to the gear just described except in two important features. The pin on the knock-off lever on the steam hook is not located in the exact center of the bonnet, but has a slight motion around that center. This motion, however, is counteracted by the fact that the knock-off lever is also mounted on a steam lever and, therefore, has exactly the same angular motion as the pin just mentioned. From this it will be evident that in this gear it is immaterial whether the valve is opening or closing. The other important difference is that instead of the knock-off cam being merely a one-sided cam, it is an inclosed cam, positively holding the pin of the knock-off toe. By this means the knock-off cam not only disengages the gear but it also engages the hook again at the right time, the idea being to do away with the rubbing friction and wear found in all types of Corliss gears where the hook is pushed into engagement by means of a spring.

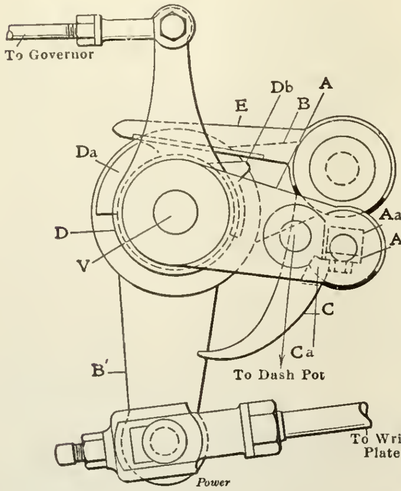

FIG. 24. MINNEAPOLIS GEAR INVERTED FOR HIGH SPEEDS

A = Drop lever or steam crank keyed to valve stem which it actuates, and connected to the dashpot rod.
Aa = Block on *AaB*.
AaB = Catchblock on drop lever *A*.
B B' = Bell crank which carries claw *C* and a rod to the wristplate. This bell crank receives its motion from the wristplate and swings loosely on the valve bracket.
C = Claw (hook).
Ca = Block on claw.
D = Knock-off lever or toe crank which swings loosely on the valve bracket.
Da = Steel toe fastened to the toe crank (operating cam).
Db = Safety toe fastened to the toe crank (safety cam).
E = Part of claw *C*.
V = Valve stem.

OPERATION

When crank *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, thereby raising *A* until *E* comes in contact with *Da* and forces *Ca* away from *Aa*, causing the release of *A*, which is immediately drawn downward by the dashpot and closes the valve.

Fig. 22 shows a gear designed by the New York Engine Company, but as that concern builds few Corliss engines, this has been very little used.

In Fig. 23 is illustrated the gear used by the Minneapolis Steel and Machinery Company on engines running at moderate speeds, and Fig. 24 shows the same gear inverted for high-speed Corliss engines. This is an adaptation of the Corliss crab claw.

Fig. 25 is a sketch of the Atlas gear, the mechanism of which is so simple as to require little or no explanation. The principal feature of interest consists in the form of the hook.

Figs. 26 and 27 illustrate the Corliss gears designed by the Southwark Foundry and Machine Company, which are designated as "Style A" and "Style B," respectively. These are modifications of the standard hook type, the difference between them being plainly shown by the sketches. The closing action of "Style B," being positive, renders it particularly suitable for high speeds.

In a class by themselves are the gears shown in Figs. 28 and 29. The first of these is the gear originally used by the Wm. A. Harris Steam Engine Company and the second is the mechanism which the writer understands to be now employed. It is claimed for the gear illustrated in Fig. 29 that it is one of the simplest releasing mechanisms used on Corliss engines at the present time, and that the absence of springs, links, etc., tends to make it very smooth running. The movement of the hook is positive; and, as the center upon which the releasing bar turns is below the plane of contact of the blocks, the disengagement is so easy as to cause no perceptible jar on the lever.

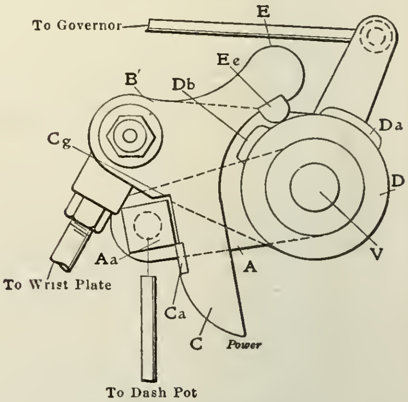

FIG. 25. ATLAS GEAR

A = Drop lever or crank arm mounted on the valve stem, which it actuates, and connected to the dashpot rod.
Aa = Catchblock on *A*.
B' = Arm shown by dotted lines, which takes the place of the usual bell crank. This arm receives its motion from the wristplate.
C = Hook carried by *B'*.
Ca = Block carried by hook.
Cg = Safety stop.
D = Knock-off cam, the position of which is determined by the governor.
Da = Projection for tripping.
Db = Projection for safety stop.
E = Trip lever.
Ee = Projection on hook for tripping.
V = Valve stem.

OPERATION

When *B'* is pushed away from the wristplate, *Ca* engages *Aa* and raises *A* until *Ee* comes in contact with *Da*, thus forcing *E* outward and causing the release of *A*, which is immediately drawn downward by the dashpot and closes the valve. Should for any reason the valve fail to close, the return movement of the lever *B'* will cause *Cg* to close the valve.

This device, or one very similar to it, has also been used by the Harrisburg Foundry and Machine Works; but, in place of the ordinary dashpot used to close the steam valve, the gear, as designed by this company, was connected with a dashpot having a steam cylinder attached. Thus the closure was effected by steam, instead of gravity or vacuum, enabling the engine to be run at higher speeds than were then considered permissible with the ordinary Corliss gear. The use of this mechanism has, however, been discontinued since the introduction of the so called four-valve engine perfected by B. T. Allen, chief engineer of the Harrisburg company.

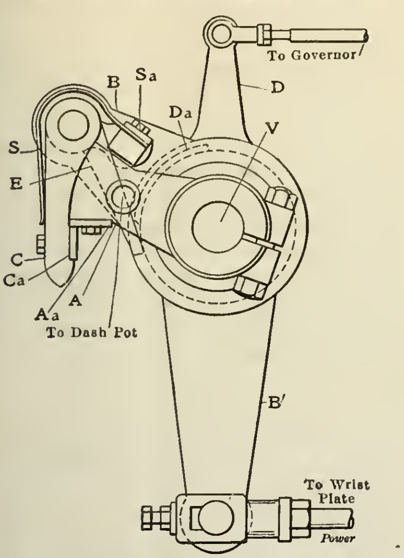

FIG. 26. SOUTHWARK GEAR, STYLE "A"

A = Drop lever or crank arm mounted on the valve stem and connected to the dashpot rod.
Aa = Catchblock on *A*.
B B' = Bell crank which receives its motion from the wristplate or directly from the eccentric.
C = Hook or pawl carried at extremity of *B* on a short shaft journaled in *B B'*. *C* is normally held by the spring *S* against a projecting face on *A*, radial to the center of the stem.
Ca = Block on hook.
D = Knock-off cam mounted on a bracket. The position of this cam is determined by the governor.
Da = Projection on *D* for tripping.
E = Trip lever mounted on the same shaft with hook *C*, on the opposite side of *B*.
S = Spring.
Sa = Spring block.
V = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages with *Aa* and causes *A* and *B B'* to move in unison until *E* comes in contact with *Da* on *D*; *E* is then pushed outward and with it *C*, thereby releasing *A*, which is immediately drawn downward by the dashpot and closes the valve. When the governor reaches its highest position the projection *Da* holds *E* in its raised position and prevents *Ca* from engaging *Aa* and keeps the valve closed. When the governor is in the lowest position, *Da* does not touch *E* and release *A*. The spring block *Sa* closes the valve if the dashpot fails to do so.

Fig. 30 is a reproduction of the well known Bass gear, which, like the two gears just described, is of a type peculiar to itself. The construction and details of operation are clearly shown by the diagram and accompanying explanation. This gear is designed to be dust proof. The valve stems are relieved from any stress, which is borne by the bonnets, practically the same object being accomplished as in the standard gears previously described. The working parts of the gear are very accessible for adjustment and removal, and the absence of the usual hook permits the use of larger wearing surfaces, such as the trip blocks present, than could otherwise be obtained in the same space.

Fig. 31 represents the first of the cam and roller gears to be treated in this article. It was designed for the Philadelphia Engineering Company's engine. The action of the mechanism may be plainly understood from the drawing and accompanying illustration. It combines simplicity with noiseless operation.

A mechanism of somewhat similar type is the Fishkill gear, illustrated in Fig. 32. With the action of the rollers there is no appreciable strain in turning the triple lever on its axis and consequently no tendency to disturb the normal action of the governor.

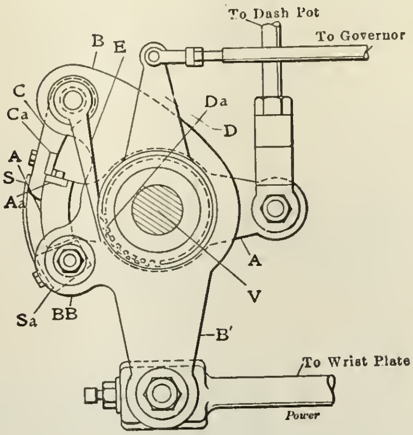

FIG. 27. SOUTHWARK GEAR, STYLE "B"

A = Drop lever or crank arm mounted on the valve stem and connected to the dashpot rod.
Aa = Catchblock on *A*.
B, B', B B = Three-arm lever which receives its motion from the wristplate, or directly from the eccentric.
C = Hook or pawl carried at extremity of *B* on a short shaft journaled in *B*, the opposite end of which carries the trip lever *E*. *C* is normally held against *A* by the spring *S*.
Ca = Block on hook.
D = Knock-off cam lever mounted on bearing supporting the gear. The position of this cam lever is determined by the governor.
Da = Projection for tripping.
E = Trip lever mounted rigidly on the same shaft with *C*.
S = Spring.
Sa = Closing block on *B B*.
V = Valve stem.

OPERATION

When lever *B* is pulled in the direction of the wristplate, *Ca* engages with *Aa* and causes *A* and *B* to move in unison until *E* comes in contact with *Da* on *D*; *E* is then pushed outward and with it *C*, thereby releasing *A*, which is immediately pushed down by the dashpot and closes the valve. When the governor reaches its highest position the projection *Da* holds *E* in its raised position, preventing *Ca* from engaging with *Aa* and keeping the valve closed. When the governor is in the lowest position *Da* does not touch *E* and release *A*. The spring block *Sa* closes the valve if the dashpot fails to do so.

Fig. 33 illustrates the details of the Watts-Campbell gear, the principal merit of which is its simplicity and noiseless operation.

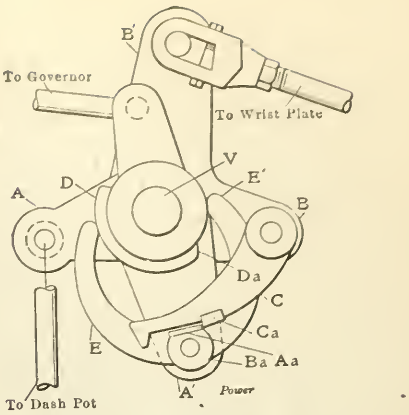

FIG. 28. ORIGINAL HARRIS GEAR

A A' = Drop lever or bell crank keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock carried by *Ba*.
Ba = Sleeve carried by a pin at the end of *A'* and holding a catchblock on top.
B B' = Bell crank receiving motion from the wristplate.
C = Pawl which takes the place of the usual hook.
Ca = Block carried by *C*.
D = Knock-off cam, loosely mounted on the bonnet, the position of the cam being determined by the governor.
E E = Curved arms extending from *Ba* and acting as trip levers.
V = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa* and raises *A* until *E'* comes in contact with *Da*. This pushes *E'* outward and turns *Aa* around sufficiently to release *A*, which is immediately drawn downward by the dashpot, thus closing the valve. Should the governor reach its lowest position, *D* would hold *E E* in such a position that *Ca* cannot engage *Aa* and the valve would remain closed. The hook at the end of *C* is also a device to insure the closing of the valve should the dashpot fail to act.

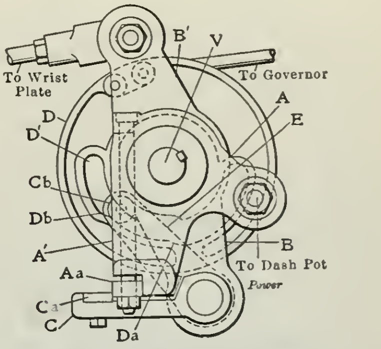

FIG. 29. LATER HARRIS GEAR

A A' = Bell crank keyed to the valve stem which it actuates. *A* is connected to the dashpot rod.
Aa = Block carried at the end of *A'*.
B B' = Two-arm lever loosely mounted on the bonnet and receiving its motion from the wristplate.
C = L-shaped hook carried by *B*, and moved by roller running in cam groove *D'*.
Ca = Block carried by *C*.
Cb = Trip roller on upper extremity of *E*.
D = Knock-off cam loosely mounted on the bonnet, the position of which is determined by the governor.
D' = Cam-shaped groove.
Da = Concentric deviation from the valve stem for tripping.
Db = Concentric deviation from the valve stem for safety stop.
E = Trip lever, keyed to *C*.
V = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *Cb* comes in contact with *Da* and strikes into the lower deviation of *D'*. This forces *C* outward and releases *A A'*, which immediately draws *A* downward by means of the dashpot and closes the valve. Should the governor reach its lowest position, *Cb* would rest in the upper deviation of *D'*, preventing *Ca* from engaging *Aa* and keeping the valve closed.

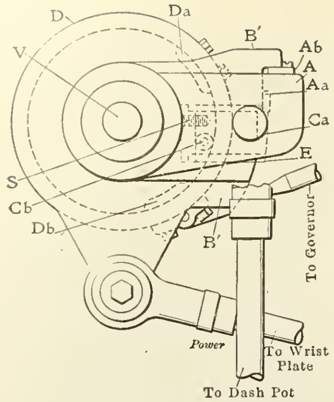

FIG. 30. BASS GEAR

A = Drop lever keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Catchblock on A.
Ab = Leather strip on a lateral projection of A, to avoid hammering of the bell crank B' on the drop lever A.
B B' = Bell crank receiving its motion from the wristplate.
Ca = Block carried by E.
Cb = Trip roller on E.
D = Knock-off cam, the position of which is determined by the governor.
Da = Projection for tripping on the inner rim of the knock-off cam.
Db = Projection for safety stop.
E = Sliding trip block, fitted in a recess of B B', which takes the place of the trip lever. This has at its outer end a die block Ca and on its inner side a roller Cb.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa and raises A until Cb comes in contact with and rides over Da; the latter causes Cb, and with it E, to slide inward toward the valve stem, thereby disengaging Ca from Aa and releasing A, which is immediately drawn downward by the dashpot, closing the valve.

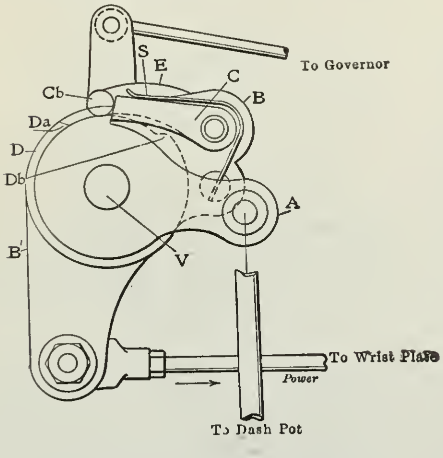

FIG. 31. PHILADELPHIA GEAR

A = Drop lever or crank arm, mounted on the valve stem, and connected to the dashpot rod.
B B' = Bell crank, which receives its motion from the wristplate.
C = Hook or pawl carried at extremity of B' on a short shaft journaled in B, the opposite end of which carries E. C is normally held by the spring S against the top of A.
Cb = Trip roller.
D = Knock-off cam mounted on the valve stem. The position of this cam is determined by the governor.
Da = Projection for tripping.
Db = Projection for safety stop.
E = Short curved lever carrying a roller Cb.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, C engages A and causes A and B B' to move in unison until Cb comes in contact with Da on D; E is then raised, and with it C, thereby releasing A, which is immediately drawn downward by the dashpot and closes the valve.

In Fig. 34 is shown another type of long-range valve gear, the working of which is similar to that of the gear illustrated by Fig. 21. A separate eccentric is used for operating the knock-off mechanism, but instead of the slotted knock-off lever and floating lever, a bell crank is fastened to the bonnet, one arm of which carries a small roller moving across the center of the valve stem. One leg of the steam hook extends to the center of the valve stem, and whenever the roller crosses the center of the stem the hook is released. This gear was used in the 'eighties on a quadruple-expansion engine built by Edwin Reynolds for the Warren Manufacturing Company, and on several large rolling-mill engines for the Carnegie Steel Company.

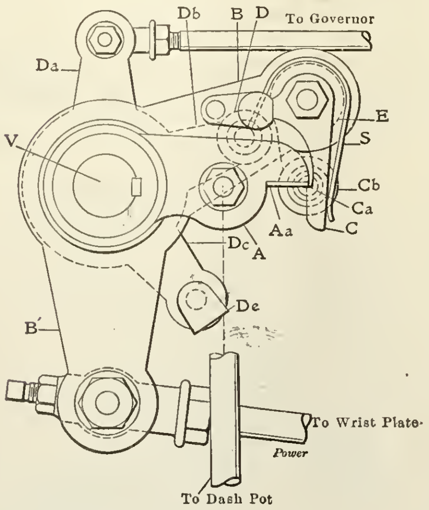

FIG. 32. FISHKILL GEAR

A = Drop lever or crank arm mounted on the valve stem which it actuates, and connected to the dashpot rod.
Aa = Block carried by A and by C.
B B' = Bell crank carried on a projection of the cylinder bonnet and receiving its motion from the wristplate.
Da, Db, Dc = Triple lever oscillating upon a projection of the bonnet, Da being connected by an adjustable rod to the governor.
C = Hook at the end of a small rock shaft, carried by B.
Ca = Notch in C.
Cb = Roller carried by E.
D = Roller carried by arm Db of triple lever at the end of a pin. Its position is determined by the governor.
De = Adjustable roller carried by arm Dc of the triple lever; its position is determined by the governor.
E = Lever carried by rock shaft on the side opposite from C.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa; A is carried along an arc concentric with the valve stem, and as Cb passes over D it is pushed outward, releasing A, which is immediately drawn downward by the dashpot and closes the valve. The arm Dc of the triple lever carries an adjustable cam Dc. Should the governor reach its lowest position Dc would come in the way of the roller Cb, which rides on top of it, thus preventing C from engaging Aa, and keeping the valve closed.

The Providence Engineering Works, successors to the Providence Steam Engine Company, and Rice & Sargent Engine Company claim to have been the first manufacturers in the world to introduce a releasing valve gear of the gravity cutoff type, where the engaging mechanism drops into position by the aid of gravity alone rather than by the aid of any springs. This, they state, permitted the old Greene engine to operate at speeds much higher than those that were then the practice. Contemporaneously with the former company, Rice & Sargent developed practically the same idea with an improved method of application. This gear, as at present perfected, is shown in Fig. 35. No wristplate is used; the valve motions are as short as possible and all parts of the releasing mechanism are made very strong with large wearing surfaces. A very clear description of the details and operation of the gear is given in Collins' book on "Valve Setting" heretofore referred to.

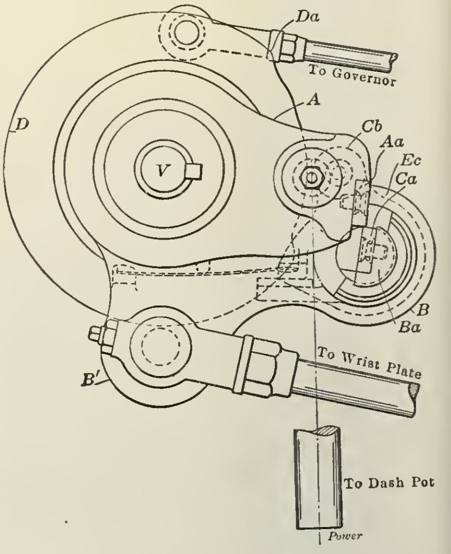

FIG. 33. WATTS-CAMPBELL GEAR

A = Drop lever or crank arm keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Block carried on the end of A.
B B' = Bell crank receiving its motion from the wristplate. At the end of B' is fixed the pin Ba, upon the inner side of which is secured the rocker arm E C carrying the roller Cb. On the outer side Ba carries a die block Ca.
Ca = Catchblock on Ba.
Cb = Trip roller on E C.
D = Knock-off cam, the position of which is determined by the governor.
Db = Safety stop.
Da = Projection for tripping.
E C = Trip lever held against D by a spring.
V = Valve stem.

OPERATION

When B' is pulled toward the wristplate, Ca engages Aa, raising A until Cb comes in contact with Da, thus forcing E C outward and releasing A, which is immediately drawn downward by the dashpot and closes the valve.

The Franklin gravity valve gear used by the Hewes & Phillips Iron Works is illustrated in Fig. 36. The latch is a bronze bar hollowed out for lightness and suspended in such a manner that gravity enables it to act properly even at speeds as high as 200 revolutions per minute without the aid of the spring, which is furnished merely as additional security. When the latch blocks engage, the contact of the latch arm takes place upon a fiber block at the side of the upper block, thus avoiding even the clicking sound which usually accompanies the operation of a releasing gear.

Following are features common to the majority of Corliss gears, of which no mention has been made in the preceding descriptions, except where it was desirable to emphasize some special point in order to show the peculiarity of construction.

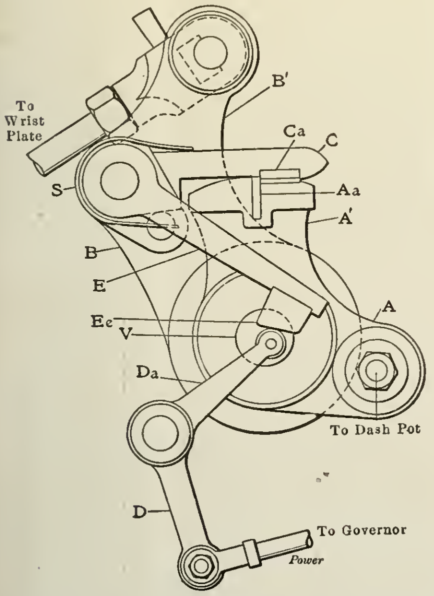

FIG. 34. REYNOLD'S "WARREN ENGINE" GEAR

$A\,A'$ = Drop lever or crank arm mounted on the valve stem which it actuates. A is connected to the dashpot rod.
Aa = Catchblock carried by A'.
$B\,B'$ = L-shaped crank receiving its motion from the wristplate. At the upper end of B' the crank is connected to the wristplate; $B\,B'$ also carries hook C.
C = Hook.
Ca = Catchblock on C.
$D\,Da$ = Bell crank connected to the bonnet; lever D connects with the governor, while lever Da, which carries a small roller at its outer end, acts as the knock-off cam. Its position is determined by the governor.
E = Knock-off lever.
Ee = Knock-off block carried by lever E.
S = Spring.
V = Valve stem.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa, causing $A\,A'$ and $B\,B'$ to move in unison until Ee comes in contact with roller on Da, which pushes E and C upward. This releases $A\,A'$, which is immediately drawn downward by the dashpot and closes the valve.

Blocks when used are so offset as to be in contact for only half their thickness, thereby enabling such changes in position as to present eight different edges to wear.

Parts made to standard gage and, therefore, interchangeable.

The use of vacuum dashpots, with dust-proof casing and air-regulating valves.

A mechanism designed in each instance with the idea of eliminating reaction on the governor.

Valve connections so arranged that all necessary adjustments can be made while the engine is running.

Wristplate or direct-connected eccentric motion so designed that the steam port may be fully opened very early in the stroke.

Exhaust valves seated by gravity.

Provision for automatically closing steam valves and keeping them shut should the gear get out of order.

Nearly all of the different types of valve gear mentioned in this discussion answer the previous requirements, and so far as actuating the admission valves of the cylinders is concerned, the results are essentially the same. They open the valves in about the same time, and they release in about the same time, and if the vacuum pots or similar arrangements are what they ought to be, the valves will be closed in practically the same time.

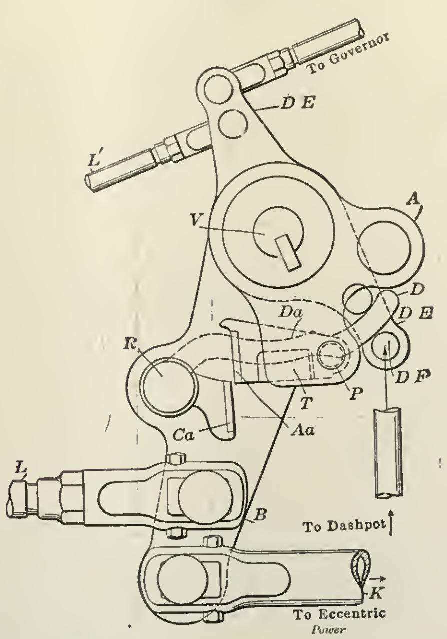

FIG. 35. RICE & SARGENT GEAR

A = Drop lever or crank arm keyed to the valve stem which it actuates, and connected to the dashpot rod.
Aa = Latchblock resting on and against block T in the rear of lever A, when at work, but turning on pivot P when acting as a latch.
B = Crank lever or rocker arm, which carries Ca, D, R, K and L; it receives its motion through K from an eccentric on the engine shaft.
Ca = Block on toe lever; the latter is rigidly attached to spindle R.
D = Cam lever rigidly attached to spindle R on rear of arm B. This lever is carried between two rolls $D\,F$.
Da = Rise on D.
$D\,E$ = Knock-off lever mounted loosely on the valve-stem journal, the upper extremity is connected to the governor rod and the lower holds rollers $D\,F$ on pins. The amount of valve opening and the length of cutoff depend upon the position of this lever, as controlled by the governor—the farther the rollers $D\,F$ are to the left, the earlier the cutoff.
$D\,F$ = Cam rolls which turn on pins in cutoff lever $D\,E$.
K = Connections, etc. (toward eccentric on engine shaft).
$L\,L'$ = Connections to a similar inlet gear at the back end of the cylinder.
R = Spindle journaled in arm B'. This spindle holds Ca and D.

OPERATION

When B is pulled in the direction of the eccentric, Ca engages Aa, thereby raising A until Da passes between rollers $D\,F$. This forces D downward, thus causing Ca to release Aa; A is immediately drawn downward by the dashpot and closes the valve. On the return movement of rocker B, cam D and toe block Ca are raised to the engaging position. Just before the end of this return movement the toe Ca raises latch Aa sufficiently to let the toe by; the latch turning on its pivot. The latch then drops in front of toe for engagement.

The valve gear operated with rollers is apt to make less noise than the one in which the leg of a hook slides over a cam; but, in the writer's judgment, experience has demonstrated beyond question that the valve gear which has the leg of the hook sliding over the cam is more simple and more durable than the other type. The tendency during the last few years, owing to the practice of running Corliss engines at a higher speed, has been toward adopting valve gears with gravity drops and dispensing entirely with springs. Springs, however, insure positive action, whereas the gravity-drop valve gear without springs may not be absolutely reliable under all conditions of work. This is so true that a majority of manufacturers have been slow to adopt it, although present indications are that a number of such gears will be thoroughly tried out in the near future.

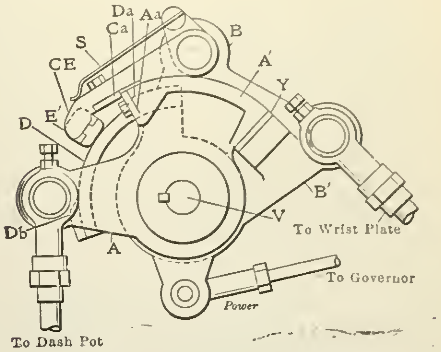

FIG. 36. FRANKLIN GEAR MADE BY HEWES & PHILLIPS IRON WORKS

$A\,A'$ = Drop lever or crank arm keyed to the valve stem which it actuates. A is connected to the dashpot rod.
Aa = Catchblock on A'.
$B\,B'$ = Bell crank which carries a hook and knock-off lever combination $C\,E$, at the bottom extremity of which the trip block E is attached. This bell crank receives its motion from the wristplate.
$C\,E$ = Hook and trip lever.
Ca = Catchblock carried by hook.
D = Knock-off cam mounted on the valve stem between $A\,A'$ and $B\,B'$. The position of this cam is determined by the governor.
Da = Projection for tripping (operating cam).
Db = Projection for safety stop (safety cam).
E' = Trip block.
S = Spring.
V = Valve stem.
Y = Safety block on B'.

OPERATION

When B' is pulled in the direction of the wristplate, Ca engages Aa, thereby raising $A\,A'$ until E' comes in contact with Da. This forces E' away, causing $C\,E$ to release $A\,A'$, which is immediately drawn downward by the dashpot and closes the valve. If for any reason the valve should not close, safety block Y on B' will act.

On entering the boiler room, ascertain if the valves between the boiler and the water column are open. Then blow down the water column, noting the return of the water in the glass, and test the gage cocks until absolutely sure of the water level. This should be done at the beginning and end of each watch.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Storage Batteries Central Stations and Large Buildings

By Norman G. Meade

Storage batteries are used in large buildings and central stations for several purposes. In general there is a reduction in coal consumption and operating expenses where they are used, due to the generating machinery being run at the load of greatest efficiency while in service and being shut down entirely during hours of very light load, the battery then carrying the whole of the load. This applies to comparatively small central stations, however, as the large stations operate throughout the entire twenty-four hours. The large companies, however, utilize storage batteries in substations in many instances, which is especially advantageous where the day load consists largely of elevators and similar disturbing machines. By their use it is possible to obtain excellent regulation regardless of the load conditions. In large office buildings also, where many high-speed elevators are in use, storage batteries are used to great advantage in smoothing out the load on the generating equipment.

A smaller generating plant is required where the battery takes the peak of the load, which usually lasts only a few hours but, when no battery is used, necessitates generating equipment of sufficient capacity to give the maximum output, which in some cases is double the normal output. The application of storage batteries to central stations is, of course, confined to direct-current systems or to direct-current distribution mains supplied by rotary converters which take alternating current from a system of high-tension feeders.

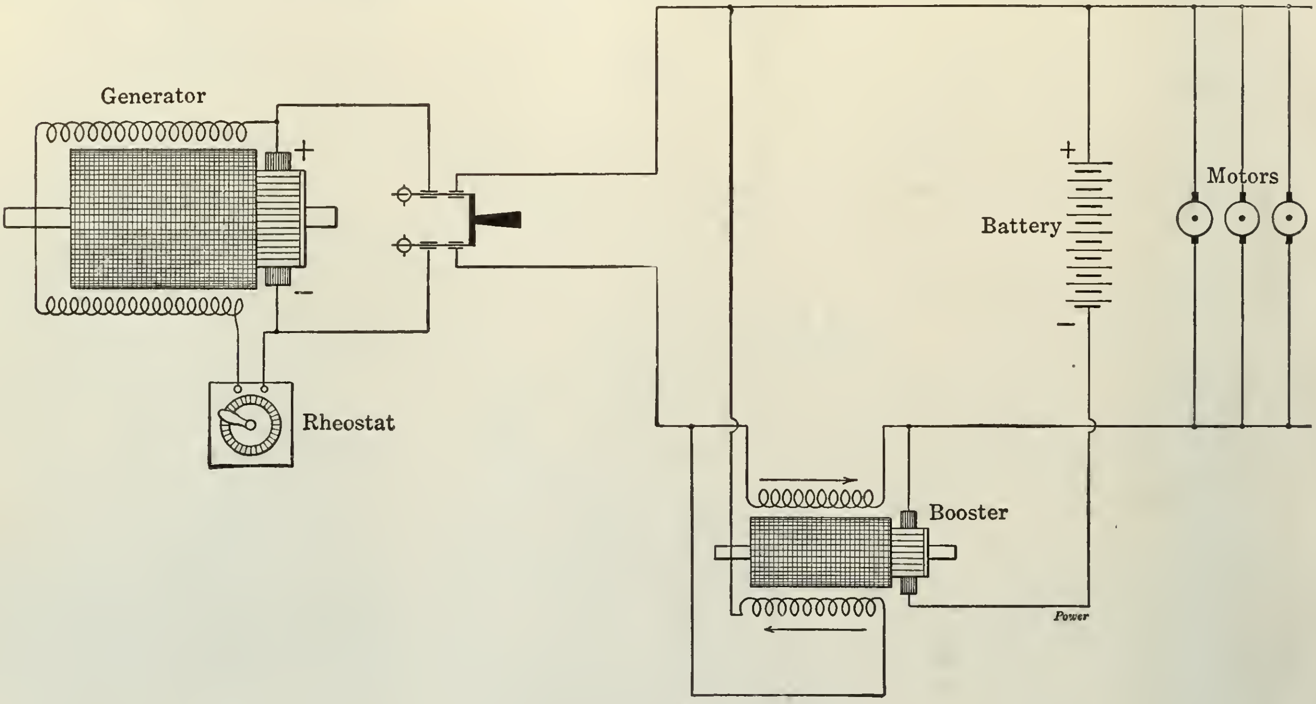

Fig. 1. Automatic Booster System

A system largely used where the load fluctuates greatly is illustrated diagrammatically in Fig. 1. The armature of a differential booster is connected in series with the battery and the two are connected across the line, as indicated. The shunt field winding is connected across the line and the series field winding in series with the line. At normal load on the generator the current through the series field winding produces a magnetizing effect equal to that of the shunt winding. The magnetism due to the series winding tends to induce in the armature an electromotive force opposing that of the generator, whereas the magnetism due to the shunt winding induces an electromotive force in the same direction as that of the generator.

When the external load equals the nor-

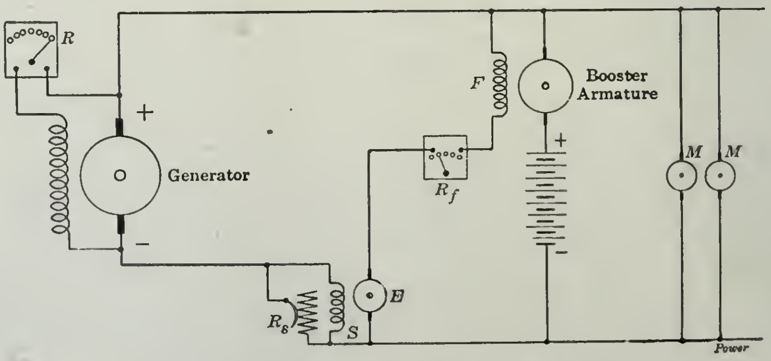

Fig. 2. Booster with Automatic Exciter

mal generator output, no current flows through the battery. An increase above normal load in the main line increases the excitation of the series field winding, which is then stronger than the shunt winding and produces a booster voltage agreeing in direction with that of the battery, which enables the latter to deliver current to the line. If the load should decrease, the diminution in the excitation of the series winding would allow the shunt winding to set up a magnetization such that the booster voltage would assist that of the generator, and current would flow through the battery in the charging direction. By means of this arrangement, therefore, any difference between the normal generator output and the actual load in circuit is absorbed or supplied by the battery as may be required, and the load on the generator is therefore practically constant. There are many varieties of windings for automatic boosters, but they all involve the principle of differential field windings just described.

Automatic boosting systems have been devised in which the field excitation of the booster is varied to meet the requirements of charge and discharge by an external apparatus controlled by the generator current. With these external control devices the booster is a simple shunt-wound generator. Fig. 2 is a schematic diagram of one such system. The external control in this case is effected by means of an exciting generator E, the field winding S of which is connected in series with the main generator circuit and shunted by an adjustable resistance R_s. The booster field winding F is connected in series with the exciter armature winding and this circuit is connected across the main line. The line potential tends to send current through the field winding F and the exciter armature, and this tendency is opposed by the electromotive force of the exciter armature. When the external load in the main line, represented by motors $M\ M$, is normal, the excitation due to normal current through the field winding S produces an electromotive force in the exciter armature equal to the line potential; consequently, no current flows in the booster field winding and the booster electromotive force is zero.

With an increase in load there will be an increase in the main line current, and consequently in the field winding S; this will increase the exciter electromotive force, which, being then greater than the line electromotive force, will send a current through the field winding F and cause the booster to generate an electromotive force agreeing with that of the battery; this enables the battery to discharge and assists the generator to carry the load. When the load decreases, the excitation of the field winding S decreases and the electromotive force of the exciter falls below the line voltage, which then sends a current through the field winding F and the armature winding of the exciter in a direction opposite to the exciter pressure. This produces a booster electromotive force agreeing with that of the generator, and current passes through the battery in the direction to charge it. Like the differential booster, this system maintains a practically constant load on the generator.

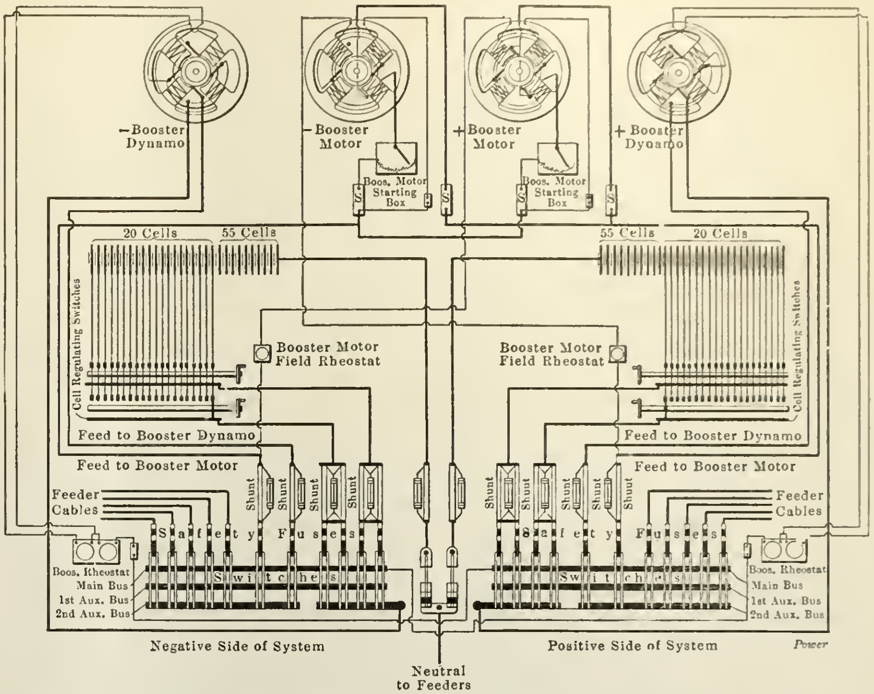

Fig. 3. Switchboard Connections of Combined Booster and End-cell Regulation for a Three-wire System

The rheostat R_f controls the booster electromotive force for a given change in the electromotive force of the exciter, and the variable shunt R_s allows a change in the proportion of the generator current through the exciter field winding S. In practice, the booster, exciter and motor are all supported on one base and the armatures are mounted on a single shaft.

The regulation of the charging current by means of a shunt-wound booster and the discharge by means of end cells, as described in the previous article* on the application of storage batteries to private plants, is also applicable to central-station service in many cases. Fig. 3 is the complete wiring diagram of such a system applied to three-wire direct-current mains. Two complete outfits, each comprising a booster, a battery and an end-cell switch, are used, one outfit being connected between the positive main and the neutral, and the other between the negative main and the neutral. This combination may be used either to maintain a steady voltage or to equalize the loads in the two divisions of a three-wire system, or both.

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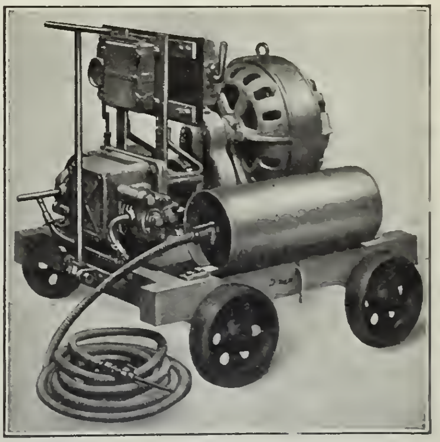
*Page 1472, August 16 issue.

A Portable Motor Driven Air Compressor

The operation of small portable machinery by electric motors has so much to recommend it in the way of convenience and economy that very extended use is now made of the method. The accompanying illustration shows a small portable air-compressor equipment recently supplied to a large flour mill and granary at Rio de Janeiro by the British Westinghouse Electric and Manufacturing Company, Ltd., which, a short time before, carried out the complete electrical equipment of these mills.



Portable Compressor Outfit

This equipment is used for "blowing out" 600-horsepower motors and for cleaning the other machinery around the mills. It consists of a standard Westinghouse air-brake compressor, driven through single-reduction gearing by a 3-

horsepower three-phase squirrel-cage induction motor, mounted, with its control panel, on a substantial wooden truck which is equipped with rubber-tired wheels so that it can be easily moved from point to point. The current for operating it is taken from plug receptacles on the mill walls and conveyed by a flexible cable to a three-pole double-throw inclosed switch on the control panel. The connections are so arranged that when the motor is thrown on the line at starting the fuses are cut out, and after the motor has attained a fair speed, the switch is thrown over to the "running" position, putting the fuses in circuit.

The importance of keeping electrical machinery free from dust cannot be overestimated, and compressed air direct from a compressor is the best form of dust remover, as it blows out all the crevices and cleans the windings thoroughly without impairing the insulation.

Performance of Railway Motors

In a paper on the Electrification of Suburban Railways, presented at the joint meeting of The Institution of Mechanical Engineers of Great Britain and the American Society of Mechanical Engineers, F. W. Carter, a prominent English engineer, gave the following interesting data regarding railway-motor performance:

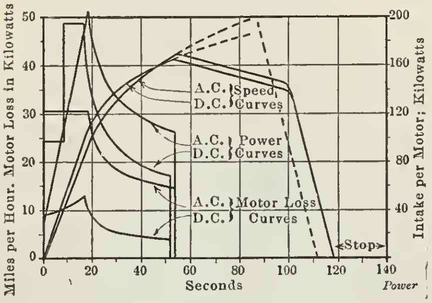


FIG. 5.* TRAIN CHARACTERISTICS

The ultimate temperature rise, in Centigrade degrees, of the hottest accessible part of an ordinary motor, entirely inclosed, after continuous service, is approximately given by the equation

$$T_2 - T_1 = \frac{7\,P}{\sqrt[3]{W^2}}$$

in which P represents the watts lost in the motor (exclusive of the gears) and W is the weight of the machine in pounds, not counting the gears and gear case.

The motor losses in suburban service are about 7 per cent. of the motor intake, with direct-current machines, and 15 to 18 per cent. with single-phase machines.

The heating of the motors has always been a limiting factor in the design of suburban-railway equipment. The use of forced draft for cooling the motors, while practicable for electric locomotives and single cars, is unpractical on multiple-unit trains in suburban service because the equipment cannot be inspected frequently and is therefore too liable to breakdowns.

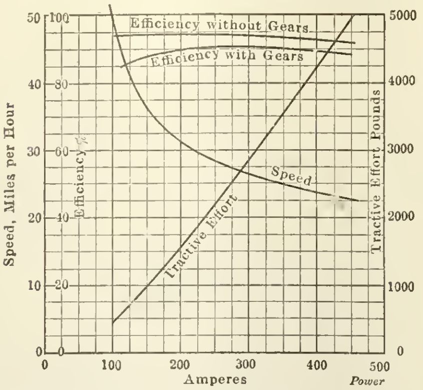


FIG. 6. DIRECT-CURRENT MOTOR CHARACTERISTICS

Fig. 5* gives speed, power and motor-loss curves corresponding to typical suburban schedule runs with direct current and with single-phase equipments, the former of which carries 20 tons per motor and the latter 23 tons. Figs. 6 and 7 show the corresponding motor characteristics. It is evident that the great difference between the systems is in the motor loss, which amounts to 6.75 per cent. of the input in the direct-current case and 17.8 per cent. in the single-phase case. Although both motors are dynamically capable of doing the work, the former alone has sufficient thermal capacity for the purpose, the single-phase motor having to dissipate three times as much energy as it can get rid of by natural cooling.

While it is, perhaps, gratifying to the electrical engineer to reflect that he can propose means fully competent to work the whole railway traffic of this country and can offer three distinct methods of doing it, it is futile, as things are, to regard the problem of electrification from any other point of view than that of the ultimate commercial advantage of the railway companies. The author's investigations have convinced him that in England, at least, the commercial advantage is only demonstrable in general in the case of heavy suburban service. Even here it should not be taken for granted, for the margin in favor of electrification is by no means overwhelming. This much can be affirmed, however, that where there is any considerable suburban traffic, an investigation of the cost of electrification and the economic results of providing improved facilities will repay the trouble and expense, and in many cases, justify proceeding with the conversion.

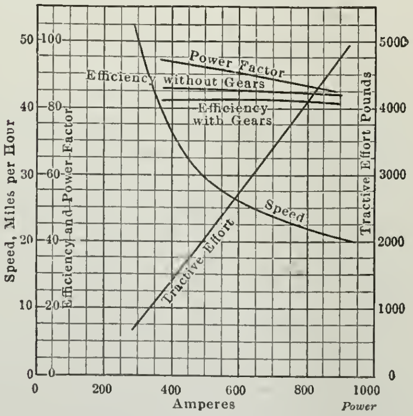


FIG. 7. SINGLE-PHASE MOTOR CHARACTERISTICS

The above remarks are intended to apply to the electrification of existing steam-operated lines. The case of entirely new railways is different and much more favorable to electrical operation. In laying out a new branch to an existing railway, again, it may be well worth while to consider electrical operation.

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*Figs. 1 to 4 are not shown here.

Economic Results from the Bernon Mills Exhaust Steam Turbine-Generator

In this department last week we published an article by Charles B. Cooke, Jr., describing a method used for electrically locking an exhaust-steam turbine to a reciprocating engine, the exhaust steam of which drives the turbine. Since then we have received the following information regarding the results obtained with the Bernon Mills equipment mentioned in Mr. Cooke's article.

On a weekly run of fifty-eight hours' duration, previous to the installation of the low-pressure turbine, the engine developed an average output of 675 indicated horsepower with a total coal consumption of 116,600 pounds, equivalent to 2.96 pounds per indicated horsepower-hour. The following summary of a test covering an equal period with the low-pressure turbine in operation indicates the striking decrease in fuel consumption that has been effected:

Duration of test, hours	58
Total coal burned, pounds	98,000
Total water evaporated, pounds	917,791
Pounds water evaporated per pound of coal	9.36
Temperature of feed water, Fahrenheit	102
Average throttle pressure, gage	82.25
Average vacuum, inches mercury	28.75
Indicated horsepower of engine:	
Right	224.58
Left	225.40
Total	449.98
Turbine power, equivalent indicated horsepower	486.00
Total power, engine and turbine	935.98
Coal per indicated horsepower-hour	1.80

If, instead of utilizing the exhaust-steam turbine, this increased output had been obtained with the original economy, the total coal consumed would have been in the neighborhood of 161,000 pounds. It is therefore evident that the saving in fuel alone, due to the low-pressure turbine, closely approaches 31½ tons per week, or about 1650 tons per annum.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

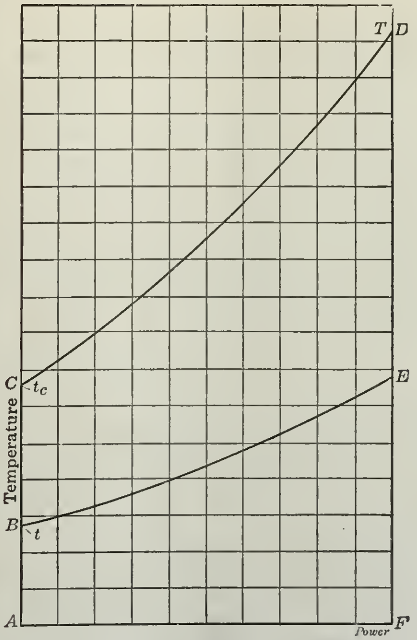
The Gas Turbine

By R. M. Neilson

Prof. Sidney A. Reeve in the issue of Power of July 12 discusses the gas-turbine question and speaks optimistically about its prospects. I have never been pessimistic about the possibilities of the gas turbine, but I have always held and expressed the opinion that an efficient gas turbine cannot be produced until we have an efficient and satisfactory means of compressing air from atmospheric to a high pressure. It is certainly interesting and instructive to compare the different cycles on which a gas turbine could be run; but no matter what cycle is chosen, efficiency cannot, I hold, be obtained without an efficient compressor. There is no sufficiently efficient rotary air compressor at present available, and the employment of a reciprocating compressor of a power great as compared with that of the turbine (and a relatively powerful compressor is essential for efficiency, as will be set forth later on) would take away the greater part of the advantage which a gas turbine might possess over a reciprocating gas engine. The building of an efficient gas turbine is, therefore, in my opinion, an impossibility just now; but there is no obviously insuperable difficulty in the way of producing an efficient rotary compressor. Several persons are working at the subject, and a suitable and sufficiently efficient compressor might be available any day; an efficient gas turbine could then, in my opinion, be built.

I have on several occasions sought to show that an efficient compressor is necessary. As justification for setting forth the reasons again (if justification is necessary), I will try this time to discuss the whole question relating to the factors which limit gas-turbine efficiency more fully than has yet been done, so as to remove any lingering doubt on the subject.

There is a limit to the degree of temperature to which the blades of a turbine can be subjected without rapid erosion, even if cooling arrangements are employed. I have placed the limit at 700 degrees Centigrade (1292 degrees Fahrenheit); Dugald Clerk places the limit considerably lower. It must not be inferred that this limiting temperature is the highest temperature which can be allowed in the combustion chamber or other accessory of the turbine. That is not true, but the blades must not be subjected to a higher temperature. The temperature of the hot products of combustion can be reduced in a divergent nozzle, and the heat energy converted into kinetic energy. Minor difficulties may be encountered in connection with these nozzles; but, in spite of the expressed opinion of Doctor Lucke (whose opinion I nevertheless value), I do not consider these difficulties insuperable. The real difficulty appears when one tries to utilize the kinetic energy in the gas on exit from the nozzle. With a high temperature at admission to the nozzle, the velocity of exit of the gas from the nozzle would be so high that the impact of this gas on a turbine blade would heat the latter above the limiting temperature. It was, I believe, the Hon. C. A. Parsons who first pointed out this difficulty, having previously proved its existence by experiment.



Illustrative Temperature-entropy Diagram

As an example, it may be said that if air at a pressure of 300 pounds absolute and a temperature of 2000 degrees Centigrade were expanded in an ideal nozzle to atmospheric pressure, the gas would issue from the nozzle at a temperature of 693 degrees Centigrade and a velocity of 5290 feet per second. If gas at this high temperature and enormous velocity were allowed to impinge on a turbine bucket, it would heat the latter much above the limiting temperature, even if the bucket could be given a velocity equal to half that of the gas.

It will not remove the difficulty to compound the turbine for velocity, that is, to absorb the energy of the high-velocity gas jet in several wheels or rings of blades. The first set of blades would probably suffer more in such a case, as there would be a greater difference between its velocity and that of the gas jet.

This temperature objection does not in itself alone prevent the building of a satisfactory gas turbine, but it puts a limit to the temperature of the gas when admitted to the nozzle. If, instead of an inlet temperature of 2000 degrees Centigrade as in the foregoing example, a very much lower inlet temperature, say 1000 degrees C. were adopted, the turbine blades could probably be made to withstand rapid erosion. The exact limit to inlet temperature would require to be carefully investigated, but there is no doubt whatever that if this temperature were sufficiently low, no trouble would be experienced with blades constructed of a suitable material.

The efficiency of a gas turbine working on any cycle in which the maximum temperature is low is, however, limited by the efficiency of the pump or compressor, and hence an efficient gas turbine cannot at the present day be constructed.

Proof of the above statement is given in what follows. The whole gas-turbine machine is referred to simply as the "engine"; it comprises a motor which converts the energy in the gas into mechanical work and a pump or compressor which absorbs some of this mechanical work. The meanings of the mathematical symbols are as follows:

$E_i =$ Overall efficiency of ideal engine on any cycle;

$E_a =$ Overall efficiency of actual engine on the same cycle;

$E_r =$ Efficiency ratio $= \frac{E_a}{E_i}$;

$E_m =$ Efficiency of motor alone = ratio of brake work to available energy;

$E_p =$ Efficiency of pump alone = ratio of available energy to brake work;

$W_i =$ Work done by ideal motor;

$W_a =$ Work done by actual motor;

$w_i =$ Work required to drive ideal pump;

w_a = Work required to drive actual pump;

n = Ratio of negative work to gross work in ideal machine $= \frac{w_i}{W_i}$.

Then

$$W_a = E_m W_i$$

and

$$w_a = \frac{w_i}{E_p}$$

Therefore,

$$W_a - w_a = E_m W_i - \frac{w_i}{E_p}$$

Now,

$$E_r = \frac{W_a - w_a}{W_i - w_i}$$

and substituting the equivalent of $W_a - w_a$,

$$E_r = \frac{E_m W_i - \frac{w_i}{E_p}}{W_i - w_i}$$

But $w_i = nW_i$ and substituting this equivalent,

$$E_r = \frac{E_m W_i - \frac{nW_i}{E_p}}{W_i - nW_i}$$

which obviously reduces to

$$E_r = \frac{E_m - \frac{n}{E_p}}{1 - n}$$

By definition,

$$E_a = E_i E_r$$

and substituting the last equivalent of E_r,

$$E_a = E_i \left\{ \frac{E_m - \frac{n}{E_p}}{1 - n} \right\}$$

It will be evident from the final equation that if n has a high value, E_a is greatly influenced by E_p; that is, with a high value of n the overall efficiency of the actual engine is greatly dependent on the pump efficiency, and it is impossible for the engine to have a high overall efficiency if the pump efficiency is low.

For example, suppose that $E_m = 0.8$, $n = 0.4$, and $E_p = 0.5$. Then

$$E_a = E_i \left\{ \frac{0.8 - \frac{0.4}{0.5}}{1 - 0.4} \right\} = 0$$

That is, no work whatever could be got out of the engine, because the pump would absorb all the mechanical work which the motor gave out; and in this example a high motor efficiency has been assumed, namely 0.8. If the value of n had been put at 0.2 instead of 0.4, the result would have been:

$$E_a = E_i \left\{ \frac{0.8 - \frac{0.2}{0.5}}{1 - 0.2} \right\} = E_i \frac{0.4}{0.8} = \frac{E_i}{2}$$

Such a value for E_a would be quite satisfactory if E_i had a high value. For example, if E_i had a value of 0.7, then E_a would be 0.35, which would be very good; and even lower values of E_i would give values of E_a which would allow a gas turbine to compete with reciprocating gas engines in point of economy.

It is therefore plain that with a non-efficient pump a high overall efficiency could be obtained only if E_i were high and n were low. Now, it is easy to show that with a constant-pressure cycle E_i cannot be high unless n is also high.

The accompanying figure is an entropy-temperature or heat-energy diagram in which the ordinates represent temperature and the areas heat energy; C—D and B—E are constant-pressure lines and B—C—D—E—B represents the cycle of a heat engine working on a constant-pressure cycle; A represents the absolute zero of temperature, 273 degrees below the Centigrade zero; t represents the temperature before compression, which will commonly be about atmospheric temperature; t_c represents the temperature after compression and T represents the maximum temperature, all absolute.

With an ideal heat engine working on such a cycle

$$E_i = \frac{t_c - t}{t_c}$$ [*]

and

$$n = \frac{t_c}{T}$$ [*]

The value of t is approximately fixed, being about the temperature of the atmosphere, and the value of T is limited by consideration of the erosion of the turbine blades, as previously discussed. To get a high value for E_i, t_c must be high; to get a low value for n, t_c must be low; it is impossible, therefore, to have a high value for E_i and a low value for n at the same time. Suppose that $t = 290$ and $T = 1300$, absolute temperatures, Centigrade. Then, if $t_c = 870$,

$$E_i = \frac{870 - 290}{870} = 0.6$$

which in itself would be satisfactory; but

$$n = \frac{870}{1300} = 0.67$$

which is impractically high.

If now t_c were made equal to 325, n would be only 0.25, which would allow a gas turbine to do some useful work even with a low pump efficiency; but the value of E_i would be only 0.11, so that E_a would be hopelessly low. It will be obvious on a little consideration that no value of t_c can be found which will be satisfactory as regards both n and E_i.

It may be argued, as has been done by Professor Reeve, that a gas turbine need not run on a constant-pressure cycle. That is so, but matters are no better with a constant-volume cycle, in which there is the same impossibility of obtaining a high value for E_i without a high value of n; in fact, in no cycle in which the maximum temperature is sufficiently low to be practicable can a high value of E_i be obtained without a high value of n. To discuss the different cycles and modifications would involve a repetition of the greater part of the paper which I read before the Institution of Mechanical Engineers (London) in 1904.[†] Moreover, the reader can prove my statement to be correct by a little scheming with entropy-temperature and pressure-volume diagrams.

Desirable Features in a Gas Engine

By L. L. Brewster

No hard and fast rules can be established for the selection of a gas engine, but certain things can be kept in mind and some of them should be indelibly impressed on the memory of any prospective buyer.

While it may not seem so important to the uninitiated, perhaps two of the most important attributes of any gas engine are accessibility and ease of adjustment. This does not mean that gas engines have to be adjusted "every little while" and nursed along to keep them going. It is based on the fact that neglect shows immediate results, which is perhaps an advantage, for it necessitates immediate attention and thereby prevents the "grand smash" which is the cumulative result of persistent neglect. The secret of satisfactory operation in many gas-power plants is the care taken of them, and the easier the job is made, the better it will be done.

For these reasons, it is always advisable to choose an engine in which such parts as the igniter mechanism, valves and pistons can be readily examined, cleaned and renewed, if necessary. A leaky inlet or exhaust valve will cut down economy and decrease power, and where an exhaust valve needs grinding every three months or so it should not be such a heart-rending job as to discourage the engineer into letting it go for six months or a year.

While dirty gas and other unfavorable conditions for which the engine is not accountable are the prime causes of wearing valves, it is found that well cooled and lubricated valves do not show the wear that uncooled valves do. The latter get overheated, warp out of shape, strike unevenly on the seats and consequently wear more in some places than in others.

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[*] For a proof of these equations the reader is referred to a treatise on thermodynamics.

[†] "A Scientific Investigation into the Possibilities of Gas Turbines," by R. M. Neilson. *Proceedings*, Institute of Mechanical Engineers (London), October, 1904.

The purchaser should remember, when buying a gas engine, that all gas engines are very economical, but their economy in regular operation is right up to the engineer, and for this reason inherent economy is a secondary consideration in the choice of an engine. Convenience in handling and care is much more important.

The lubricating system is a matter of much importance, particularly that serving the cylinders and valves. An inclosed crank case with splash lubrication of the piston cannot be called the best method. The inclosed crank case does not permit the operator to observe the operation as he should and I know of some cases where the inclosed case hid the existence of an overheated main bearing until the resulting smoke escaped through the top of the case; then most of the damage was done.

The proper way to oil the piston is to deliver the minimum practical amount of oil and so time its delivery that it goes in on the piston (preferably between the rings) and is not fed at such times that it will be burned up and do no work. Most of the larger horizontal engines lubricate the piston in this manner and it has proved their salvation in some cases. Whenever possible, particularly in large plants or on large engines, copious lubrication should be supplied to all parts which do not come into contact with the flame, and the lubricating system for these parts should be so arranged as to collect, filter and again use the oil.

The cooling system of any gas engine is highly important. It should be positive in operation and all parts coming into contact with burning gas should be well cooled. It should be made easy for the engineer to observe the quantity and temperature of water flowing to each part and most builders provide for this by making the overflows from all parts visible. Even if well provided with cooling water, the cylinder barrel or head, or both, may be so built as to embody thick masses of metal which are very liable to become overheated, unless an abnormal amount of water is circulated over them. These thick lumps of metal are sometimes the cause of hot spots in the cylinder. Hot spots are one of the causes of "back firing" and back firing sometimes shuts down the engine. Uniform thickness of metal surrounding the cylinder and combustion space is therefore preferable.

When comparing engines of different makes and types do not think that the heaviest engine is always the strongest or best. In the first place, compare actual sizes by comparing piston displacements. Ascertain the total net area of the piston or pistons and this will give an accurate basis on which to compare the power which may be expected from each engine. Some engines are rated at a very high piston speed and may have an actual piston displacement much smaller than others of the same rating, though rated at a lower speed. For this reason piston displacements should be figured for a given piston speed.

Do not be deceived by the given *estimated* weights of engines as set forth in the specifications, for these are not actual but estimates, and the buyer is very liable never to weigh his engine after it is installed. A better and fairer way to compare engines is to compare the relative sizes of crank shafts, crank pins, piston rods, and crosshead pins and shoes if the engine has them, and all these should be stated in percentages of the cylinder diameter, for the reason that unless the engines are all of the same type any other comparison is entirely unfair.

For instance, a single-cylinder, single-acting engine of 100 horsepower, say, should have at least four times the piston area of a four-cylinder single-acting engine of the same power. Consequently the size of the parts in the first engine should be much larger than those of corresponding parts in the latter. Most crank shafts are at least 50 per cent. of the cylinder diameter, and in side-crank engines the best are over 60 per cent. Crank pins are usually made 50 per cent. in the side-crank engine.

It is safe to assume that the character of the service governs very largely the choice of type of an engine. It is quite obvious that for driving electric generators, textile machinery or any other service where uniform speed is essential, the engine should give as nearly uniform a crank effort as possible and should, of course, govern well. Uniform crank effort is usually produced by increasing the number of power impulses per revolution; which means the use of several cylinders or the usual combination of double-acting cylinders in the tandem and twin-tandem forms.

Such arrangements, in combination with good governing, will give results which make possible the operation of engine-type alternators in parallel and this service demands about as uniform speed as any.

If uniform speed is not essential and the power is moderate, it is perhaps better to choose a substantial single-cylinder engine in order to reduce the number of parts. Multiplicity of parts increases the liability to derangement and simplicity is a virtue which in many cases is important, particularly when the class of the attendance is not of the best.

While not a part of the engine, the auxiliaries, which must accompany a gas engine of any size, are important factors in the successful operation of the plant. The source of ignition current should be reliable and where continuous operation is imperative it should be provided in duplicate.

The air compressor and tank should be of ample size. The tank should have capacity enough to turn the engine over much more than a few times, for it frequently happens in starting that the proper explosive mixture is found only by repeated trials.

CORRESPONDENCE

Setting Small Gas Engines

A gas engine requires a very solid foundation. Concrete is universally considered to be the best material for this work, but if the engine is not properly set the firmness of the foundation amounts to very little. I have seen and tried most of the common ways of setting gas engines and have evolved a method which, as far as I know, is original. Although I have used it only for gas engines, it will be found of value in setting other heavy machinery.

In setting an engine, I first build a good solid concrete foundation. After it has had about three or four days to dry, I place the engine on it and line up. Then, by means of as many jacks as may be necessary, I lift the engine as high as possible without letting the anchor bolts get out of the holes in the base, taking care to keep the engine in line with the shaft it is to drive.

Lifting the engine as described gives opportunity to fill the holes around the anchor bolts with cement. After this is done, a thick layer of cement is spread on the foundation where the engine base will rest and the engine lowered into it, leaving the engine level and supported by the jacks until the cement has set. The jacks may then be taken out and the nuts screwed down. If the work is carefully done, the engine and foundation will be as solidly united as if they were in one piece and the engine will run as smoothly as though driven by steam.

In making the anchor bolts I join two rods to form a large inverted U and bed them in the foundation. They cannot pull out or turn around and are cheaper to make than any of the usual forms. The bends at the bottom should not be too sharp, as that would increase the tendency to break at that point.

H. K. Wilson.

New Bedford, Mass.

When a gas engine stops suddenly, in nearly every instance a loose or broken connection in the ignition system is the cause. Such stops are also caused, however, by the stoppage of the gasolene supply, as by dirt in the pipe, an air pocket, or water in the carbureter. Lack of lubrication will stop an engine but not suddenly, without warning.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Trouble with a Barometric Condenser

Some years ago I was employed in a large street-railway power plant when they changed over from surface to barometric condensers. When the new condensers were first installed the rule was to obtain the vacuum before starting the engine as we had done with the surface condensers. One holiday afternoon, one of the engines was held in readiness for service; the engine was oiled up and the vacuum started, but no steam was admitted past the throttle. Holiday loads in street-railway service are rather erratic, hence the engine was not required until about two hours after getting the vacuum.

The assistant engineer finally received orders to start, which he did very carefully. The engine had made only about 25 or 30 revolutions when something gave away in the low-pressure cylinder. We did not know at first just what had given way but upon examination found that the low-pressure piston had been pulled from the rod, having received a slug of water at the crank end.

Salt water was used for condensing purposes and as salt water was found in the cylinder it was at once obvious that the damage was caused by the condenser instead of condensation of the steam.

Shortly after this incident I was transferred to another station of the same company where a new vertical cross-compound engine had just been installed. The exhaust pipe from this engine led to the basement, where a feed-water heater was placed in the exhaust line. From the heater the pipe ran to the top of the power-house roof, where the connections were made to the condenser and to the atmospheric exhaust pipe.

Soon after the engine was placed in regular service, we noticed that at times, when opening the throttle with the vacuum already on the condenser, a loud pounding and surging would occur in the exhaust pipe; in fact, it smashed a 40-inch exhaust elbow located in the basement. Here, again, there was salt water in the exhaust pipe, which was evidence of trouble with the condenser.

After this we received orders to start the engines before building up the vacuum and no further trouble was experienced.

My explanation of the trouble is as follows: When starting the condenser the pipe from the condensing cone to hotwell is probably full of water. When admitting steam to the high-pressure cylinder, or even opening the bypass to the low-pressure cylinder, a considerable quantity of air is liable to be forced from the exhaust pipe into the condenser cone and down to the hotwell. The column of water would then be mixed with air, making it much lighter in weight. The small amount of steam necessary for turning over the engine may, by the time it reaches the condenser cone, be condensed, and represent a tenuity greater than the column of water and air is able to overcome.

A. S. Smith.

Boston, Mass.

Results with a Homemade Planimeter

The accompanying diagram was taken from a 12x30-inch Corliss engine running at 90 revolutions per minute with a steam pressure of 95 pounds gage. The scale of the indicator spring was 60 pounds to the inch.

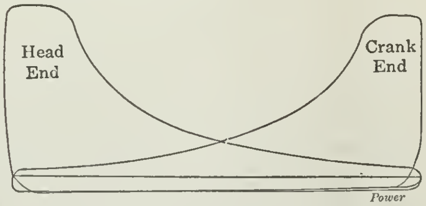


Diagram Used in Comparing Planimeters

The areas were found by three methods; first, by using an Amsler polar planimeter; second, by a method of approximation by dividing the card into a number of rectangles; third, by using a homemade planimeter, similar to the one described in the July 12 issue of Power. The three methods were used for the purpose of comparison and to show the practical value of the homemade planimeter in figuring the indicated horsepower of an engine. The results were as follows:

Method of Finding Area of Card.	Area Head End, Square Inches.	Area Crank End, Square Inches.	Length of Card, Inches.	Head End M.E.P., Pounds per Square Inch.	Head End I.H.P.	Crank End M.E.P., Pounds per Square Inch.	Crank End, I.H.P.
Amsler Polar Planimeter.	2.85	2.72	3.937	43.5	33.6	41.5	32.05
Approximate Method . . .	2.864	2.735	3.937	43.9	33.9	41.8	32.4
Homemade Planimeter. .	2.8	2.6	3.937	42.8	33.2	39.8	30.8

Total indicated horsepower were 65.65, with the Amsler polar planimeter; 66.3, with the approximate method; 64, with the homemade planimeter.

Assuming that the value found with the Amsler planimeter for the indicated horsepower is correct, the errors involved by using the other two methods were found in the following manner:

(66.3 — 65.65) ÷ 65.65 = 0.99 per cent. Error involved by use of the approximate method,

(66.65 — 64) ÷ 65.65 = 2.5 per cent. error involved by using the homemade planimeter.

J. E. Rock.

Lansing, Mich.

Clogged Radiator

In a large building heated by a single-pipe steam-heating system it was the practice each spring to remove a large radiator in the main corridor and store it until the following autumn. For several seasons the weather had been so mild it had not been needed but on a cold day during the past winter a call came for the radiator and it was hurriedly installed. When turned on it began filling with water and pounded vigorously. The valve was examined and found to be in good condition. Several kinds of air vents were tried and a special drip from the heel of the riser to the radiator was installed. The trouble still continued and, furthermore, an adjoining radiator began to act in a similar manner. The radiator had been turned on end before installing to remove any dirt which might have found its way into it.

It was disconnected for internal inspection and with the water that flowed out there were a large number of walnuts and an assortment of materials used by squirrels for building nests. The nuts had acted as a check valve to prevent the condensation from flowing out, and several which found their way into the valve on the branch line had acted in a similar manner toward the other radiator.

Lewis C. Reynolds.

Willard, N. Y.

License Laws

Engineers in Massachusetts feel a certain amount of pride in their license laws, yet no one knows better than they that there is room for improvement. In looking over the provisions of the proposed New York State license law, there are found some features that seem to be an advance in the right direction. One is the limitation of special licenses to thirty days; this is an excellent idea. The special license has always been a weak point in the Massachusetts law. No matter how many men holding regular licenses are out of work, a manufacturer may pick up a cheap man and help him to get a license, which license may be renewed without limit. The examiner is supposed to see to it that the employment of the man is a safe proposition, but sometimes influence is brought to bear and the examination is very easy. There is no basis for an argument for the special-license provision on the ground of safety; it merely suits a number of manufacturers who are looking for cheap men, and engineers who are too lazy to do the work required for obtaining a regular license. Some restrictions have been put upon this abuse, but Massachusetts should copy the New York clause on special licenses.

On the other hand, there are features in the New York bill that do not seem good. License laws are in the interest of public safety. If they were for the purpose of raising engineers' wages, there might be some excuse for taxing the engineers to maintain the expenses of the examiners' department; but this is not the purpose, and I doubt if it has any such effect. The public, which is protected, should pay the bills. In this State a fee of $1 is charged for examination. Once a license is secured, there is no further taxation and all renewals are free. New York proposes to tax the engineer $3 for every examination and $2 a year for renewals.

Two dollars from every engineer in the State ought to produce a large annual income, which would make the examining department a paying proposition. But safety, not profits, should be its aim. The engineer, who rises soon after the boss goes to bed, and goes over to the factory to warm it on a winter's morning; who is the last to leave the factory at night; who must be posted on all things in order to get his license; and who for all this receives but a little more than half that which a printer receives for eight hour's work; he is to be made the "goat" once more, and must stand for not only the price of the operation of the new safety measure, but just a little more. Engineers in other States have meekly submitted to such treatment. Here is a chance for New York engineers to protest.

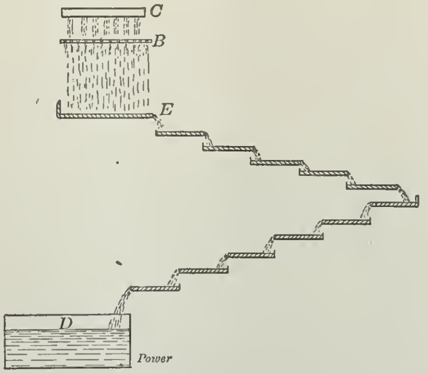
The provision that a man must have had three years' experience as fireman before he can apply for a fireman's license, looks like trying to lift one's self with his own boot straps. How will it work out? Is it necessary in order to insure safety? If a man cannot learn enough around a boiler plant in less than three years to make his employment as a fireman a safe proposition, he must be pretty stupid. Some men learn very quickly. We want these bright men. But such a law will tend to drive them elsewhere. It would seem that the Massachusetts plan of leaving the amount of necessary experience to the judgement of the examiner is more just and more practical.

William E. Dixon.

Hudson, Mass.

A Simple Cooling Tower

The illustration shows a simple form of cooling tower that is adapted to small plants. It can be built within a limited space and at a small cost. The large radiating surfaces exposed to the free circulation of air makes it exceedingly efficient. It is constructed of planks but, unlike many types of cooling towers, it is built in steps. The current of water is first broken up by falling through holes in the bottom of basin *C* and through the wire gauze *B*. It then falls 6 or 8 feet to the platform *E* and gradually finds its way over the various steps to the hotwell *D*. The steps must be true and level and should be placed far enough apart to allow a free circulation of air past the falling water. The hotwell should be large and shallow so as to allow sufficient radiating surface; a groove will answer this purpose very well.



Series of Steps for Cooling the Water

In any cooling tower the main object is to bring the water in contact with the atmosphere in particles as small as possible. Many modern cooling towers are now using spray heads through which the water is forced under pressure; but this requires considerable power, and where there is room a simple tower of the type herein described will be found advantageous.

C. R. McGahey.

Sheffield, Ala.

Repairing a Main Bearing

We had a modern Corliss engine direct connected to a 250-kilowatt, three-phase generator on which the main bearing had twice been allowed to get so hot that the babbitt had started to flow. Each time the quarter boxes had been removed and dressed but the bottom bearing, owing to the difficulty of removal, had not been touched. As a result the bearing always ran so hot that it required continual attention to keep it from burning, hence it was decided that the bottom bearing must come out.

The local machinist claimed that such an operation would take several days and would cost about $50. The conditions of the service were such that this machine could be shut down only on a Sunday; therefore I determined to attempt to accomplish the task during this limited time with the aid of the chief engineer and two assistants.

The armature was slid along so as to clear the field and the cap was loosened on the outer bearing. After removing the cap, the wedge which is used for adjusting the hight of the bearing, was pushed in to its limit and blocking placed under the crank disk. Then the trouble began. The adjusting wedge was designed to be pushed in by means of a large screw, provided with a head for drawing the wedge back again; but when we attempted to draw the wedge back it was found that the head had been broken off and there was no room to get anything behind the wedge to drive it back. This had us puzzled for some time, but we finally conceived the idea of placing the engine on one center, loosening the quarter blocks to give about ½-inch play after putting in a strip of wood to take the shock, and then working the shaft back and forth by admitting steam to alternate ends of the cylinder. This loosened the wedge sufficiently to enable us to push it all the way back.

Next, iron plates were placed between the wedge and the bearing to make up for the amount we had raised the shaft and the wedge was again screwed in and blocking placed under the disk. We repeated the operation until the shaft was raised sufficiently to permit the bottom block to be turned around to the top of the shaft and taken out. The babbitt was found to be quite smooth but all the oil grooves had been completely filled with the metal. It was a short job to cut new oil grooves and put the blocks back into position. The shaft was then lowered by reversing the process by which it had been raised.

Everything was in place and the engine ready to run by 9 o'clock that same evening and the total expense amounted only to the wages of the two extra men for the day.

G. E. Miles.

Salida, Colo.

Should One or Two Boilers Be Installed?

At the present time there is being built an addition to one of our school buildings. The architect's plans call for the heating to be done by one large boiler. There developed some opposition to this on the part of certain members of the board of education; they believed that two smaller boilers would be better as one could be cut out in mild weather and in the spring and fall, thereby effecting a considerable saving in fuel. I agreed with their views but we were overruled by those in favor of one boiler. I would be pleased to have the opinions of POWER readers as to which is the better plan.

O. LINDBLADE.

Sioux City, Ia.

Useful Belt Drives

The accompanying illustrations show arrangements of belt drive that do away with cross belts which are often a nuisance. In Fig. 1 the pulleys *B* and *C* are driven in opposite directions by using the idler *D*. This method of belting is often used to great advantage in operating sawmill carriages, for by using a friction pulley in connection with the shafts driven by the pulleys *B* and *C* the carriage is driven forward by one shaft and reversed by the other.

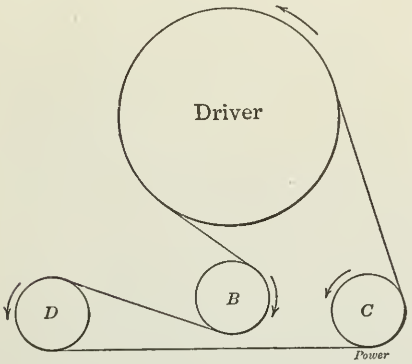


FIG. 1. DRIVEN PULLEYS ROTATING IN OPPOSITE DIRECTIONS

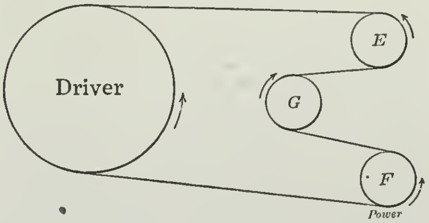


FIG. 2. DRIVER AND DRIVEN PULLEY ROTATING IN OPPOSITE DIRECTIONS

Fig. 2 is a variation of this principle. By arranging *E* and *F* to run as idlers and the idler to act as the driven pulley the latter will run in the opposite direction to that of the driver. In both drives herein described the belt is used on both sides.

The foregoing suggestions were embodied in an article by H. N. Saxton, published a few years ago in POWER. They are so simple and useful as to bear repeating.

JAMES E. NOBLE.

Toronto, Can.

The Care of Pistons

To obtain the best economy from the steam, special attention should be given to the piston. It must be properly lubricated to avoid cutting the cylinder, and must be kept central with the rings and tight to avoid leakage of steam. In fact, it is the part of the engine which should receive the most frequent attention in order to keep it in the best possible condition, and the time spent will yield a better return in dollars and cents than if given to any other part of the engine.

There are nearly as many different styles of pistons as there are of engines. Years ago, when engines were not so highly perfected, the piston consisted of a simple plunger packed with a gasket made of braids of rope yarn tapering for about 18 inches at each end and long enough to go around the piston with the tapers overlapping. The various layers of braids were placed so that the tapers occurred on alternate sides of the piston. Melted tallow was applied with each layer and beaten down with a wooden plug and hammer until the depth of the piston was reached, after which the junk ring was screwed down and more tallow applied. In a later type of piston a single split ring was adopted. The back of this ring was ground or filed to allow for the usual hemp packing behind the ring; the object was for the ring to take up the wear and the hemp and tallow to keep the piston steam tight.

The next improvement was the adoption of a babbitted piston. The piston was made solid and had a deep groove around its circumference. Holes were drilled in the face of the piston connecting with the bottom of the groove. Babbitt was then pounded in till the groove was full; after it was cooled the piston was turned to fit the cylinder tightly. When these pistons became worn the adjustment was made by driving a drift pin alternately in the holes, forcing the babbitt out to the walls of the cylinder. Great care and good judgment had to be used in doing this in order to force it equally at all points, and yet not force out enough babbitt to cause undue friction.

A solid piston having a single split ring in the center was next used, and gradually more rings were introduced and also an expansion ring which assisted in keeping the steam rings tight against the walls of the cylinder. Then the sectional variety was introduced, which consisted of rings in sections, dovetailed into each other and held in place by circular rings or springs. These worked very well, but in all cases it was necessary for the pistons to receive good care in order that they might work well, and the man who points with pride to his engine cylinders and states that they have not been opened for years is not giving his engine the care it should receive.

CHARLES H. TAYLOR.

Bridgeport, Conn.

Equivalent Evaporation of Steam Boilers

A boiler horsepower is expressed by the evaporation of 30 pounds of feed water per hour at 100 degrees Fahrenheit into steam at 70 pounds per square inch gage. To compare boiler capacities it is usual to reduce both the standard and the given evaporation to an equivalent evaporation of feed water at 212 degrees Fahrenheit into steam at atmospheric pressure (212 degrees Fahrenheit). This is commonly called "equivalent evaporation from and at 212 degrees."

Equivalent evaporation is obtained by

1. Subtracting the heat units in one pound of water at the feed-water temperature from the total heat units in one pound of steam at the given pressure.
2. Dividing the result by 970.4.
3. Multiplying the quotient by the number of pounds of water evaporated under the given conditions.

The product equals the number of pounds of water evaporated from and at 212 degrees Fahrenheit.

Let

H = Heat units in one pound of steam at the given pressure,

H_1 = Heat units in one pound of feed water at the given temperature reckoned above 32 degrees Fahrenheit,

L = Latent heat of steam at atmospheric pressure = 970.4,

W = Pounds of water evaporated per hour under given conditions,

$E\,E$ = Equivalent evaporation in pounds of water per hour from and at 212 degrees Fahrenheit.

Then,

$$E\,E = \frac{H - H_1}{970.4} \times W$$

An example may serve to illustrate this more clearly.

Given the evaporation of 30 pounds of water at 100 degrees Fahrenheit into steam at 70 pounds gage pressure. Find the equivalent evaporation from and at 212 degrees Fahrenheit.

$E\,E = (H - H_1)\ W \div 970.4,$

$= [1179.9 - (100 - 32)] \times 30 \div 970.4,$

$= 34$ pounds of water per hour evaporated from and at 212 degrees Fahrenheit.

W. VINCENT TREEBY.

London, Eng.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Settings of Return Tubular Boilers

The editorial in the issue of August 2 on "Settings of Return Tubular Boilers" calls attention to "a weak point in boiler practice." It is a well known fact that "the side walls and rear connections" often do develop cracks within a short time after the boiler is put into use. It is obvious that there must be a reason for this. In small plants where boilers of this type are largely employed, as in "sugar-houses" on the sugar plantations of Louisiana for instance, the "engineer" in charge thinks nothing of having the boiler settings "done over" at the approach of each grinding season. He has the time to do the repair work, as the boilers are not in use for more than three to five months of the year.

However, it becomes a different thing for the man who has only one boiler, and that boiler has to be kept hot every day of the year. The editorial referred to above recalls to my mind a good example of bad setting. The case I have in mind is that of a large crack running from top to bottom, across the whole side wall, and a little toward the rear, in the brick setting of a small return tubular boiler. The boiler was used to run a machine shop during the day, while the fire was banked at night.

The plant had been in operation only three months, and the whole side wall threatened to come down. This was due, no doubt, to the fact that the setting was built by a "local brick mason," who did not allow for longitudinal expansion. The only way in which to avoid such things is for the makers of return tubular boilers to do as some makers of water-tube boilers do, put in the settings themselves.

J. E. Poche.

New Orleans, La.

Packing a Feed Pump

In Power for August 9, Mr. Kramer asks for information regarding packing for boiler-feed pumps handling water of a temperature ranging from 100 to 195 degrees Fahrenheit. I think that the trouble is not due to the packing or the method of applying it but to the wide range of temperature. I have had the same trouble with a piston-packed pump. My cure, which was effective, was to use one pump when the engine was running and the feed water was at about 200 degrees, and another pump at night and on Sunday when the feed water was practically cold. Before this arrangement was put into effect the packing would last only about five or six weeks. Since that time one packing is good for several months.

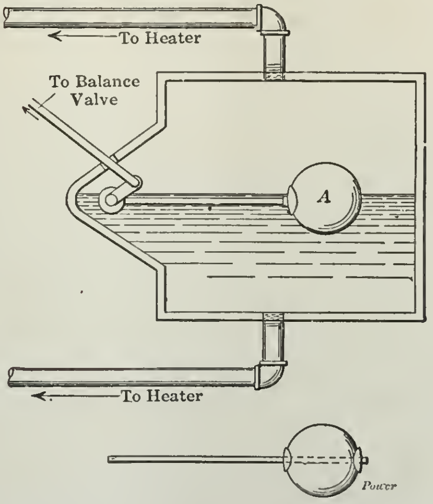
R. L. Bevis.

Terre Haute, Ind.

Leaky Floats

In the August 16 issue, Mr. Granfield does not state whether the leaky float has any attachment to it or whether it is independent. Nor does he say whether the leaks are caused by corrosion. His troubles recall to my mind the trouble I experienced some time ago with a float.

The float was used to regulate the water level in an open heater through a series of levers operating a balanced valve which admitted the water fed to the heater. The connection of the stem to the float was as shown in the upper part of the illustration, that is, the end of the stem was threaded and screwed into a collar which in turn was soldered to the float. The leak always occurred at the soldering joint as shown at *A*. No sooner had I repaired the float than the trouble would repeat itself.



Original and Improved Connections of Float and Rod

Eventually, the idea came to my mind that greater strength would be obtained by passing the stem through the ball and making two joints. This I did, with a stem of sufficient length, as shown in the lower portion of the figure. I never experienced any trouble thereafter.

Luke Marier.

Fall River, Mass.

Central Station versus Isolated Plant

Continuing the discussion of the topic "Central Station versus Isolated Plant," I should like to answer the letter by Warren B. Lewis, in the August 9 issue. He states that electric current is a commodity which is made and sold on much the same basis as any other commodity. I grant that it mav be made on the same basis, in other words, by firms in their own building and with their own machinery, but how is it sold? Does not the public allow these companies the right to run their wires through the streets, either underground or overhead as the case may be, continually tearing up the roads to renew their conduit, etc.? In other words, after that current has left the power house it should belong to the public and it does belong to the public. It comes under the head of a Public Service Corporation and under this head it has no right to discriminate. Because I ride up and down town in the street car twenty times a day I am not allowed a lower rate for my fare than the man who may go up and down once a day; because I burn 2000 cubic feet of gas per month, I do not have to pay more than the man who burns 2,000,000 cubic feet a month.

The gas companies of New York City have been forced to make an 80-cent rate by the Public Service Commission. Is there any reason why the electric company should be allowed to go "scott free?" Even I believe that under certain conditions the central-station company should be allowed to make a sliding scale, but these conditions are:

That they charge a certain fixed sum per kilowatt of maximum demand. Take, for instance, the central station located in a busy portion of the city and let it be loaded to its full limit; let it take on some large building, such, for instance, as the City Investing building. It certainly would have to increase its plant to a great degree to supply the current for it, and yet this building would be able to get a cheaper rate, because of its tremendous demand, than the little cigar store on the corner that would not require a single bit of increase in plant installation. The little fellow has to make up this discrepancy caused by the large sum spent on the increased installation for the big plant. You certainly cannot call this legitimate and fair.

Regarding a cost of three-fourths of a cent per kilowatt-hour this, of course, is ridiculous on the face of it, but with careful operation under the same conditions that the central station has regarding the distance from fuel, etc., even the small isolated plant running anywhere near its full load can certainly make electric current cheaper than it can be bought from the central station, unless the central station is lucky enough to have water power. I know one firm in this city which will guarantee the maximum fuel cost of two-thirds of a cent per kilowatt-hour, under the condition that they are allowed to install the plant and operate it as they wish.

One of the things that I decry greatly regarding the argument of the central station is, that the interest and depreciation on a plant are figured at 10 or even 15 per cent. on the investment, while the Edison company, of New York City, in its report to the Public Service Commission figures on less than 1 per cent. depreciation. This is as much too low as its argument is too high, but it is only another case of central-station exaggeration.

Mr. Lewis is perfectly right in saying that the purchaser has the privilege of going out and employing a consulting engineer. Under certain conditions it is probably more advisable, although perhaps more expensive, to use the central-station current, such as in apartment houses and places of that kind. But he will often run in contact with consulting engineers who are on the quiet employed by the central-station company and who will advise using central-station current whether it is really advisable or not. Let the central station avoid discriminating in rates and it will have as large although perhaps a somewhat different field as it has with its discrimination or, at least, the returns will be as great.

ROSSITER HOLBROOK.

New York City.

Pumping Layout for Criticism

In looking over the August 9 issue of Power I noticed H. B. Fillmen's pumping layout which is submitted for criticism. As he wants air-lift information, I will try to help him by describing the system which I am operating. The compressor is of the vertical duplex belt-driven type. Its cylinders are 7x9 inches in size. It was built to run at 190 revolutions per minute and is guaranteed to lift 50 gallons of water per minute with a total lift of 150 feet. We run the compressor at a speed of 140 revolutions per minute and lift water 110 feet. The compressor handles 5000 gallons per hour under these conditions. The discharge pipes in the wells are 2½ inches in diameter; the air pipe is 1 inch in diameter. The small diameter of the air nozzles is about ¼ inch. The submergence is 60 per cent. of the total lift.

Based on the above figures, Mr. Fillmen's outfit would require a compressor having 205 cubic feet per minute displacement and 2½-inch discharge pipes from each well. The pipes should be independent of each other and go over the top of the tank with as few bends as possible.

The submergence should never be less than 60 per cent. of the total lift. The gage pressure is proportional to the weight of the water in the discharge pipe; but as there are alternate layers of air and water, it is hard to guess what the pressure actually is. With a lift of not over 125 feet his compressor would consume about 20 horsepower. My compressor makes a good load for a 7-horsepower motor.

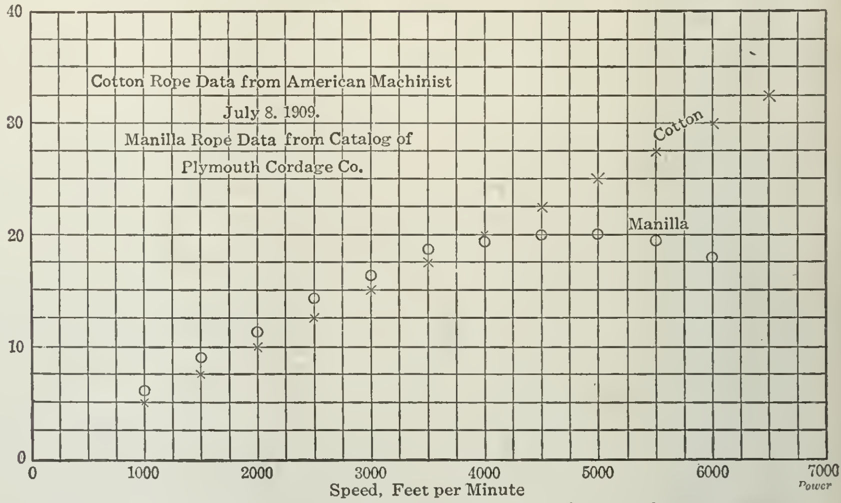
R. E. HENSLEY.

Hico, Texas.

Design of Rope Drives

In the issue of August 16, the article by Francis H. Davies, on "The Design of Rope Drives," interested me greatly.

I find that while Mr. Davies refers in the last part of his article to the fact that cotton rope is to be preferred, he has laid out his plot, Fig. 1, with data for manila rope, and limits the speed of a 1¼-inch rope to practically 4500 feet per minute. This, as will be observed by referring to the accompanying figure, does apply to manila rope, but Edward Kenyon quotes very much different results for cotton rope, showing that the effect of centrifugal force is very much less for cotton rope than for manila, so that the power of cotton rope very closely approximates the theoretical value up to speeds of 7000 feet per minute. This is due, no doubt, to the softness and the tendency to cling which exist in cotton rope, and which do not exist in manila rope.



EFFECT OF CENTRIFUGAL FORCE ON CAPACITY OF 1¼-INCH COTTON AND MANILA ROPES

I write this to call Mr. Davies' attention to this fact and to ask him if he has not used manila-rope data in his plot rather than cotton.

HENRY D. JACKSON

Boston, Mass.

Depreciation

In reading W. H. Bryan's letter on the above subject in Power for August 9, I noticed some points open to criticism. His remarks on appreciation of real estate seem to be based on the assumption that depreciation will be written against both land and plant. I believe that it is the almost universal practice to exclude the value of the real estate from any depreciation charge; indeed, there are certain firms which write depreciation against the plant less the appreciation on real estate, although most large corporations carry this appreciation as a sort of "secret reserve," unless the difference between book and actual value becomes very marked.

In favor of the fact that the method of charging off a fixed percentage of the previous year's book value never completely wipes out the account, it may be urged that no plant ever becomes entirely valueless, as any machinery, or even old buildings, have a certain scrap value.

There is this objection to a "depreciation fund," in order to make any real difference between this and the method of writing off a fixed amount each year, say 5 per cent. on a 20-year basis (which amount is debited "Operating," and credited "Reserve for Depreciation"), the amount carried in the "depreciation fund" must be put into high-grade bonds, or other similar securities, and these securities held inviolable. This may tie up much-needed funds at very inconvenient times.

A vast deal of the depreciation written

off is what has been termed "depreciation due to advance in art," that is, the early consigning to the scrap heap of good machinery to make way for that which will operate more cheaply. As an instance of the slow depreciation in well kept machinery I remember visiting an iron foundry in 1904 in which a single-acting steam engine was driving a fan blower. The owner told me that to his knowledge the engine had been at work 38 years, and it was old when he first saw it. This shows that what seems to be a popular depreciation period, 20 years, could often be exceeded as far as wear on the machinery is concerned.

DONALD M. LIDDELL.

Grasselli, Ind.

Effect of Scale

In the brief tests reported in the issue of July 26, a reduction of nearly 20 per cent. in fuel cost is attributed to removing scale from the boiler; but it is clear that if the temperature of combustion had been constant, the difference would have been very much less. With natural draft, clean tubes account for a difference of only 35 degrees Fahrenheit in the flue gases, and with forced draft, only 99 degrees Fahrenheit. If the combustion temperature were 2000 degrees, the fuel saving would be 1¾ and 5 per cent. respectively, due to the removal of scale.

The presence or absence of scale inside the tubes could make no difference to draft or firing efficiency; but both tests before scale removal show bad firing, and both tests after scale removal show careful firing, as shown below.

	Ashpit Air Pressure. Inches.	Pounds Coal per Square Foot of Grates.	Equivalent Evaporation per Pound Coal. Pounds.
NATURAL DRAFT:			
Before scaling....	0.24	11.6	6.12
After scaling.....	0.35	15.6	7.56
FORCED DRAFT:			
Before scaling....	1.10	19.7	6.47
After scaling.....	0.85	22.4	7.25

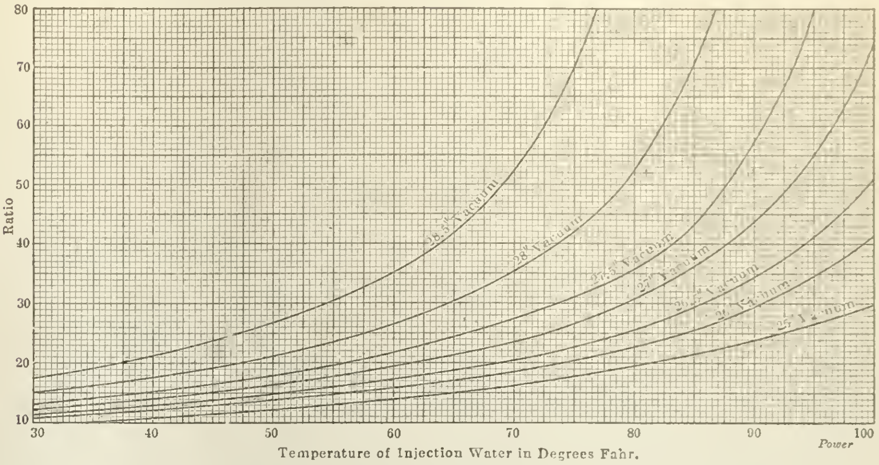
Note that less draft was used to burn 22.4 pounds of coal than 19.7 pounds. The obvious conclusion is that the correct control of the firing and the draft has a far greater influence on fuel bills than scale can have.

CHARLES ERITH.

London, England.

Condenser Economy

In connection with the article on condenser economy in the August 16 issue, the accompanying curves may be of interest to POWER readers as they show the amount of water required for condensing under various conditions. It is usual to allow a difference of about 5 degrees between the temperature due to the vacuum and that of the mixture, and add this difference to the injection temperature to secure the correct ratio of condensing water. For instance, if a 28-inch vacuum is desired with 70-degree water, the ratio will be about 42 to 1.



CURVES SHOWING THEORETICAL RATIO OF COOLING WATER REQUIRED AND CONDENSED STEAM

While the discharge temperature should be kept as high as possible, the tendency of most engineers is to use excess water to take care of the maximum load. This produces a lower discharge temperature than is desirable. With steady load, however, this temperature can be carried at the maximum.

W. G. STARKWEATHER.

Boston, Mass.

Pump Problem

I desire to offer a solution to the problem presented in the issue of August 9, by W. E. Thompson, concerning the boiler pressure necessary to operate a 10 and 16 x 8 x 18-inch compound feed pump working against 185 pounds pressure when running at 35 strokes per minute.

In an installation where the pump is set close to the boiler which it supplies and is conncted thereto by a well-proportioned steam pipe, the number of strokes per minute depends more on the amount the throttle valve is open than on the boiler pressure. With a long run of uncovered pipe a considerable drop in pressure might take place due to condensation.

As steam is taken full stroke, the secondary pressure bears the same relation to the primary as do the respective cylinder volumes to each other, and we are therefore only concerned with the pressure in the primary cylinder.

It is current practice in designing a feed pump to make the steam-piston area two or three times that of the water piston. The average practice varies between 225 and 233 per cent. This compensates for all losses through condensation, friction, etc.

Referring to the problem, the area of the water piston is 50.26 square inches, and this with the pressure of 185 pounds per square inch makes a total of 9329 pounds to work against. The area of the high-pressure piston is 78.54 square inches. Dividing this figure into 9329 the quotient is 118.5. Then with 118.5 pounds steam pressure the pump would be in equilibrium. Increasing the pressure 25 per cent. gives 148 pounds, which is the minimum pressure required in the high-pressure cylinder to insure satisfactory operation.

The pressure in the secondary cylinder may be determined by the following proportion, neglecting loss by condensation, etc.:

H.P.Cyl. vol. : L.P.Cyl. vol. ::
L.P. : H.P.

Hence,

$$100 : 256 : : x : 148$$

$$x = 57.8$$

The pressure, therefore, in the secondary cylinder, with conditions as above mentioned, will be 57.8 pounds.

TRUMANS D. HAYS.

Cambridge, Mass.

Engineer's Wages

In recent numbers of POWER I have read what some have to say about the engineer and his wages. Possibly to get straight on this important subject we should try to distinguish the *engineer* and the *stopper and starter*. Salary counts for nothing. I have seen good engineers who were working for $45 per month apparently satisfied, while I have run across a lot of stoppers and starters who were receiving $85 per month and who positively could not keep the piston in the middle of the cylinder. The engine would either knock itself hot or the engineer would key it up so tight that it would melt out the bearings and cause a shut down. There is a wider variation among engineers as a class than among the members of any other craft. Lots of fellows will use $45 worth of oil per month; they put it on so freely that one has to lay boards around the engine foundation so as to be able to get close enough to work on the engine. Other men can be found who will run a large plant on $45 worth of oil per year. This may sound like a fish story, but the facts are drawn from actual cases. In some plants not a spot of oil can be found on the floor, not a spot of rust can be seen and no steam liberated by defective packing can be discovered. The machinery shows a better finish than when it left the shops, due to the engineer's constant industry and pride. This man gets $150 per month. He is an *engineer* not a *stopper and starter*. He can tell when a piston is loose in the pump cylinder. He will not draw up one side of a pump gland so much tighter than the other as to cause the pump to hang.

The wages count for nothing.

We cannot all be the president; but if a man amounts to something and puts his ability on the market in the proper way, he will receive a fair price for it.

The card of a union will not make an engineer. I wager that a union would make the conditions worse than they now are. Let us get right at this matter. If I remember correctly, the N. A. S. E. was formed for a very good purpose. But what has happened? Almost anyone can get into the N. A. S. E. What is the result? Some of those who should wear a button in the lapel of their coat are somewhat ashamed to be a member. There is no use in dodging the truth of this. Let us correct matters. Let a man know something. A school mistress is compelled to have some knowledge before she may teach.

Some men with very little ability get responsible positions. Some one will vouch for the applicant, possibly over the telephone, not knowing anything of the fitness of the man. In he goes. If someone were to ask him to make a Corliss valve work tighter by increasing the compression, he would be lost.

And now, some men are short-sighted enough to say, let us have a "union." Worse than ever, unless all post signs change. This would mean that a good man would have to work for the wages of a poor man, in other words, equal wages. Let us work along some better line. Let us have an examining board which cannot be bought, to see how much a so-called engineer knows. Determine whether he is a two-dollar man or a four-dollar man, then classify him. In many small plants where one man both fires the boiler and looks after the engine, all the valves are bright and clean, the piping is well kept up and no leaks are evident. The engineer often is a $45 man. The job may not be worth any more than this. He is contented, for he has his home. But if he wanted to go out for a better job he could hold it.

Let us stand on our merits not on unionism. Consider some of the N. A. S. E. meetings; lots of good can come from these, but often one gets the floor who knows less than any one else and the posted man goes away in disgust. It would be better to let the best men make the addresses and let the so-called engineers ask the questions "to learn." Do not try to get on top before the foundation is built; you may fall. It is not a matter of what we want to be but what we are. Many so-called engineers working for two dollars per day are not worth it.

I am "from Missouri," when it comes to a union for education I must be shown.

C. R. McGahey.

Sheffield, Ala.

One of Watt's Letters

In the August 16 issue we reproduced a copy of a letter by James Watt to Matthew Boulton. We have received numerous requests to print a "translation," so that with it's aid the text of the letter may be more easily followed.

Our interpretation of the letter is given below. Some of the words, due to the impaired condition of the original manuscript or to cramped penmanship, cannot easily be deciphered. Those of which we are uncertain are set in italics.

"Birmingham, May 10, 1777.

Dear Sir:

Yours of the 8th before me—Salmons man is making wheel engine boiler and seems to do tolerably well.

I am astonished how Joseph deceived himself and us in his *effect* of Bow boiler. I have not seen him since—but I remember he told me that there was a prodigious odds in the quality of some of their coals by others. Did you say whether pushing in the damper to a certain degree did not produce a better or as good an effect as opening the hole in chimney—and was you absolutely certain that no feed got into the boiler during time of experiment?

I think that you are in a good train at Shadwell and recollect no more cautions on that head.

I have made the best apology I could to Mrs. B. though she talks the words of wrath; she is in very good humor and all are well.

When I see the table I can reason upon *copper baylers*.

I saw the Battering Ram, or devil incarnate, go today above 60 strokes per minute and work its own regulators. All the *Hampers* were assembled. They thanked God that Webb could not *make* feeders in. He told them he had one of these and on hand — Moore wants a score of large ones for his own use. The story is all over Birmingham and I expect we shall have customers by the dozen. I imagine that they may be made to work forges and tilts fast enough for any purpose.

Chesburg Iron has turned out damned bad, his piston rod for fuel *bupey* was very well forged by Dixon and upon heating it to float it fell in two at the shooting and Jop rejoiced in his heart thereat, but not Jop nor Joseph could weld it again—nor would it weld to any other iron by no trick.

Dixon is fagotting one out of Sweedish small bars.

The branch of Battering Rams may turn out very *cursed*, as I dare say nobody will attempt them with common engines, and I don't know if they could be made to do. I have invented an admirable thing for opening the regulator which acts by a spring and does it quicker than thought. All Webb's fears are that when a larger one goes at the rate that his does that no body will come near it.

It has demolished all the fixtures many times already, and I suppose must be wholly made of cast iron.

Dangerfield has been here and sett Jop to growling and drinking for two days this week.

I should make myself easy about profits, if any were coming in at all, but a total stagnation as has been hitherto cannot do at all—I am clear that you should make your bargain sure before you leave London, and lett us know what we are to gett with some probably, at least.

As to the Cornish affair. It has also struck me that I should go first and that something should be done before you come.

Adieu.

I wish you a clear head and a firm heart on Tuesday. Pray weigh the coals and observe the quality. Keep well with Rothwell. I think he deserves it.

Yours,
J. Watt."

Diagrams for Criticism

In answer to J. D. Robertson's inquiry in regard to indicator diagrams in the August 16 issue, if the steam pressure at the throttle is 150 pounds there must be an unreasonable amount of wire drawing in the steam ports. However, I think that the 150 pounds must be at the boiler because with a scale of 60, the pressure at the point of admission is only 95 pounds.

I would advise that a steam-pipe diagram be taken. To do this, tap a hole in the steam pipe above the throttle and place a connection in it for the indicator; hook the cord up to the crosshead in the usual way and take a diagram. If the drop in pressure shows up here then the steam pipe or header is too small; if not, then the fault is in the steam ports, leaky valves or excessive clearance.

William H. Magee.

Bayonne, N. J.

Some Points Favoring Compression

By Prof. R. C. Heck

In the June 28 issue there appeared a review of Professor Dwelshauvers-Dery's experiments with compression, the general conclusions of which were not favorable to compression. The author points out certain limitations in these experiments and proceeds to prove that compression, when properly applied, is advantageous.

The exposition and review of the experiments of Professor Dwelshauvers-Dery on the effect of compression in the steam engine, published in the June 28 issue, opens up a field for discussion. The experiments were undoubtedly made with care and skill, and the observations may be accepted as correct; yet the conditions were so far from normal as to constitute an extreme, not a typical, case. The investigation was essentially a study of the thermal action upon the cylinder walls; but with a small engine at low speed this action is so exaggerated that deductions from the results are far from being applicable to large engines at usual speeds.

It cannot be denied that much of the traditional theory is very incomplete, failing to give due weight to quite obvious elements of the problem. However, the fact that ridiculous claims have been made for the economy of compression is no reason for going to the other extreme and saying that compression is always harmful, or that it has no effect upon economy. The subject can easily be put into rational form, even though the exact quantitative relations must remain on an empirical and perhaps indeterminate basis.

The simplest way to study the effect of compression is illustrated in Fig. 1. The steam caught in the clearance when the exhaust closes at *E* is compressed by the piston up to the point *F*, and then by the entering steam up to *G*. In expansion this steam is supposed to retrace the curve *GFE*, giving back the work that was expended in compressing it. The only loss of effect is the area *AFG*, which represents a waste of available work through a throttling action upon the entering steam. Some claim that the purpose of compression is to diminish this waste area; others state that the purpose is to fill the clearance space so that less steam will have to flow into it from the boiler before the working stroke of the piston begins.

Consider this latter phase of the subject first. If the expansion curve is carried up to the initial pressure at *J*, the distance *PJ* represents the volume which the total steam in the cylinder would have at that pressure. Of this, *PG* is clearance steam, and *GJ* is fresh working steam, representing the steam received from the boiler. The quotient obtained by dividing the working-area *ABCDEF* by the steam quantity *GJ* is an expression (although incomplete and not in the simplest units) for the thermal efficiency of the engine.

Now consider the other view of the function of compression. If there is no compression, in the ordinary sense, the residual steam in the cylinder at *K* will still be compressed, entirely by the entering steam, up to *L*. The area *EFK* will be added to the work diagram and the amount *LG* to the steam received. Because of the much larger waste area *AKL*, the efficiency according to the ratio of *ABCDK* to *LJ* will be less than it was with compression beginning at *E*. The decrease from *AKL* to *AFG* represents the entire apparent saving by compression; for several reasons the actual saving is considerably less.

A complete analysis of the manner in which the work area *AKL* is wasted would be rather lengthy. Briefly stated, as entering steam flows through a narrow valve, opening into a space of lower pressure, it acquires a high velocity and a part of its stored energy is changed into kinetic energy of the flowing stream. There is no way of applying this kinetic energy usefully, and as the steam comes to rest this energy changes back to heat. When the operation of filling the clearance space is finished, there is in the cylinder a volume *LA* of new steam. If admitted at full pressure behind a moving piston it could have done the work *LAMV*, and then accomplished further work during expansion. In acting upon the clearance steam this admission places only the portion *KLVM* of the available work *ALVM* where it can do work upon the piston. The steam of volume *LA* stands ready for expansion, with the work *ALK*

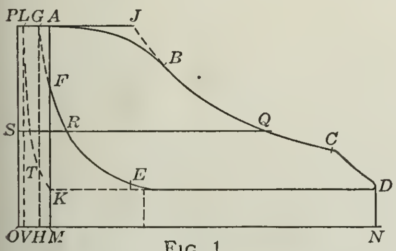


Fig. 1

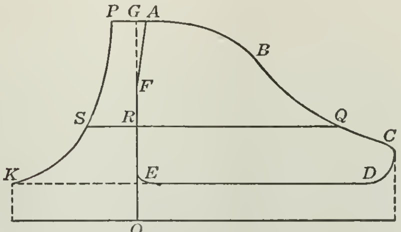


Fig. 2

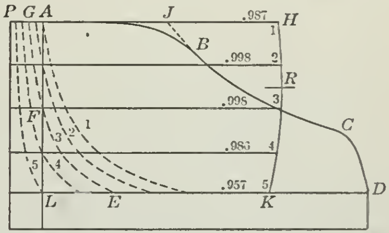


Fig. 3

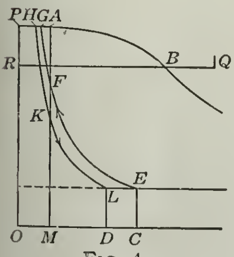


Fig. 4

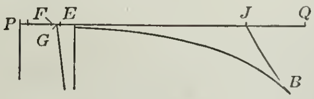


Fig. 5

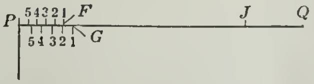


Fig. 6

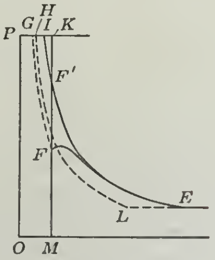


Fig. 7

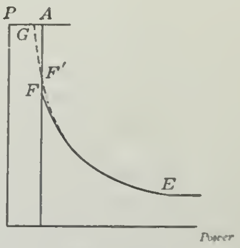


Fig. 8

returned as stored heat; but only a small portion of this heat can be reconverted into useful work. In this simplified description, the effect of the cylinder walls upon the quality of the steam has not been taken into account; this will be discussed further on.

Reverting to the statement that in Fig. 1 the apparent saving by compression is represented by the reduction of the wasted work from *AKL* to *AFG*, the causes which tend to neutralize this gain will now be considered; of these the first is the diminution of the effective expansion. In Fig. 2, through the expedient of rectifying the compression curve *EFG*, Fig. 1 is changed in such a manner as to separate distinctly the expansion operations of the clearance steam from that of the working steam. It may be well to imagine a movable partition or diaphragm separating these two bodies of steam. During the compression *EF*, Fig. 1, this diaphragm is against the piston face; at *F* it begins to act as a dividing wall. In Fig. 2, the volumes are measured from this movable dividing surface; for instance, *RQ* represents working steam and *RS* clearance steam; these are laid off with *R* on the fixed reference line, instead of *S*. Thus the clearance effect is eliminated, and the diagram *ABCDEF* shows the steam action as if it were carried out in a cylinder without clearance.

The purpose of this diagram is to emphasize the fact that, because of the large space which the clearance steam occupies as it expands to low pressure, the effective volume of the cylinder for the expansion of the working steam is reduced. The net result of combining this loss of expansion with the gain due to compression is shown in Fig. 3. Keeping a constant condition of admission, expansion and release, as shown by the curve *ABCD*, the compression is varied from zero to full pressure. For the present, the compression curves are assumed to be alike, all of the form $pv=C$. With each of the five compressions, the effective work area, as *ABCDEF*, is measured and is divided by the steam measure *J G* (*G* being considered movable and representing the end of compression in all cases). Values of the resulting ratios are laid off horizontally from *OP*, on lines through the points of intersection of *AM* with the compression curves; and the curve *HK* shows how this efficiency measure varies with the amount of compression. The maximum point at *R* is found by the purely graphical method of drawing a vertical tangent to the curve *HK*.

From *K* upward there is at first a gain, due to a decrease in the work wasted by the throttling effect of the steam which fills the clearance space. But this is gradually balanced by the loss due to decreased expansion of the working steam till above *R* there is a falling off in efficiency. Evidently, with earlier cutoff and lower release pressure at *C*, this effect of incomplete expansion becomes relatively less, and the point *R* will be higher up the curve. At the same time, with *PJ* becoming shorter, a change in the position of *G* will have a greater relative effect, *HK* will have a greater curvature and the compression will be of more importance as affecting economy. Fig. 3 confirms the generally accepted fact that the action of the ordinary single-valve, variable cutoff gears, which give more compression with earlier cutoff, is just what it ought to be for the large clearance engines on which such gears are used.

The method applied in Fig. 3 is not limited to ideal conditions but is applicable to general use. With actual compression curves, however, the initial point *E* is somewhat more definite than the point *F*. The efficiency curve *HK* can be based either directly on positions of *E*, or on the positions of *F* derived from *E*.

Now consider, for simplified conditions, the effect of the initial condensation by the cylinder walls upon the performance of the clearance steam. In Fig. 4 the total steam in the cylinder at cutoff, volume *RB*, would fill the space *RQ* if none were condensed. It is proper to take the clearance steam as dry at *E;* if compressed along the curve $pv=C$, it will be slightly wet at *G*. Assuming that during expansion the steam is homogeneous, the quantity of clearance steam will follow the expansion curve *H K L*, similar to the main expansion curve. At the maximum pressure, the ratio of *PH* to *PG* will be a little less than that of *RB* to *RQ*. Therefore, the clearance steam actually goes through the cycle *EGHL*, with a negative work output, or with a net absorption of the work represented by this closed figure. The line *L E* shows the reëvaporation, which takes place during the exhaust period.

The cycle diagram, as Fig. 4, at once disproves the assumption that the clearance steam gives back during expansion all the work that it received in compression. For present purposes, however, it makes very little difference whether the loss of work due to the cylinder action be divided between the working steam and the clearance steam, or considered as a whole. The important step is the change that must be made in the steam measure by initial volume, the length *GJ*, Fig. 1.

On the upper part of an indicator diagram reproduced in Fig. 5, *BJ* is the expansion curve carried up to admission pressure, *PQ* is the volume the total steam would occupy if dry, *PE* is the dry volume of the clearance steam, and *PF* is the volume of the clearance steam of the same quality as that of the total steam at *J*. The true measure of the steam is now either *EQ* in dry steam, or *FJ* in terms of steam of definite quality. As only relative results are sought in applying the method used in Fig. 3, the latter length will be the more convenient to use.

It has become to be generally accepted that the actual amount of steam condensed by the cylinder walls in the early part of each working stroke is, for a given engine, a nearly constant quantity; that is, it is little affected by a change in the load or by cutoff. If this be true, *FJ*, Fig. 5, for use in the method of Fig. 3, will vary as shown in Fig. 6. The quality of the steam at the beginning of expansion is fixed by the ratio of *P J* to *P Q*. As the clearance steam increases in amount, so does its absolute shrinkage from *G* to *F*. The result is that *FJ* decreases less rapidly than *GJ* with rising compression, and the upper part of an efficiency curve such as *HK*, Fig. 3, will be inclined a little more toward the left, and the maximum *R* will be somewhat lower.

In the Dwelshauvers-Dery experiments, the engine gave a compression curve similar to *EF*, Fig. 7, and the whole cycle for the clearance steam must have been of the form *EFGHL*. The remarkable thing about these results was the decided increase in cylinder condensation with greater compression. In terms of Fig. 3, all the tests made by Professor Dwelshauvers-Dery were above the maximum point *R*, and the curve *RH* had an extreme curvature toward the left.

As already remarked, this is a very extreme case. The usual result in engines where the conditions tend toward cylinder condensation is a slight droop of the compression curve, such as is shown in Fig. 8. As the steam begins to be condensed as soon as it reaches a temperature above that of the metal, the curve falls below the theoretical curve, rising to *F* instead of to *F'*. The action is of the same character as that in Fig. 7, but does not go nearly so far.

There is still some room for skepticism as to the entire absence of leakage in the experimental engine at Liège. Other experiments, notably those of Callender and Nicholson, quoted in Peabody's "Thermodynamics," have shown that a slide valve, although apparently tight when at rest, may leak considerably when in motion. Assuming, however, that no leakage occurred, the subject may be dealt with as follows : Through compression by the piston, the temperature of the clearance steam is soon raised above that of the inner surface of the cylinder walls. In a small engine, the amount of surface exposed to a given weight of steam is very large; and in a slow running engine there is plenty of time for heat transfer. The result is a compression curve of the form shown in Fig. 7, and the entering steam comes in contact with walls which are coated with moisture. The chief argument in favor of a steam jacket is not that it makes the cylinder walls many degrees hotter as that it insures their being dry. If, under certain conditions, early compression tends to produce just the opposite effect, we can readily see that it may be decidedly harmful.

Now just as the steam jacket loses its power for good with an increase in the size and speed of the engine, or with a lessening of the influences tending toward initial condensation, so does also this harmful phase of compression rapidly grow less. It is for this reason that the general applicability of the Dwelshauvers-Dery deductions is denied.

The method applied in Fig. 3 can be used equally well in the case of constant power instead of constant cutoff. Then as compression is increased, cutoff will occur later. Again, the upper part of curve *HK* will be bent toward the left, and the maximum efficiency will occur at a lower value for compression.

The proposition which was formerly advanced so frequently, that compression by warming the cylinder walls diminishes initial condensation is now generally discredited. The real purpose of compression is to diminish the waste of available work represented by the area *AKL* in Fig. 1. Referring to the compression curve in Fig. 3 it will be apparent that to raise the point *F* half way from *L* to *A* does far more good than to raise it all the way. Furthermore, every departure from the simple conditions illustrated tends to make the efficiency less with high compression than it appears in that figure. Hence the practical conclusion is "that moderate compression does benefit economy; there is a considerable range, roughly centered on the halfway point (midway between *L* and *A* in Fig. 3) over which change in compression is neutral in its effect; and if compression is carried too high it produces thermodynamic waste.

From the purely mechanical standpoint, the statement that compression helps to bring the reciprocating parts quietly to rest and prevents shock when the motion is reversed is inadequate and not entirely correct. The reversal of the stroke at the dead center involves nothing in the nature of a shock. If the engine were driven by a motor, with the cylinder heads removed, the moving parts would reverse at the end of the stroke without shock; for although the maximum inertia force is present, it varies at that point less rapidly than at any other part of the stroke. The reversal of the pressure on the pin joints would, in this case, occur near the middle of the stroke. Compression does contribute toward quiet running in that it provides for a more gradual reversal of steam pressure which takes place somewhere near the dead center. The rise of pressure shown by a compression curve is less rapid than that caused by the flow of steam past a quickly opened valve into the clearance space, especially if this space is small.

A strong point in favor of compression is that it diminishes the pressure on the pin and shaft bearings while the crank is near dead center. Over this range of motion, the turning moment is so small that the useful work done in turning the shaft is small, regardless of the driving force acting along the connecting rod; a large force, however, produces its full friction effect at the crank pin and shaft journal, because these joints are turning at full speed. By balancing the forward-acting inertia forces of the reciprocating parts near the end of the stroke, compression helps the mechanical efficiency of the engine. It absorbs work which the machine could use only at a disadvantage, and stores it for later delivery when the conditions are more favorable.

Solution of a Steam Problem

By C. E. Burgoon

In the July 19 issue a problem in thermodynamics was cited and solutions requested. The present article furnishes an exhaustive solution of the problem from a purely theoretical point of view.

Referring to "A Steam Problem," which appeared in the July 19 issue of POWER, the following solution may be of interest to many engineers. The problem as stated was as follows: Given initial pressure 100 pounds absolute; exhaust pressure 15 pounds absolute; quality of entering steam, dry saturated; quality of exhaust steam, 0.9; volume of clearance, 0.4 cubic foot. Find the weight of the steam at *b*, Fig. 1, and its condition, assuming that there is no heat transfer to the cylinder walls.

The volume of the water, which is mixed with the steam, is small compared to the volume of the steam, and has been neglected in the solution. It would not change the results, materially, but would make the solution much more complicated.

The weight of 1 cubic foot of saturated steam at 15 pounds absolute pressure is 0.03805 pound; and the weight of 0.4 cubic foot (clearance volume) equals

$$0.4 \times 0.03805 = 0.01522 \text{ pound.}$$

The exhaust steam has a quality of 90 per cent., hence the weight of steam and water in the clearance volume equals

$$0.01522 \div 0.90 = 0.01691 \text{ pound.}$$

The total heat above 32 degrees Fahrenheit contained in one pound of expanded moist steam is given by the formula

$$H = q + x p$$

where,

H = B.t.u. per pound of the steam;
q = Heat of the liquid;
p = Internal latent heat;
x = Quality.

Substituting the constants for steam at 15 pounds absolute pressure and 90 per cent. quality in the foregoing formula:

$$H = 181.3 + 0.90 \times 896.2 = 987.88 \text{ B.t.u.}$$

and the total heat contained in 0.01691 pound of steam equals

$$987.88 \times 0.01691 = 16.70 \text{ B.t.u.}$$

The 0.4 cubic foot of steam, in the clearance volume, must be compressed to 100 pounds absolute pressure, the work (heat) of compression being taken from the admission steam. The volume, quality and heat contents must be determined at 100 pounds absolute pressure in order to determine the amount of admission steam, its quality after doing the work of compression and the quality of the mixture.

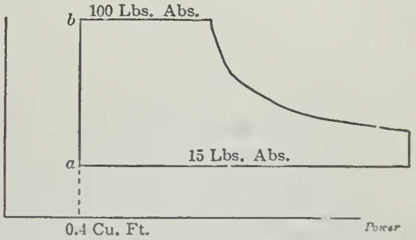


FIG. 1. DIAGRAM PRESENTED IN PROBLEM

Since there is to be no heat interchange between the steam and the cylinder, the compression will be adiabatic; and the entropy at the beginning and end of compression will be the same. The entropy at the beginning of compression (15 pounds absolute and 90 per cent. quality) equals the entropy of the liquid, 0.3140, plus 90 per cent. of the entropy of evaporation, 1.4409; that is,

$$E = 0.3140 + 0.90 \times 1.4409 = 1.6107$$

and the entropy of saturated steam at 100 pounds absolute pressure equals the entropy of the liquid, 0.4748, plus the entropy of vaporization, 1.1273; that is,

$$E_2 = 0.4748 + 1.1273 = 1.6021.$$

It should be noted that the total entropy of saturated steam at 100 pounds absolute pressure is less than the total entropy of the steam at 15 pounds absolute pressure and 90 per cent. quality. This indicates that the steam will be superheated at the end of compression; and the entropy of superheat equals

$$E_S = E_1 - E_2 = 1.6108 - 1.6021 = 0.0087$$

The absolute temperature of the superheated steam, at the end of compression, may be found from the formula,

$$Log._e T_1 = (E \div S) + Log._e T_2$$

where,

T_1 = Absolute temperature of the superheated steam;

T_2 = Absolute temperature of the saturated steam;

S = Specific heat of superheated steam;

E_S = Entropy of superheat.

Substituting the value of T_2 for 100 pounds absolute pressure, E_S as determined, and $S = 0.55$ in the above formula,

$$Log._e T_1 = (0.0087 \div 0.55) + Log._e 787.4$$

and,

$$T_1 = 800$$

The degrees of superheat $= T_1 - T_S = 800 - 787.4 = 12.6$.

Although steam in a static condition does not contain the external latent heat, the latter must be considered when calculating the work done in compressing the steam. The work done in compression is equivalent to the difference in the heat contents at the beginning and at the end of compression. The total amounts of heat in the steam at the beginning and at the end of compression are given by the following formulas:

$$H_1 = (q_1 + x\,r_1) \times W$$

and

$$H_2 = (q_2 + r_2 + S \times D) \times W$$

where

H_1 = Total B.t.u. in one pound of the steam at beginning of compression;

H_2 = Total B.t.u. in one pound of the steam at end of compression;

q_1 = Heat of the liquid at pressure corresponding to beginning of compression;

q_2 = Heat of the liquid at pressure corresponding to end of compression;

r_1 = Heat of vaporization at pressure corresponding to beginning of compression;

r_2 = Heat of vaporization at pressure corresponding to end of compression;

x = Quality at beginning of compression;

S = Specific heat of superheated steam at that temperature and pressure;

D = Degrees of superheat;

W = Weight of steam in pounds.

Substituting the constants for the steam at the beginning of compression in the first formula,

$$H_1 = (181.3 + 0.90 \times 969.1) \times 0.01691 = 17.81 \text{ B.t.u.}$$

Substituting the constants for the steam at the end of compression in the second formula,

$$H_2 = (298.5 + 887.6 + 0.55 \times 12.6) \times 0.01691 = 20.17 \text{ B.t.u.}$$

The work of compression expressed in heat units equals,

$$20.17 - 17.81 = 2.36 \text{ B.t.u.}$$

The volume of one pound of superheated steam may be found by the formula,

$$P\,V = 0.5962\,T - P\,(1 + 0.0014\,P)\,C$$

where

V = Volume in cubic feet per pound;

P = Absolute pressure in pounds per square inch;

T = Absolute temperature, Fahrenheit;

C = Constant (see Peabody's steam tables, page 76).

Substituting the constants corresponding to the condition of the steam at the end of compression in the foregoing formula may be expressed

$$100\,V = 0.5962 \times 800 - 100\,(1 + 0.0014 \times 100)\,0.211.$$

$V = 4.529$ cubic feet per pound and the volume of 0.01691 pound of steam equals

$$0.01691 \times 4.529 = 0.0766 \text{ cubic feet.}$$

The clearance volume to be filled with admission steam is

$$0.4 - 0.0766 = 0.3235 \text{ cubic feet.}$$

The work required to compress the clearance steam will be taken from the external latent heat of the admission steam. But all of these heat units will not be used for this purpose; and as they cannot be retained by the steam as external latent heat they go to superheat the admission steam; this will be explained later.

The admission steam has raised the clearance steam to 100 pounds absolute pressure; but to determine the quality of the steam, it is necessary to assume a quality, and then check the assumed value. After trying three or four different values for the degrees of superheat, 80 degrees checked fairly close, and the solution will be continued with this value.

The volume of steam at 100 pounds absolute pressure and 80 degrees of superheat as given by the foregoing pressure-volume formulas is as follows:

$$100\,V = 0.5962 \times 867.4 - 100\,(1 + 0.0014 \times 100)\,0.147.$$

$$V = 5 \text{ cubic feet per pound.}$$

The volume of the admission steam in the clearance was found to be 0.3234 cubic foot; and its weight equals

$$0.3234 \div 5 = 0.0647 \text{ pound.}$$

The external heat of one pound of dry saturated steam at 100 pounds absolute pressure is 81.9 B.t.u., and the external latent heat of 0.0647 pound equals

$$81.9 \times 0.0647 = 5.30 \text{ B.t.u.}$$

The heat available for further heating the admission steam equals the total external latent heat minus the heat converted into work to compress the clearance steam, that is,

$$5.30 - 2.36 = 2.94 \text{ B.t.u.}$$

This number of heat units would raise the temperature of one pound of steam at 100 pounds absolute,

$$2.94 \div 0.55 = 5.35$$

degrees (0.55 being the specific heat of steam at this pressure) and 0.0647 pound of steam.

$$5.35 \div 0.0647 = 81$$

degrees. The degrees of superheat were assumed to be 80, and the calculated value, based on this assumption, is 81 degrees, which checks very closely.

The total heat in 0.01691 pound of clearance steam at 100 pounds absolute pressure and 12.6 degrees of superheat equals

$$H = (298.5 + 805.7 + 0.55 \times 12.6)\;0.01691 = 18.80 \text{ B.t.u.;}$$

and the total heat in 0.0647 pound of admission steam at 100 pounds absolute pressure and 80 degrees superheat equals

$$H = (298.5 + 805.7 + 0.55 \times 80)\;0.0647 = 74.29 \text{ B.t.u.}$$

Hence, the total heat in the steam, in the clearance volume, equals the heat in the clearance steam, after it has been compressed to 100 pounds absolute pressure, plus the heat in the admission steam, after it has compressed the clearance steam.

$$74.29 + 18.80 = 93.09 \text{ B.t.u.}$$

and the total weight of the steam in the clearance volume equals the weight of the clearance steam, 0.01691, plus the weight of the admission steam, 0.0647;

$$0.01691 + 0.0647 = 0.08161 \text{ pound.}$$

The heat in 0.08161 pound of expanded saturated steam is,

$$H = (298.5 + 805.7)\;0.08611 = 90.11$$

B.t.u., the heat due to superheat equals

$$93.09 - 90.11 = 2.98 \text{ B.t.u.;}$$

and the degree of superheat equal

$$2.98 \div (0.55 \times 0.08161) = 66 \text{ degrees.}$$

By making use of the pressure-volume formula, the volume of one pound of steam at 100 pounds absolute pressure and 66 degrees of superheat is determined as follows:

$$100\,V = 0.5962 \times 853.4 - 100\,(1 \div 0.0014)\,0.158.$$

$$V = 4.91 \text{ cubic feet;}$$

the volume of 0.01691 pound equals

$$0.01691 \times 4.91 = 0.083 \text{ cubic foot;}$$

the volume of 0.0647 pound equals

$$0.0647 \times 4.91 = 0.317 \text{ cubic foot;}$$

and the volume of 0.08161 pound equals

$$0.08161 \times 4.91 = 0.40 \text{ cubic foot;}$$

Under the conditions given at point *a* of the diagram, the clearance volume would contain 0.01691 pound of steam at 15 pounds absolute pressure and 90 per cent. quality and containing 17.81 B.t.u. and under the conditions given at point *b*, the clearance volume would contain 0.08161 pound of steam at 100 pounds absolute pressure and 66 degrees of superheat and contains 93.09 B.t.u.

The constants used in this solution were taken from Peabody's new steam tables, excepting the logarithms, which were taken from "Kent's."

The statement was made earlier in the discussion that "part of the external latent heat of the admission steam would be converted into heat of superheat, and that this would be explained later."

The boiler forces 0.0647 pound of steam out against 100 pounds absolute pressure; and the work done by the boiler equals the external latent heat of the steam or

81.9 × 0.0647 = 5.30 *B.t.u.*

Referring to Fig. 2, area *c d h g c* represents the external latent heat (5.30 B.t.u.) of the admission steam; area *c k f h g c* represents the external latent heat (2.36 B.t.u.) of the admission steam required to compress the clearance steam; and area *c d f c* represents the external latent heat (2.94 B.t.u.) of the admission steam which superheats the admission steam.

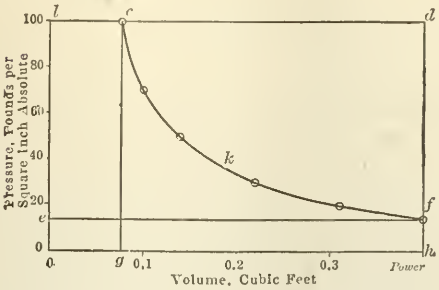


FIG. 2. EXTERNAL LATENT HEAT OF ADMISSION STEAM

Assume a diaphragm to be placed between the clearance steam and the admission steam to prevent steam from mixing or an interchange of heat. Also assume that the admission steam is let into the clearance volume by steps, and that a diaphragm separates each step. The pressure of the steam which fills the first step—say from 15 to 30 pounds—is reduced without doing any work, hence the steam is superheated; when the steam fills the second step—say from 30 to 60 pounds—the steam in the first step is compressed to 60 pounds pressure and is superheated to a higher degree of superheat. The same is true of the succeeding steps until the clearance volume is full of steam at 100 pounds absolute pressure. The diaphragms are now removed, the steam mixes and comes to a constant and common temperature.

The writer invites other engineers to criticize or comment on the foregoing solution; and if it is not right, he sincerely hopes that someone will give the correct solution. The problem contains several points which are not generally understood by technical and practical engineers.

Minnesota N. A. S. E. State Convention

The attendance and amount of interest shown in this meeting, which was the first in the history of the State organization, was a surprise to many who had the pleasure of being there. In fact, the convention was pronounced as equal to any of the State meetings held in the Central West, and the educational program, consisting of lectures by prominent men skilled in their different lines was more than usually elaborate for a State convention. The character of the mechanical exhibit was also high and all expressed satisfaction as to the results of the meeting.

DELEGATES TO THE MINNESOTA STATE CONVENTION

The convention was called to order in the mayor's reception room, city hall, by H. E. Chase, chairman of the local committee, after which there followed an address of welcome by Alderman Dwyer, representing the mayor. Herbert P. Keller, mayor of St. Paul, also made a pleasing address to the visitors. Governor A. O. Eberhart was represented by his secretary, Ralph W. Wheelock, who commended the activity of the engineers in forming a State organization. Responses to these addresses of welcome were made by F. W. Raven, national secretary, and J. H. Harris, of Chicago. Mr. Raven outlined the history of the organization and brought a message of greeting from all of the national officers.

The educational work included an illustrated lecture by Charles L. Pillsbury, consulting engineer for the State board of control, on the subject of "The Mechanical Equipment of Buildings and Industrial Plants." During the course of the lecture there were shown views of a number of prominent steam plants with their different characteristics and interesting comment was made on the peculiar conditions which in each case determined the method of solving the problems for that particular installation.

J. C. Larson, of Dubuque, Iowa, gave an informal talk on "Steam Turbines," in which he illustrated by sketches and diagrams the difference between the action and reaction type of machines, after which he took up the peculiarities of the different makes of modern equipment, calling attention to the features wherein they differed from each other.

J. L. Barnard talked entertainingly on "Incandescent Lamps," bringing out a number of points which are not generally appreciated by the users.

Another lecture enjoyed by the delegates was by Alex Cunningham, of the Hartford Steam Boiler Inspection and Insurance Company, who spoke on "Inspection of Boilers from the Insurance Standpoint." After giving an outline of the history of boiler insurance he took up the question of boiler inspections and recommended that the hammer test be used exclusively for examination of boilers, as it was his belief that this method, using the eye and the ear, for the detection of defects, is safer and more rational than the hydrostatic test commonly applied. Using the hammer test, an inspector must penetrate to every part of the boiler and will be more liable to detect undesirable conditions existing. Furthermore, in using the hydrostatic test, he believed the boiler is frequently overstrained, sometimes even to the elastic limit of the plates, thereby leaving them in worse condition than they were before the test was applied.

"Analysis of Water Pertaining to Steam" was the subject of a lecture by E. A. Converse, of Chicago. The subject was especially interesting to all present on account of its importance in the daily operation of steam plants. The different reactions in water analysis were shown, and by means of a complete equipment of chemical apparatus the simplicity of the operations was demonstrated. Many analyses of water throughout the Middle West were given and the variation in character of waters in the same district was clearly shown, thus demonstrating the necessity for analyzing the feed water for getting the best results.

One of the most important things to come before the convention was the matter of improving the character of the license issued to steam engineers in the State of Minnesota. In this connection State Senator M. L. Fosseen, in an address on "The License Law, as It Now Is and the License Law as It Should Be." brought out the fact that even the ordinary citizen is rapidly becoming convinced that more rigid examinations and a higher standard of ability should be required from those who are to have charge of steam boilers. Senator Fosseen believed that this question should be agitated without delay in the interest of the public and expressed the hope that some immediate progress in this direction would be made.

Following Mr. Fosseen's address, E. P. Gould, secretary of the Central States Exhibitors' Association, addressed the delegates on the necessity for coöperation between the supplymen and the State association in making a success of their annual meetings.

Entertainment was liberally provided for the visitors, a theater party at the Orpheum being one of the features. An excursion to Lake Minnetonka by trolley and a boat ride on the lake, through the courtesy of the Minneapolis Commercial Club, were thoroughly enjoyed by all.

Officers for the ensuing year were elected as follows: H. M. Germain, No. 7, of St. Paul, president; F. J. Strife, No. 4, of Winona, vice-president; James McGeary, No. 7, of St. Paul, secretary; Albert Johnson, No. 7, of St. Paul, treasurer; P. B. Wells, No. 2, of Minneapolis, conductor; W. E. Strutt, No. 2, of Minneapolis, doorkeeper; Walter Dickinson, No. 2, of Minneapolis, trustee for three years; Joseph Hanson, No. 7, of St. Paul, trustee for two years, and Peter Ulstrum, No. 2, of Minneapolis, trustee for one year; H. E. Chase was indorsed as State deputy.

In closing, St. Paul was chosen as the place of the next meeting.

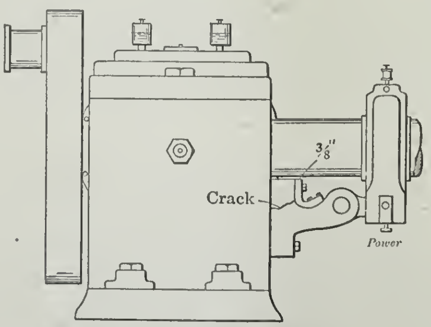
The mechanical exhibit was held in the Grand Army room, city hall, firms having representatives in attendance being as follows:

Williams Brothers Boiler and Manufacturing Company, Minneapolis, Minn.; Bryan-Marsh Company, Chicago; Cadwell & Brown Company, Minneapolis; George B. Carpenter Company, Chicago; Crane Company, Chicago; G. M. Davis Regulator Company, Chicago; Dearborn Drug and Chemical Works, Chicago; Detroit Lubricator Company, Detroit; Electric Machinery Company, Minneapolis; Erie City Iron Works, Erie, Penn.; Fidelity and Casualty Company, New York; Garlock Packing Company, Palmyra, N. Y.; D. M. Gilmore Company, Minneapolis; Fred L. Gray Company, Minneapolis; Greene, Tweed & Co., New York; Hanks-Price Company, Minneapolis; G. M. Harrison, Minneapolis; Hartford Steam Boiler Inspection and Insurance Company, Hartford, Conn.; Hawkeye Boiler Compound Company, Chicago; Hills-McCanna Pump Company, Chicago; Jenkins Brothers, New York; Johnson Service Company, Milwaukee; Keasbey & Mattison Company, Ambler, Penn.; George M. Kenyon, Minneapolis; Keystone Lubricating Company, Philadelphia; King Oil Company, Minneapolis; R. A. and R. M. Laird, Minneapolis; London Guarantee and Accident Company, London, Eng.; Lunkenheimer Company, Cincinnati; Lyons Boiler Works, De Pere, Wis.; William McMillan & Co., Minneapolis; Minneapolis Steel and Machinery Company, Minneapolis; Northwestern Equipment Company, St. Paul, Minn.; W. S. Nott Company, Minneapolis; Osborne High-Pressure Joint and Valve Company, Chicago; Peerless Rubber Company, New York; Penn Oil and Supply Company, Oil City, Penn.; Plant Rubber Company; William Powell Company, Cincinnati; POWER AND THE ENGINEER, New York; Power Equipment Company, New York; Quaker City Rubber Company, Philadelphia; Revere Rubber Company, Boston, Mass.; Sim D. Rollins Company, Minneapolis; *Southern Engineer*, Atlanta, Ga.; Under-Feed Stoker Company of America, Chicago; Viscosity Oil Company, Chicago; R. B. Whitacre Company.

Temporary Engine Repair Job

Engineers of the South are frequently called upon to do repair work which requires more ingenuity than is exercised by engineers in the North where machine shops are more plentiful. In the case of a releasing gear engine, the piston was broken from some unknown cause, and a piece sticking in the exhaust port caught the valve as it was closing, with the result that the outer bearing of the exhaust valve shaft was broken at a point where it was bolted to the pillow block of the engine, as shown in the illustration.

As the engine was needed, the engineer in charge decided that a reinforcing plate of sufficient strength could be made to fit in the curve of the casting, enabling it to give further service until permanent repairs could be made. A piece of ⅜-inch iron of suitable width was obtained and forged to conform to the shape of the bearing casting, after which holes were drilled in the iron and bearing casting, the latter being tapped for ⅜-inch bolts. When these were secured and the a bolt screwed through the hole in the reinforcing plate corresponding with the hole in the broken end of the casting, the job was found to be so satisfactory that the same bearing is still in use.



DETAIL VIEW SHOWING REPAIR

Open each door to boiler at least once a day and look carefully for leaks at every possible point, noting tube ends, seams and blowoff pipe in furnace. Examine carefully the fire sheets for bulges, bags, or blisters.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 we printed and circulated 1,875,000 copies of POWER.

Our circulation for August, 1910, was 170,000.

September 6........... 35,000
September 13.......... 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

American Engineering

It has been said, in a mixed spirit of levity and criticism, that Europeans build machines to work, while Americans build machines to sell, and with much humility and mortification we are compelled to admit that some of the examples cited to back up the argument are indefensible. The most recent and striking (no pun intended) of these is the automobile, but as this is not a motor-car paper we shall not go into details beyond the general statement that an enormously greater proportion of American motor cars than of the least meritorious national group of European cars are of bad design and worse construction. But leaving out motor cars, it is uncomfortably true that in general the European design and quality of workmanship embodied in all machinery excepting machine tools and printing presses have been perceptibly—yes, obviously—superior to ours, and Europeans usually account for all of it on the theory that Americans, both builders and users, are in such a hurry that they will not allow sufficient time to be spent on a design to weed out bad features and improve lame ones. That this is true to some extent is undeniable. It is also true that in many cases inferior products have been turned out because the makers were convinced that the "trade demanded" that kind and would not pay the price of first-class manufactures. In still other instances, we are sorry to admit, the builders mean well but do not know enough about engineering to realize that their product is crude; consequently, they have deluded themselves into thinking it excellent.

However, the two kinds of error last cited will eventually be cured, either from the inside or by those who buy, but there exists a more serious condition in American industries than either of these, and one which is a heavy handicap; that is the lack, on the part of the business end, of a proper appreciation of the technical department of a manufacturing establishment. In practically every case where success has been achieved on merit, the men who controlled the business policy were men of either technical training or long practical engineering experience. In almost every case where inferior product is turned out—excepting the two classes first mentioned—the business heads are men whose sole idea is to build just as cheaply as possible and to grind out the machines or apparatus just as rapidly as possible. This class of "manufacturer" is much too prominent in almost all branches of machinery building.

In other words, it is not that American engineering is intrinsically poor but that it hasn't had a full chance because so many of our large establishments are controlled by dollar-grabbers who are either so small-souled that they care nothing for such sentimental considerations as industrial reputation, quality of product, etc., or so narrow-minded that they don't realize that sound engineering is a permanent foundation for which there is no adequate substitute.

Thank heaven there are enough exceptions to prevent us from losing hope, and the number is steadily increasing.

The Boiler Inspector

As in all pursuits, there are many qualifications required for the making of a good boiler inspector; for example, honesty, temperance and energy; but there are two qualifications of paramount importance, these being confidence in himself and the possession of plenty of backbone. The man who is not certain of himself cannot possibly make a success of boiler inspecting, and unless he has backbone to stick to what he believes is right, regardless of seductive arguments or threats to the contrary, his failure is certain.

The boiler inspector is not only responsible to his company (which generally assumes a considerable monetary risk on his statement of facts), but he is morally responsible for the safety of the lives of those who are required to be in the vicinity of the boilers he inspects.

He should turn a deaf ear to arguments intended to gain his consent for a continuance of conditions that his better judgment tells him are dangerous. In inspecting a boiler plant there are always points noted where improvements might be made; but the inspector must be able to draw a sharp line dividing changes that are necessary for safety and those which may or may not be made at the option of the owner; and such distinction should be plainly evident in his report. Often an owner will make the threat to cancel his policy if certain repairs are demanded, but if the inspector is sure of his ground he is serving the best interests of his company to stand firm, and the chances are ten to one that he will win out without the loss of the risk. Nothing serves to inspire the boiler owner with confidence in the inspector's ability so readily as having his bluff promptly "called." In the future there will be clear sailing.

The boiler inspector is, of course, only human and therefore will make mistakes. When he is convinced that he is in error a frank acknowledgement will generally serve to place him more firmly in the confidence of the owner. If boiler-insurance companies could only obtain instruments capable of determining the confidence (coupled with ability) and backbone possessed by applicants for positions in their inspecting corps, it would be a simple matter to select the proper candidates.

Vacuum

The makers of steam turbines claim and realize great advantage from working with a good vacuum; it may be interesting to make a comparison with the conditions for a reciprocating engine.

The reciprocating engine with a large expansion, a pumping engine for example, should have a good vacuum, 28 inches being obtained in good practice. Such an engine may have a reduced mean effective pressure of about twenty-five pounds; a difference of one pound means four per cent. in the economy of the engine, and one pound is the difference between twenty-six and twenty-eight inches.

But the reciprocating engine does not expand down to the back-pressure; good practice for a pumping engine is a terminal pressure of five pounds, absolute. The reason for this is that it takes about two pounds more effective pressure to make the engine go without load; this may be high for a good modern engine, but it is not enough for our present purpose. Now twenty-eight inches of vacuum is about one pound absolute, and if the expansion were carried down to that back pressure, the mean pressure during the expansion would be about three pounds (i.e., the mean of five and one), and subtracting the one pound back pressure would give two pounds mean effective pressure, which would just make the engine go.

Again, the volume of steam at one pound pressure is somewhat more than four times what it is at five pounds; consequently it would require an engine four times as large to expand down to back pressure and that most emphatically would not do.

It must be borne in mind that the reciprocating engine, as previously pointed out, gets a large advantage from a good vacuum without requiring the enormous size required to expand to the corresponding back pressure.

Coming to the case of a steam-turbine, the first idea to get clearly in mind is that the expansion must be carried in the turbine to the vacuum because the steam is discharged at the terminal pressure, whatever that may be, and there is nothing corresponding to the back pressure of the return stroke of a steam engine. But the turbine has two important conditions: in the first place, it is necessary only to add a row or rows of blades to obtain the expansion to a good vacuum, and there is comparatively little disadvantage in having long blades at that end of the turbine; in the second place, the addition of more rows of blades does not add to the mechanical friction of the turbine. The case is more favorable than just stated, for there are certain devices for avoiding undue lengths of blades. In the Parsons turbine they increase the blade angle and reduce the number of blades; this arrangement increases the effective space between the blades and so avoids too great blade length. But increasing the angle also increases the steam velocity, which is even more effective in avoiding too great length of blade. It may be noticed that in marine steam turbines the low-pressure barrel has a large number of rows at the end which have the same length and that these blades are set at a larger angle and especially for the last rows, are spaced wide apart. There is undoubtedly some disadvantage in this arrangement, but at any rate the blades are kept within due bounds.

Getting Ahead

Many an engineer has wondered why he does not get on with his work and why an acquaintance who is no better informed on engineering subjects, is no more faithful in the performance of his duities and does not keep his plant in any better condition, draws more pay and occupies a better position; also why, when an engineer is required for a new uptodate plant, the acquaintance gets the job. The general answer is that there is some vital difference between the engineer who does the wondering and his friend who gets the job. It is not reasonable to suppose that the other man succeeds in every case by bluffing, because the law of averages would turn the tables sometimes—bluff cannot win in every case. In nearly every case it is merit that enables that "other" engineer to get the job; it is the ability to put to practical use the knowledge he has and that he is constantly obtaining.

Of the thousands of power plants scattered throughout the country, it is improbable that there is a single one that cannot be made more efficient in operation—if the proper things are done; and the man to do the proper things is the engineer. But if he sits in a cushioned chair, sticks his feet up on the desk and puffs away at a cheap cigar, content with himself and the world and with the belief that his plant is better than Smith's, the proper things will not be done.

An engineer does not get an increase in salary for keeping the plant running, for seeing that the oil and waste do not run short, that the engine is wiped off, or that other routine work about the engine room is carried out each day. Not at all. There are hundreds of men of very limited experience who can do all these things, even to holding down the old arm chair.

An engineer cannot go to his employer and demand an increase in salary because he has had the engine started two minutes before the whistle has blown each day for the past ten years. Nor can an increase in pay be expected because the engine has been kept running regularly during working hours for that period. When an engineer can go to headquarters and make intelligent suggestions for improvements of the plant that will save money, and can "make good" after his ideas are carried out, then he is in a position to demand more pay.

What is the advantage of understanding all types of CO_2 recorders, for example, if that knowledge is not put to practical use? It will do no good to drop into Jones' place on the way home and advise him that from ten to thirteen per cent. of CO_2 should be obtained under ordinary working conditions, when the adviser does not know whether the CO_2 in his own plant exceeds five per cent. or not. Nor does it help matters to possess a knowledge of flue-gas analysis and keep it to yourself. The man in the office doubtless knows little, if anything, about this opportunity for saving. It is up to the engineer not only to know about it but to see that his employer knows about it.

The foregoing is only one of many possible opportunities for improvement. The engine cylinder may need reboring, but the president of the company is not usually in a position to know about it, and even when he is it does not tend to advance the engineer's interests to sit down and wait for the head of the establishment to suggest improvements. It is far better for the engineer to be in position to make such suggestions himself.

Summing all this up in a single sentence: It it not the everyday attention to routine work that makes an engineer so valuable that he "gets ahead" steadily, although that is one of the requisites of success; it is the ability to see and make use of opportunities for protecting the interests of his employer.

Probably the best way to make old junk into good machines is to break it up and send it to the foundry.

Don't be stingy with the waste. A clean engine will last longer and do better work.

Inquiries of General Interest

Allowance for Shrinkage Fits

How much allowance should be made for a shrinkage fit for a crank pin 4 inches in diameter?

J. D. C.

There is considerable diversity in practice and probably no fixed rule that will fit all cases can be given, as much depends on the quality of the material, length of fit, workmanship, etc. In one shop an allowance of 0.0008 inch is allowed for each inch of pin diameter, which for a 4-inch pin would be 0.0032 inch; in another shop 0.001 inch for each inch of pin diameter plus a constant of 0.003 inch is the regular allowance, which for a 4-inch pin would be 0.007 inch, or more than double the allowance first given. Both shops turn out first-class work and their crank pins never come loose.

Leakage Through Metal

Is there any leakage through the pores of metal pipes or tanks containing air or gas under a pressure of 200 to 250 pounds?

V. R. E.

Not unless the material of which the retaining vessel or pipe is made is defective. Compressed air and carbonic-acid gas tanks have been known to carry a pressure of over 2000 pounds per square inch for weeks without showing a measurable loss in pressure.

Horsepower of Falling Water

What horsepower would be developed by a column of water 24 inches in diameter in falling 25 feet?

H. D. W.

As the velocity of freely falling water increases with the distance through which it falls, the column could have a diameter of 24 inches at one point only of its length. Assuming that it has this diameter at the bottom, and calculating its velocity by the formula

$$V = 8.02\sqrt{h}$$

in which v = velocity and h = head, and substituting 25 for h, the result given below is obtained.

$$V = 8.02\sqrt{25} = 40.1$$

feet per second.

The cross-sectional area of a column of water 24 inches in diameter is 3.141 square feet, and at a velocity of 40.1 feet per second the flow will be

$$3.141 \times 40.1 = 125.95$$

cubic feet per second. At 72 degrees water weighs 62.3 pounds per cubic foot and the energy of 125.95 cubic feet of water falling 25 feet is

$$62.3 \times 25 \times 125.95 = 195{,}167.125$$

foot-pounds per second, which divided by 550, the foot-pounds per second in a horsepower, gives

$$195{,}167.125 \div 550 = 354.8$$

horsepower as the energy of a 24-inch column of water falling 25 feet.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Contents of Circular Taper Tanks

I have a round tapering water tank which is 12 feet across the bottom, 10 feet at the top and 12 feet deep. How shall I calculate its capacity in cubic feet or gallons?

W. M. L.

In form, the tank is the frustum of a cone, the cubical contents of which is found by adding together the areas of the top and bottom and a mean proportional between them, and multiplying the sum by one-third the depth.

Area of Top.......................:	78.54	square feet
Area of Bottom......................	113.10	square feet
Mean Proportional............	94.24	square feet
Sum......	285.88	square feet

$$12 \div 3 \times 285.88 = 1143.52$$

cubic feet, the contents of the tank.

There are 7.48 gallons in a cubic foot. Therefore the tank contains

$$1143.52 \times 7.48 = 8563.52$$

gallons.

Boiler Pressure Below the Water Line

In a horizontal return-tubular boiler the steam gage shows a pressure of 80 pounds. What is the pressure at any point below the water level?

C. S.

A steam gage shows the pressure of the steam in the boiler at the surface of the water. At any point below the surface the pressure is increased by the weight of the water above it. With a pressure of 80 pounds in the boiler, the pressure would be increased 39/100 of a pound for each foot in depth of the water.

Width of Butt Straps

Why is the outside strap of a butt joint made narrower than the inside one?

F. R. A.

The outside strap on a butt joint is made narrower than the inside one, because the width is not necessary for strength and the closer pitch of the second row of rivets makes tight calking possible.

Steam Temperature for Any Pressure

Is there any rule for finding the temperature of steam from its pressure?

W. H. E.

To find the temperature of steam for any pressure multiply the sixth root of the pressure in inches of mercury by 177.2 and subtract 100.

To get the pressure in inches of mercury multiply the steam pressure by 2.041.

Covering Sectional Heating Boilers

Please give me simple directions for applying asbestos covering to sectional cast-iron house-heating boilers.

A. S.

Sectional boilers for house heating usually have a bead molding running along the edge of each section for the purpose of holding the covering in place. The covering should be made into a thin plaster with water and applied in several coats. The first should not be more than ½ inch in thickness and should be put on as roughly as possible. Each coat should be thoroughly dried before the next is applied. The last or finishing coat should be about ⅛ of an inch thick and mixed as stiff as it can be worked and smoothed with a trowel.

Effect of Air Supply

What is the effect upon combustion if too little air is supplied through the fire? If too much air is supplied?

J. A. S.

If too little air is supplied the fire burns slowly and combustion produces carbon monoxide with only one-third of the heating power of carbon dioxide, which is formed when the supply of air is sufficient. If too much air is supplied it causes a waste of heat as it is necessary to heat the surplus quantity of air, and the oversupply of air has a tendency to cool the gases below the igniting temperature.

Water Pressure

Will the water in a tower 120 feet high and 15 feet in diameter exert the same pressure in the mains as a tank at the same hight with an 8-inch riser?

V. K. S.

The pressure per square inch in the mains will be the same in both cases if the water level is the same, as the pressure depends on the hight of the column and not on its shape.

New Power House Equipment

Turbine Mechanical Gearing for Slow Speed Steamships

According to a paper read at the spring meeting of the Institute of Naval Architects, by C. A. Parsons, the helical and double-helical gear, of fine pitch, suited to high speeds of rotation was first introduced by Doctor De Laval, of Stockholm, and has been extensively used in connection with his turbine for many years with entire success, and at a moderate cost of maintenance.

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

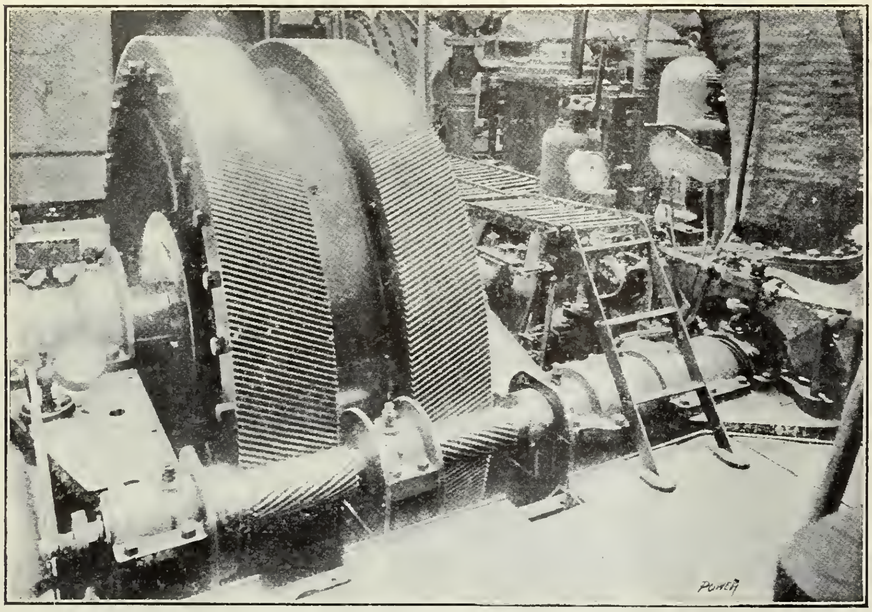


Fig. 1. Reducing Gear with Casing Removed

Mr. Parsons had several experimental sets constructed. One of these was a double-helical gear of the De Laval type, made in 1897, gearing from 9600 revolutions of the turbine to 4800 of the dynamo and transmitting 300 horsepower. The efficiency was estimated by the heat-loss method to be above 98 per cent. This gear was cut in an ordinary universal milling machine without any special precaution as to accuracy and in spite of the obvious irregularity of the teeth it ran well except that it made considerable noise. Gears that have been recently cut by special machinery run with very little noise. A recent experimental set of gearing from 2000 to 400 revolutions, transmitting 300 on a Heenan-Froude water-brake dynamometer, gave a total loss in the gear case, including friction of gear and bearings, of 1½ per cent.

A turbine reducing gear of the design made by the Parsons Marine Steam Turbine Company, Limited, 97 Cedar street, New York City, has recently been placed in the "Vespasian," a slow-speed cargo vessel. The gear is shown in Fig. 1. It consists of pinions on the turbine shaft meshing with large helical gears on the propeller shaft, the teeth in each gear being cut helical. The gears as arranged form a double-helical gear.

In order to eliminate strain on the shaft a flexible coupling is inserted on the shaft between the turbine and small pinion.

The gear is made of cast iron with two shrunk forged-steel rims. The wheel is 8 feet 3½ inches on the pitch circle, having 398 teeth, double helical with a circular pitch of 0.9854 inch. The wheel has a total face width of 24 inches, and the teeth have an inclination of 20 degrees to the axis. The pinion shafts are made of chrome-nickel steel, and have a diameter of 5 inches on the pitch circle, with 20 teeth at 0.9854 circular pitch, the ratio of the gears being 19.9 to 1.

Fig. 2 shows a plan and elevation view of the turbine and reducing gear of the steamer "Vespasian." The weight of the turbine installation was 25 tons less than the reciprocating units which were displaced.

The propelling machinery consists of two turbines in "series," that is, one high pressure and one low pressure, the high-pressure turbine being placed on the starboard side of the vessel and the low-pressure turbine on the port side. At the after end of each of the turbines a driving pinion is connected, with a flexible coupling between the pinion shaft and the turbine, the pinion on each side

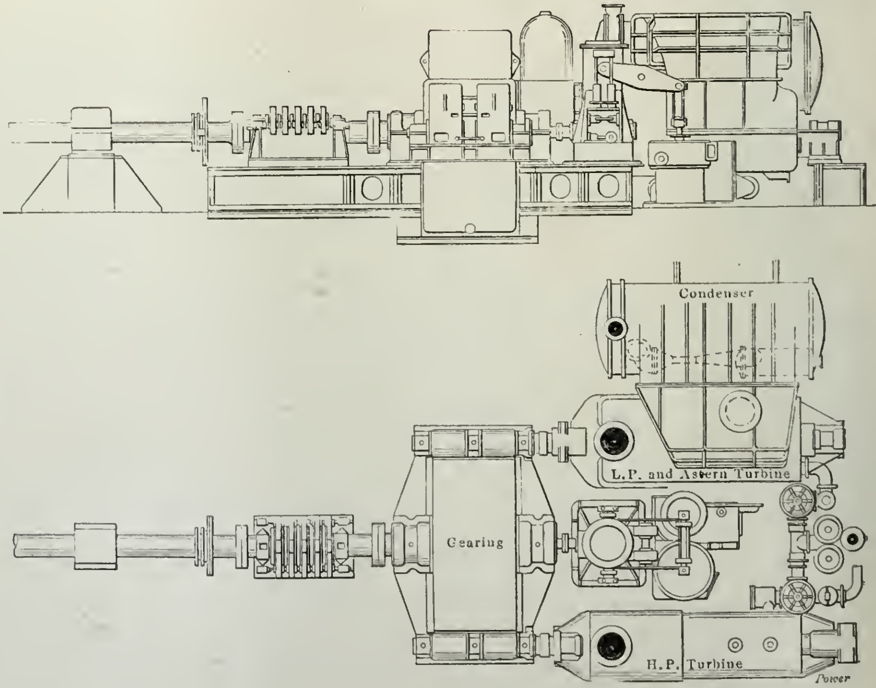


Fig. 2. Plan and Elevation of Turbine and Reducing Gear

of the vessel being geared into a wheel, which is coupled to the propeller shaft.

The following table gives the data and results of a run made off the Tyne on April 11 this year, at varying revolutions:

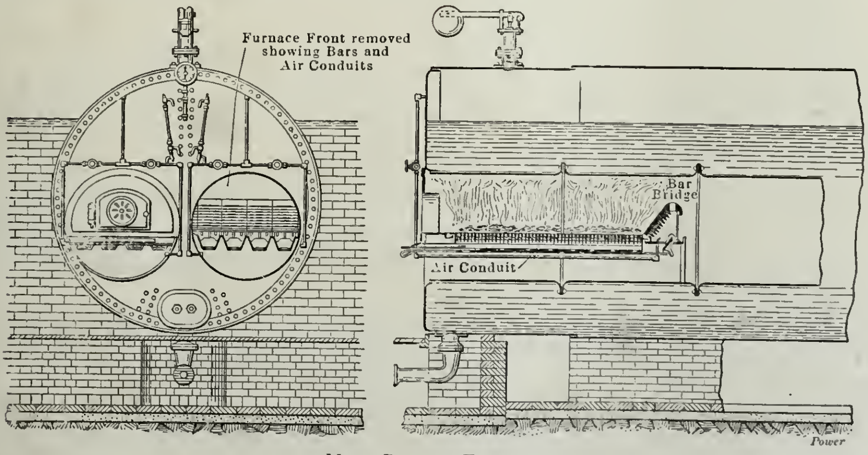
Speed in knots	8.4	9.56	10.5	10.66
Revolutions per minute	56.5	65	71.3	73.3
Boiler pressure	145	144	140	145
Initial pressure high-pressure turbine, lb	60	86	110	121
Initial pressure low-pressure turbine, in. hg	15.2	12.5	7.1	5.5
Vacuum, in. hg	28.8	28.8	28.7	28.5
Barometer, in. hg	29.9			
Shaft horsepower	456	740	980	1,095
Water consumption per hour, main engines, lb	9,070	12,000	14,480	15,670
Water consumption, all purposes, lb	9,670	12,620	15,120	16,370
Water consumption per shaft horsepower, main engines, lb	19.8	16.2	14.8	14.3

A reversing turbine is incorporated in the exhaust casing of the low-pressure turbine. The turbine and pinion-shaft bearings are under forced lubrication, similar to ordinary turbine practice. The teeth of the pinions and of the gear wheel are lubricated by means of a "spray" pipe extending the full width of the face of the wheel. Independent oil pumps are fitted for supplying oil to the bearings and gear wheel.

The high-pressure turbine is 3 feet maximum diameter by 13 feet over-all length, and the low pressure 3 feet 10 inches diameter by 12 feet 6 inches length. The turbines are similar in design to a land turbine, being balanced for steam thrust only; the propeller thrust is taken up by the ordinary thrust block of the horseshoe type, which is fitted aft of the gear wheel.

A New Boiler Furnace

Although a fireman who understands his work can, under general conditions, fire boilers of the Lancashire pattern with economy and without producing an undue amount of smoke, there are cases when the provision of some additional apparatus to effect one or both of these results is beneficial; for instance, when the draft is deficient or when the boiler is too small for its work.



Hill Boiler Furnace

The boiler furnace illustrated herewith is, according to *The Engineer*, being made by Messrs. J. and P. Hill, of Park, Sheffield. The furnace belongs to that class in which the combustion is assisted by means of induced draft produced by a series of small steam jets. The chief features of its construction will be seen in the accompanying drawings. They consist of cast-iron troughs, or conduits, arranged longitudinally along the furnace between the front and back dead plates.

Transversely across these conduits—which are between 8 inches and 9 inches wide, according to the diameter of the flue—short fire bars are placed. The conduits are open at the front end to admit air through the bars, but are closed at the rear end. Their bottom sides are perforated with holes corresponding to other holes in hand-operated slides. These slides permit the stoker to regulate the air supply and riddle the fine ash into the flue. The bars are of special form and are placed much more closely together than is usual, the air spaces being not more than ⅛ inch wide. The bridge is really a continuation of the grate, the bars being specially constructed to fit cast-iron boxes forming the bridge proper, the bridge being finished off at the top by means of a cast-iron cap secured by two prongs that span the box.

This cap has cast in it a large hole through which the mixture of steam and air passes into the stream of gases passing over the bridge. The dead plate is bolted to the furnace front, these being the only bolts used in the erection of the furnace, and the conduits have flanges which fit into recesses in both the front and end dead plates. The boxes forming the bridge are secured together by means of bolts, with which the flames do not come into contact. The draft is induced along the conduits by means of a series of small steam jets, and, the boxes being closed up at the end farthest from the front, the air and steam pass vertically up through the bars, causing the flames to take an almost vertical course before passing over the bridge. The smoke and products of combustion are dealt with by means of a further steam spray through the bridge above mentioned, the amount of air being variable at will by means of the valve shown in the back dead plate. This valve is mounted on spigots, and can be opened or closed at will by means of a handle in front. It will be observed from the sectional views that there are no extraneous pipes or other fittings in the flues to give trouble, and that no drilling of the boiler front is necessary. The whole of the parts can be put in, or removed, in a few hours.

NEW PUBLICATION

The Chemist's Pocket Manual (second edition). By Richard K. Meade. The Chemical Publishing Company. Easton, Penn., 1910. Leather, 455 pages, 4x6 inches; 39 illustrations; numerous tables. Price, $3.

Presumably the mathematical tables in this edition were also in the first one (with which the reviewer is not familiar); if so, it is regrettable that some kind friend did not inform the author of their uselessness. Of what practical value is a three-place table of logarithms, or a table of trigonometric functions in one-degree intervals, or a table of roots and powers of numbers ranging from 1 to 100 by units, or—but further enumeration is unnecessary. These useless tables do not detract from the value of the chemical data contained in the book, but they occupy space unprofitably. The tables which are useful are arranged in a sort of haphazard fashion, no definite system of arrangement being evident. Many of the charts in the book are called "tables," some of them are not labeled at all and nearly all of them are poorly executed, as to the mechanical reproduction. Twaddell's name is spelled incorrectly, being disrespectfully written "Twaddle." On page 110 "carbon dioxide" is written for "carbon monoxide," and several similar evidences of carelessness are noticed by the reviewer in various parts of the book.

The manual contains a vast quantity of highly useful material, but the arrangement and treatment of the subject matter are surprisingly amateurish.

BOOK RECEIVED

"Factory Organization and Administration." By Hugo Diemer. McGraw-Hill Book Company, New York. Cloth; 317 pages; 6¼x9¾ inches; 150 illustrations; indexed. Price, $3.

Meeting of the A. I. M. E. in the Canal Zone

The members of the American Society of Mechanical Engineers have been invited to join with the American Institute of Mining Engineers during their visit to the canal zone, October 21 to November 15, so far as accommodations will permit. The party sails from New York on October 21 on the Hamburg-American liner "Prinz August Wilhelm," touching at Havana and Kingston *en route* to Colon. A full week will be spent in the canal zone, during which time the party will inspect the work on the canal in addition to holding professional sessions. The cost of the trip, including berth and meals on the steamer, railroad transportation, meals and accommodations at the Hotel Tivoli in the canal zone, will be $200. This rate holds good until September 20, after which an additional charge of $50 will be made by the steamship company. Further details regarding the arrangements for the trip may be obtained from Dr. Joseph Struthers, assistant secretary of the Institute.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

WINDMILL. Robin W. Vaughan, Los Angeles, Cal. 968,003.

INTERNAL COMBUSTION ENGINE. Chas. Y. Knight, Oak Park, Ill., assignor of one-half to Lyman Bernard Kilbourne, Chicago, Ill. 968,166.

INTERNAL COMBUSTION ENGINE. Arthur Forbes Scott, Bradford, Eng. 968,200.

TWO-CYCLE GAS ENGINE. Raymond Jacob Leach, Seekonk, Mass. 968,244.

STEAM TURBINE. Jan Procner, Pabianice, Russia. 968,262.

ROTARY ENGINE. Albert C. Savidge, Sunbury, Penn. 968,268.

INTERNAL COMBUSTION ENGINE. Timothy J. Shanahan, Elkhart, Ind., assignor of one-half to Ewing C. Trott, Chicago, Ill. 968,274.

INTERNAL COMBUSTION ENGINE. Thomas A. Martin, Chicago, Ill., assignor, by mesne assignments, to Sears, Roebuck & Co., Chicago, Ill., a Corporation of New York. 968,380.

FLUID-PROPELLED DIRECT-ACTING ENGINE. Wilhelm Mauss, Brakpan, Transvaal. • 968,381.

BOILERS, FURNACES AND GAS PRODUCERS

STEAM BOILER. Alfred A. Olson, Riverside, Ill. 967,961.

FLUID HYDROCARBON BURNER. David P. Moore, Washington, D. C. 968,252.

OIL BURNER AND DRAFT APPLIANCE. Fred A. Stevens, San Francisco, Cal. 968,281.

GAS PURIFIER. Henry I Lea, Pittsburg, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 968,370.

POWER PLANT AUXILIARIES AND APPLIANCES

IGNITER FOR EXPLOSION ENGINES. Elbridge W. Stevens, Baltimore, Md. 968,280.

WATER-SOFTENING APPARATUS. Kent W. Bartlett, Madison, Wis., assignor to Northern Water Softener Company, a Corporation of Maine. 966,071.

VALVE. Samuel H. Woodbridge, Newton, Mass. 966,150.

WATER-SEALED PRODUCER VALVE. Franklin G. Hobart, Beloit, Wis., assignor to Fairbanks, Morse & Co., Chicago, Ill., a Corporation of Illinois. 967,919.

FURNACE DOOR. Carl J. F. Johnson, Cleveland, Ohio. 967,927.

FEED-WATER HEATER FOR BOILERS. Willard C. Barnes, South Weymouth, Mass. 968,023.

BLOWOFF COCK. Ralph W. Erskine, Pittsburg, Kan. 968,037.

PUMP. William Henry Clark, Hoboken, N. J. 968,126.

OIL BUSHING AND PACKING RING. James J. Coyne and David R. Llewellyn, Shamokin, Penn. 968,132.

CARBURETER. Walter C. Westaway, Rockford, Ill., assignor, by direct and mesne assignments, to J. W. Duntley, Chicago, Ill. 968,215.

PULLEY GUARD. George H. Winslow, South Hadley, Mass. 968,304.

PLUG VALVE. Thomas Keenan, Brooklyn, N. Y., assignor to Keenan Regulator and Steam Fitting Company, New York, N. Y., a Copartnership. 968,362.

PACKING GLAND OR COLLAR. Jules Naeder, Seine-et-Oise, France. 968,390.

CRANK AND YOKE CONNECTION. Harlan P. Wellman, Ashland, Ky. 968,437.

DEVICE FOR GOVERNING AND REGULATING STEAM ENGINES. Hans Christian Holm, Brooklyn, N. Y. 965,904.

HOSE COUPLING. Taylor Evans, Centralia, Wash. 965,998.

EJECTOR. Miles B. Parsons, Birmingham, Ala. 966,048.

FLUE CLEANER. Elza H. Reiter, Elgin, Ill. 966,051.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC FURNACE. Johannese Harden, London, England, assignor to the Gröndal Kjelin Company, Ltd., London, Eng. 967,909.

ELECTRIC FURNACE. James Henry Reid, Newark, N. J. 968,079.

ELECTRICAL POWER TRANSMISSION. John Godfrey Parry Thomas, Chiswick, England. 968,290.

DYNAMO-ELECTRIC MACHINE. Henry G. Reist, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 968,420.

CIRCUIT-CONTROLLER-RETARDING DEVICE. Edward A. Halbleib, Rochester, N. Y. 968,468.

SERIES LAMP RECEPTACLE. Charles D. Gervin, New York, N. Y., assignor to John H. Dale, New York, N. Y. 966,681.

SHORT-CIRCUIT DETECTOR. Frederick C. Reineking, Jersey City, N. J., assignor to Rankand Company, a Corporation of New York. 966,766.

BRUSH HOLDER FOR DYNAMOS. Ernest C. Ketchum, Boston, Mass. 966,839.

ELECTROMAGNETIC SEPARATOR. Frank J. Phillips, Chicago, Ill. 966,855.

ELECTRIC-LAMP SOCKET AND SECURING MEANS THEREFOR. William C. Tregoning, Cleveland, Ohio, assignor to the Tregoning Electric Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 966,966.

ELECTRIC WATER HEATER. Herbert N. Roche, San Francisco, Cal., assignor to Thomas B. Gray, San Francisco, Cal. 967,058.

MISCELLANEOUS POWER PLANT TOOLS

WRENCH. Martin K. Olson, Louisburg, Minn. 967,962.

JACK. John Mosiman, Sedalia, Mo. 968,254.

RATCHET WRENCH. James A. Repass, New Augusta, Ind., assignor of one-fourth to Henry H. Hammer and one-fourth to Ernest M. Hornaday, Indianapolis, Ind. 968,080.

WRENCH. John J. O. Dethlefs and Willis Wisner, Chehalis, Wash. 967,889.

WRENCH. William E. Golden, Fremont, Ohio. 967,899.

WRENCH. Oscar Zimmerman, St. Helen, Ore. 968,106.

JACK. Le Roy Willour, Ashland, Ohio, assignor to the Ashland Manufacturing Company, Ashland, Ohio, a Corporation of Ohio. 968,223.

VISE. Lawrence R. Blackmore, Arlington, N. J. 968,444.

CHAIN TONGS. Philip Farrel, Flushing, N. Y. 968,335.

TOOL FOR SPREADING BOILER TUBES. August P. Gerald, Jersey City, N. J. 968,338.

TONGS. George E. Thurston, Colby, Wash. 967,463.

WRENCH. Lotte Kovatch, Oliveburg, Penn. 967,788.

WRENCH. Malcolm G. Ewer, Detroit, Mich. 967,355.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power
and The Engineer

Vol. 32 NEW YORK, SEPTEMBER 20, 1910 No. 38

THE engineer or fireman who has fallen into the habit of finding fault with everything about his plant is about as disagreeable a companion as can be found.

Some men make themselves so obnoxious by continual fault finding that it is a relief to their fellow workmen when their watch is over.

This fault is generally the outcome of some fancied or real grievance, generally fancied. The injured feelings of the man finally develop into chronic "grouch," and when this stage of mental disorder has been reached nothing short of a severe lesson will bring relief, and often the lesson will produce no results.

Some men work under conditions that are well-nigh unbearable, but even in such cases many of the disagreeable features could be removed if they were given proper attention.

It does not take a visitor long to detect the chronic "kicker," as he begins to portray the disadvantages of this and that, magnifying the smallest faults and many times ignoring those of greater magnitude.

A case in point is as follows: A certain boiler room has a very smooth concrete floor which, when damp, is quite slippery; otherwise the boiler room is ideal, being large, light and well ventilated.

The only real "kick" to which the fireman is justly entitled is on account of the slippery concrete floor, which would probably prove more satisfactory had its surface been corrugated.

But instead of working out some means for overcoming this condition, such as nailing a piece of old rubber to the heels of his working shoes, the fireman when talking with a stranger, gives the impression that the boiler room is hardly a fit place to work in.

"Kicking" over trivial matters that can be easily remedied is little short of childish. Most men would find a way of quietly overcoming such difficulties.

It is poor policy to criticize to a stranger, or a friend for that matter, the plant in which you are working, because you are telling a disinterested party something that is, in a way, a family affair and should be so considered.

The stranger may be a director of the company; he may be a good engineer who will see that the fault is not altogether in the plant. Or the hearer may be a manufacturer "sizing you up" to determine whether you are a fit person to put in charge of his steam plant.

If an engine is worn out don't forever "kick" about it, but get it in decent running condition. Many an engineer has made a name for himself by doing this very thing.

To be "grouchy" or to complain continually does not pay. Friends are not gained in this way, and enemies multiply. If it is impossible to say a good word about the chief, the plant, or the firm you are working for, say nothing.

Mum is the word.

Metropolitan Home Office Plant

By F. L. Johnson

The power plant of the tallest office building in the world is equipped to give the widest range of service. Besides the regular features of heat, light and ventilation, there are refrigeration, cold storage, filtered air and water, and a host of minor conveniences demanded by the present-day tenant. A feature is the unique system of keeping records of the various departments.

If an engineer should be asked to design a power plant for a modern office building which was to stand on a piece of ground the value of which could be closely approximated by covering it with fifty-dollar gold certificates, it is highly probable that an efficient and reliable installation would be sought. It was a problem very much like this that was solved when the power plant for the home-office building of the Metropolitan Life Insurance Company was designed. In an office building there are so many kinds of service to be considered that each must be treated in its relation to all of the others. At the outset is the power problem. This includes everything requiring the expenditure of energy, and when the requirements are as carefully approximated as is possible, the relation of this feature to the heating must be considered. The power must be furnished for the entire year, while heat is needed for only a portion, and if the steam machinery selected will furnish exhaust steam sufficient for heating during the coldest weather, it will be extravagant in the amount of steam used per horsepower-hour. The problem to be solved is to correlate the heating requirements with the motive-power machinery so that the total fuel used for both heating and power purposes shall be the least quantity possible.

In the equipment of this building it was intended that for several months of the year the exhaust-steam supply for heating should be supplemented by steam from the boilers through automatic pressure-reducing valves. This arrangement, while apparently increasing the amount of live steam taken from the boilers in the cold season, made it possible to generate the power needed in the summer months by economical units, so that a measurable degree of economy is secured.

The main building occupies an entire city block, 425x200 feet, which, with its eleven stories and the 75x85-foot tower at one corner reaching a hight of 700 feet above the sidewalk, contains upward of 25 acres of floor space.

Steam is furnished by eleven Babcock & Wilcox boilers in the subbasement, having a total capacity rating of 3500 horsepower. They are arranged on either side of broad alleys, and are served with con-

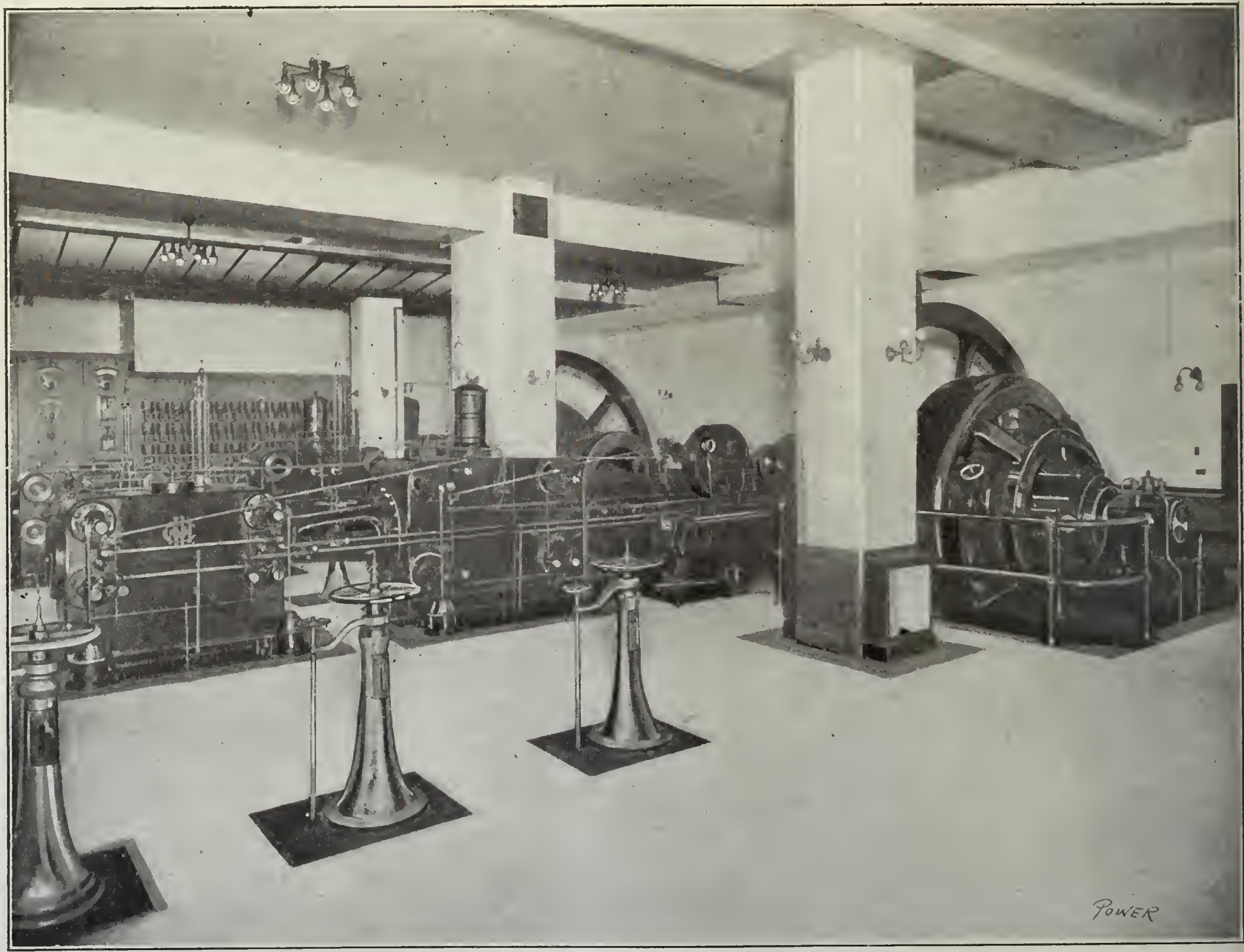


Fig. 1. View of Engine Room from Visitors' Gallery

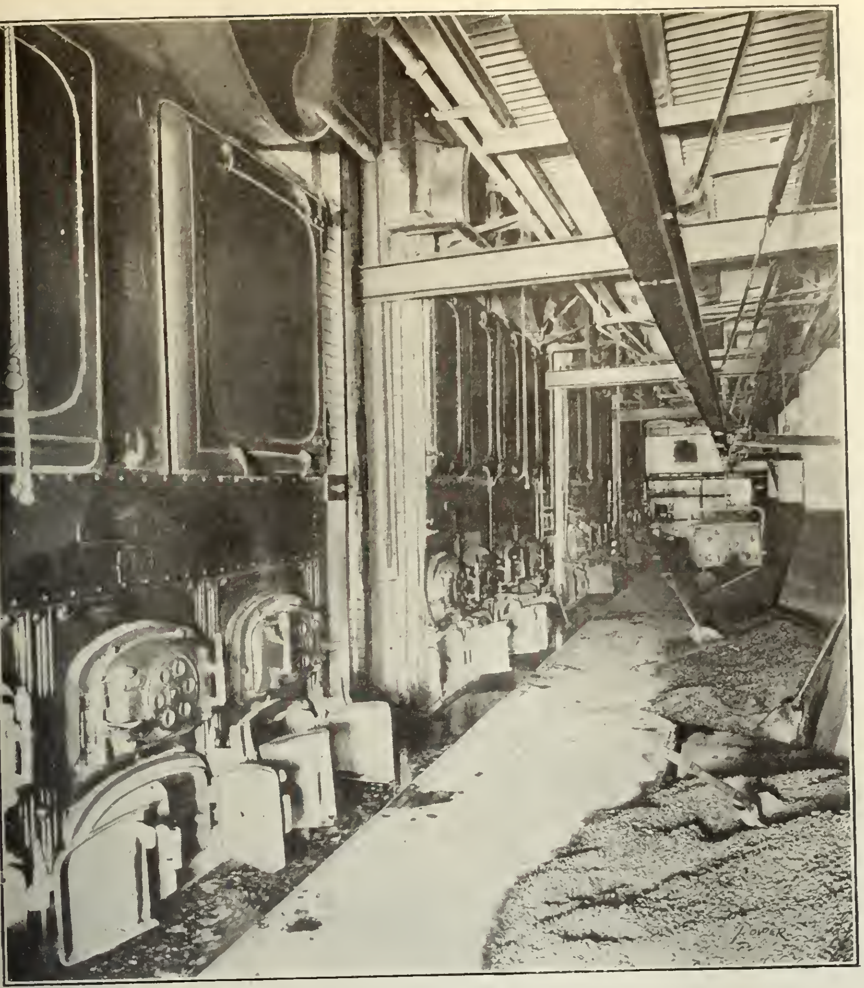


FIG. 2. A VIEW IN THE BOILER ROOM

veniently arranged tracks, both on the floor and overhead, for the handling of coal and ashes. The coal and ash tracks lead to the bunkers of 4000 tons capacity under the sidewalks and to the ash elevators. Each boiler is independently connected to a steam gage on the gage board in the engine room, which shows plainly to the engineer on the floor above at all times the number and location of every boiler under pressure. There are also on the gage board a Bristol recording steam gage and a recording thermometer connected to the uptake leading to the chimney.

The boilers are equipped with superheaters furnishing steam through the comparatively long pipe under the engine-room floor with 50 to 60 degrees of superheat at the engines.

In the engine room there are five compound noncondensing engines, each having a direct-current generator mounted on the shaft. One is an Allis-Chalmers cross-compound of 650 kilowatts capacity; the other four are Rice & Sargent tandem compounds—two of 200, one of 300 and one of 400 kilowatts capacity. There are in the same room three three-cylinder compound noncondensing Laidlaw-Dunn-Gordon high-pressure elevator pumps, each having a capacity of 720 gallons per minute against a pressure of 800 pounds per square inch. There is also a compound duplex Worthington direct-acting elevator pump of 300 gallons capacity per minute against the same pressure, and a pilot-valve pump which furnishes the water for the operation of the elevator pilot valves at a pressure much below the 800 pounds per square inch operating on the elevator plungers. The high-pressure pumps deliver the water directly to the pressure tanks which are connected to six weighted accumulators located in an alcove adjoining the engine room.

One interesting feature of the plant is the mammoth report chart in the chief engineer's office. It is a graphic record of each day's work taken from the reports and put on the chart, which shows the relation of every factor in the plant to every other one. It consists of a "board," about 12 feet long and 4 feet high, on which the different kinds of service are represented by colored twine passing over and held in place by tacks driven into the "board" at the proper points. In the illustration near the top of the board *B* is the temperature base line from which the exhaust and live-steam lines *A* and *C*, respectively, are plotted. All of the lines above *B* represent exhaust steam and the line *C*, which is below *B*, the live steam used to supplement the exhaust when needed. The line *D* is the outdoor temperature curve and shows how closely the heating requirements follow this temperature. The line *E* shows the number of kilowatts generated, each point in the line representing the output for a single day. The consumption of city water for all purposes is represented by the line *F*, and its relation to the other lines on the board may be easily traced, particularly in the case of the line *G*, which records the amount of fuel used.

From the Twenty-third street postal substation, which is in the Metropolitan building, to the general post office downtown, the mail is transmitted in pneumatic

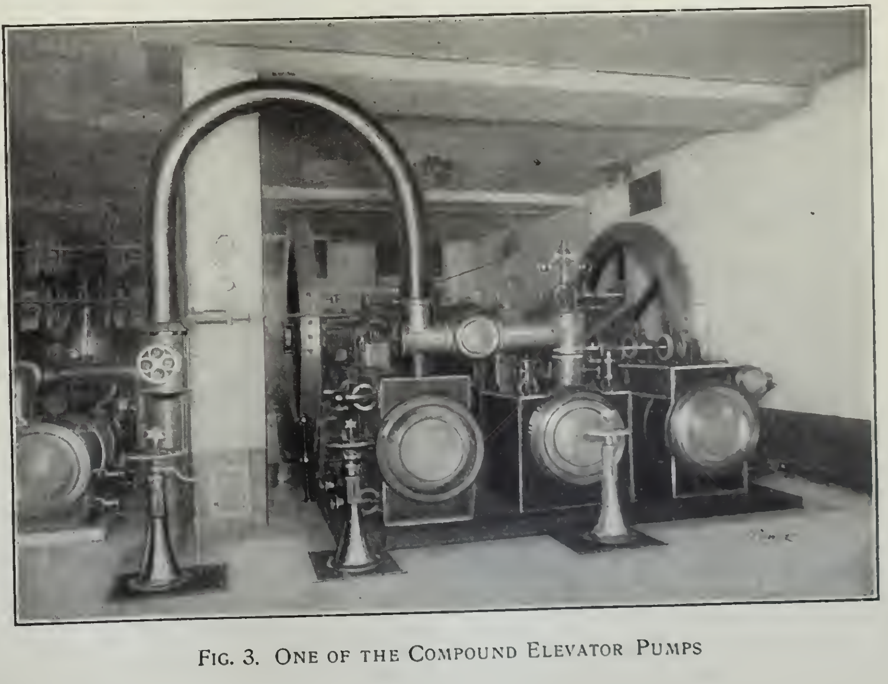


FIG. 3. ONE OF THE COMPOUND ELEVATOR PUMPS

tubes. The air is furnished by the plant and the power used from day to day is recorded by the line *H*.

Elevator service, which means a summary of the number of cars in service and a totality of trips, is shown by line *I;* and the labor line, which is more regular in its irregularity than any of the others, is designated by the letter *J*. Refrigeration is marked by the line *K*, while the comparative hights of the electrical peak loads for each day are recorded by the base line *L*.

Each of the eight lines below the outdoor-temperature line is calculated from different base lines which are not shown. The highest and lowest points reached each week give a clear idea of the range over which each service operates.

There are many little "kinks," so called, in different parts of the plant which are of more than passing interest to the observer. One is a ball-bearing swivel cable connection on the end of the elevator cables, which prevents any tendency to strain from the twisting and untwisting of the cable that takes place at every change of tension during every trip of the car. The device is the invention of Charles Bavier, chief engineer of the plant. It was, however, found to be unpatentable because years before an artillery officer had used the same device to allow a chain to rotate without kinking. Its earlier application has one compensating feature, the engineer can use his own device without having to pay a royalty to some patent-owning shark, as many another has.

In the laundry there is an efficient washing machine constructed along original lines. It consists of a sheet-copper

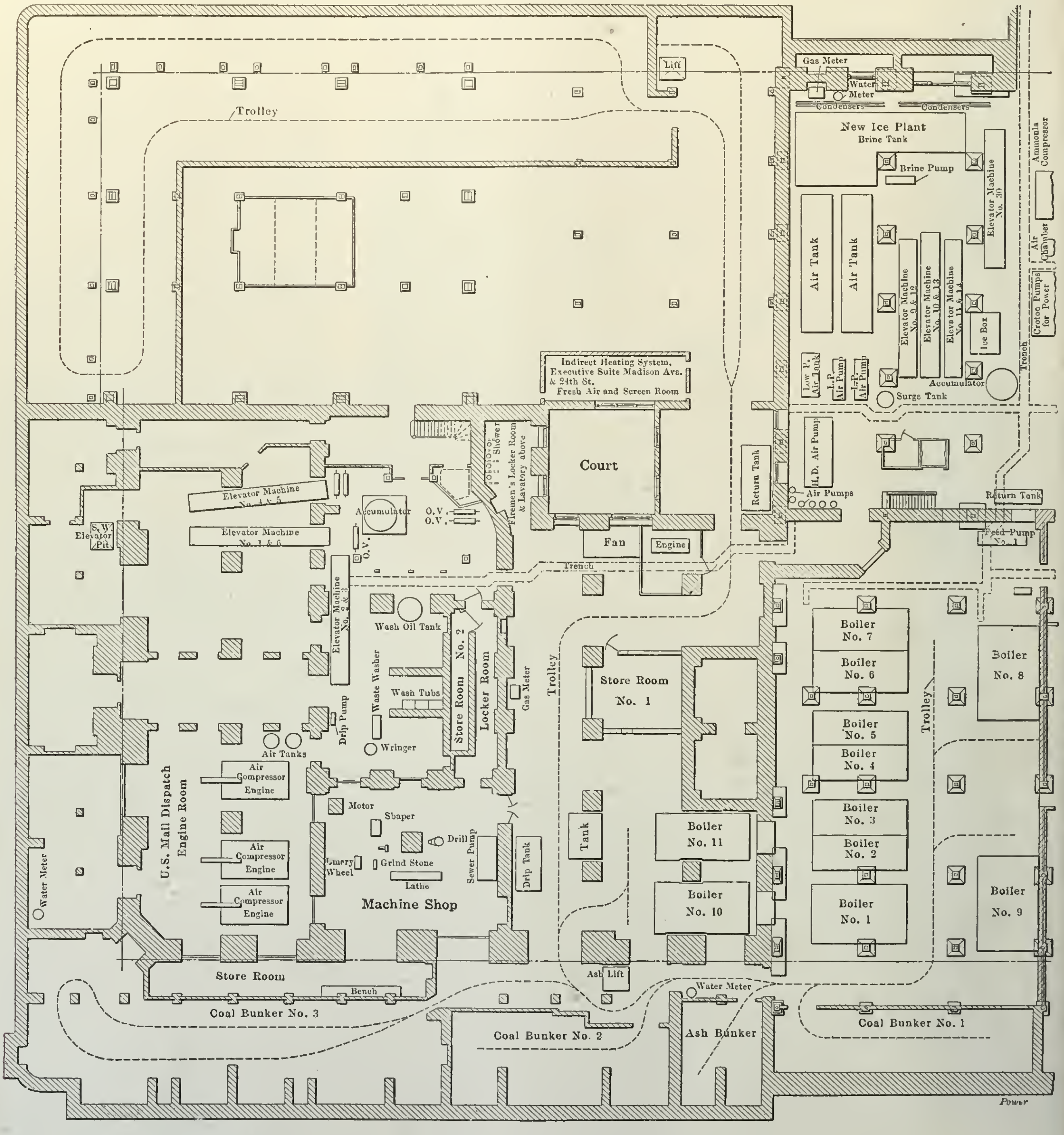


FIG. 4A. SUB-BASEMENT PLAN SHOWING LOCATION OF FOUNDATIONS AND MACHINERY

box about 5 feet long and 1 foot square with tapering ends. It is supported on trunnions at the middle of its length and when in use is slowly revolved by a small electric motor. Whatever is to be washed is put inside with soap, or washing powder, as may be desired, and cold water. The cover is put on and the motor started. At the end of an hour or a day, or any other period, the box is opened and the contents removed. As the box revolves end over end, the contents drop from one end to the other, the pyramid-shaped ends acting to give a gentle squeeze at the end of each drop and to turn over the garment, or whatever is inside, at each change of direction. As a washing machine, the kink is perfect. The smooth sides, tapered ends and slow motion drive the water through the fabric in a different direction at every turn and cause practically no wear.

Near this machine are the drier and the steam-heated ironing table and pressing roll.

To the sightseer the tower clock is an interesting feature. Located some 350 feet above the street it is visible over a great part of the city. Some of the details are quoted.

"The dials are built up of reinforced concrete faced with vitreous blue and white mosaic tile. Each dial is 26 feet 6 inches in diameter. The figures on the dial are 4 feet high and the minute marks 10½ inches in diameter.

"The minute hand measures 17 feet from end to end, 12 feet from center to point, and weighs 1000 pounds; the hour hand measures 13 feet 4 inches from end to end, 8 feet 4 inches from center to point and weighs 700 pounds. They are built on iron frames, sheathed with copper, and revolve on roller bearings.

"The driving power of this huge mechanism is electricity; none of the many devices connected therewith require any manual operation, the entire installation being automatic.

"The master clock, located in the di-

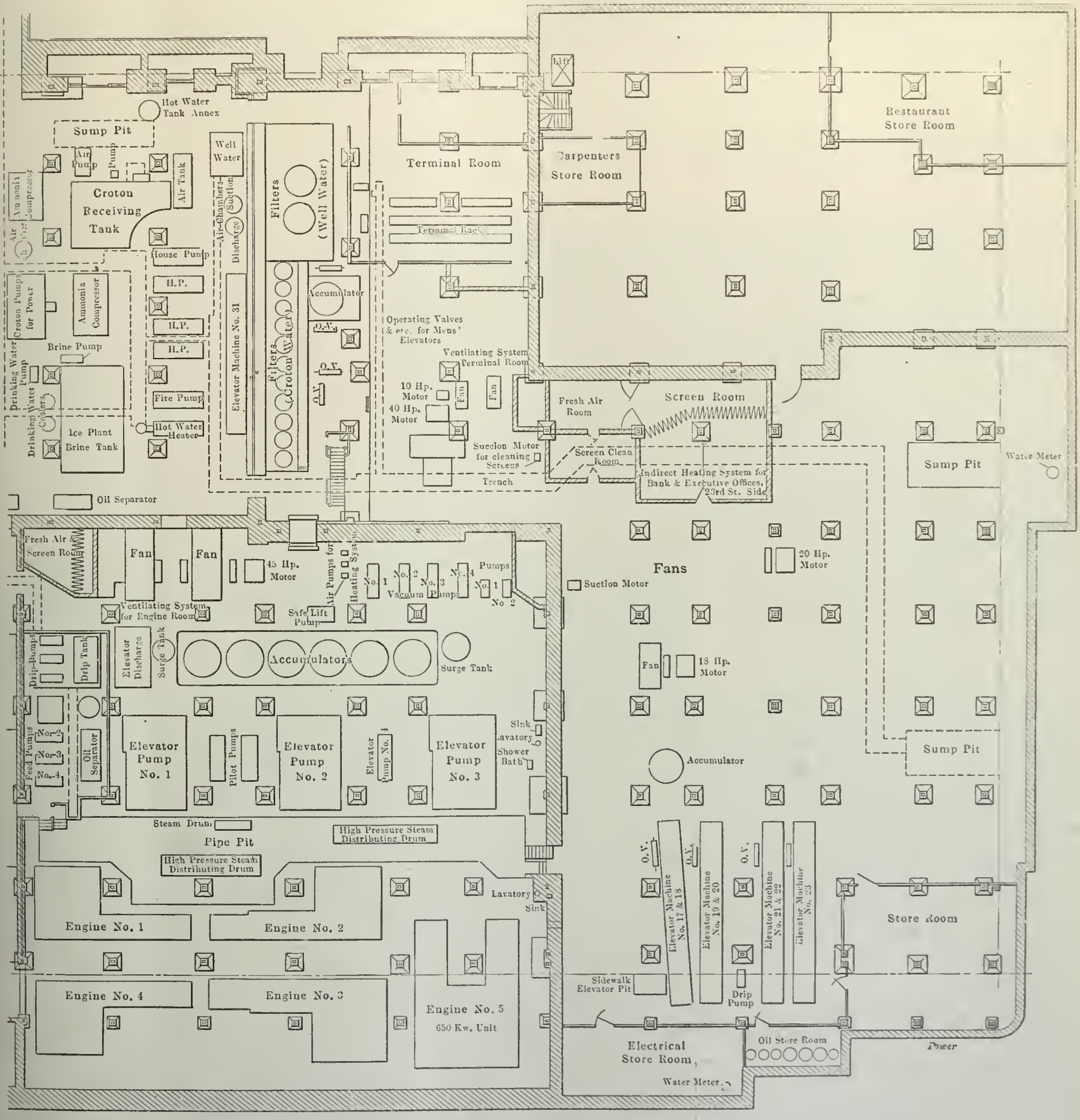


Fig. 4b. Sub-basement Plan Showing Location of Foundations and Machinery

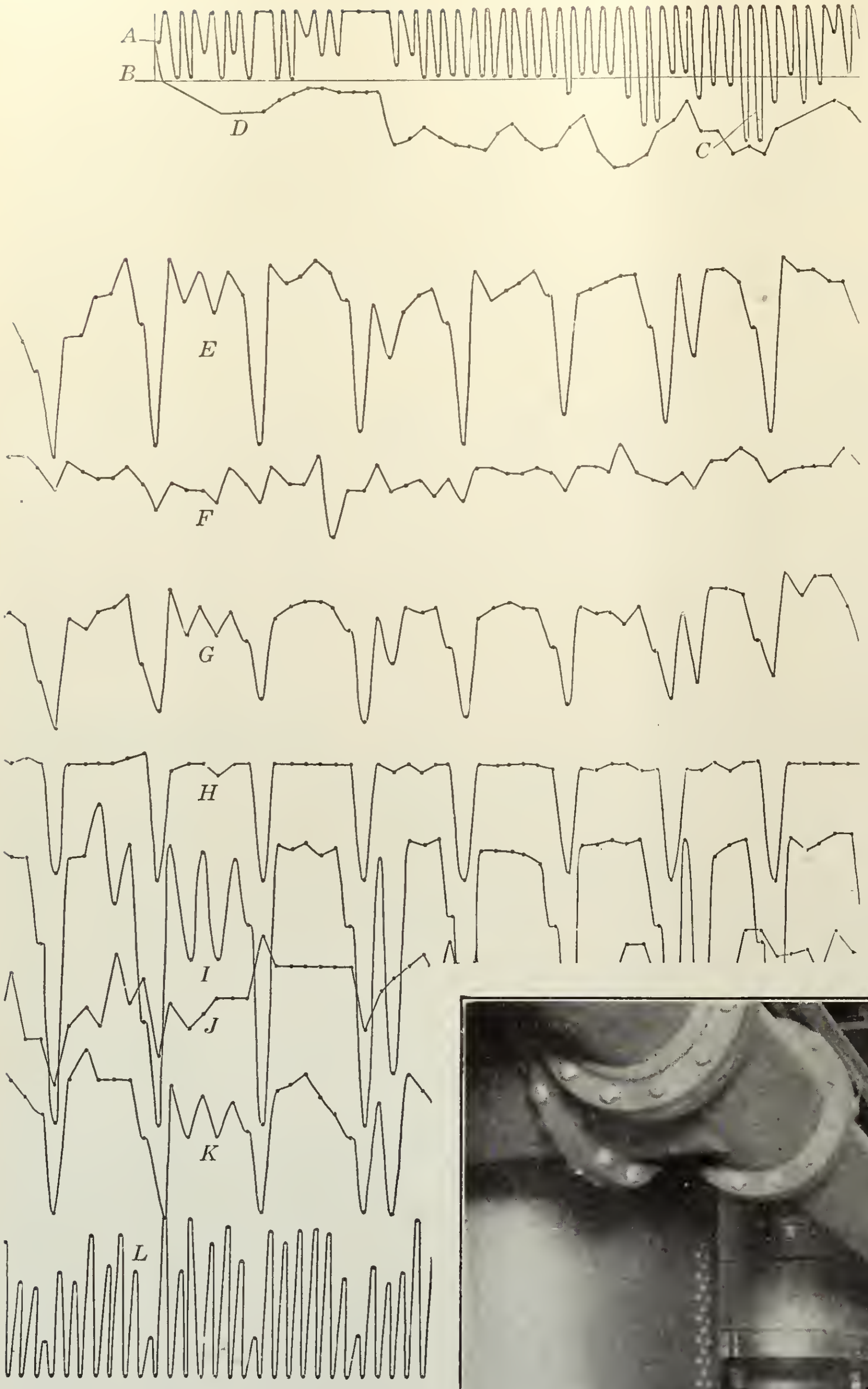


Fig. 5. Eight Weeks' Daily Individual Service Record

rectors' room on the second floor, not only controls the entire tower-clock outfit, but about 100 other clocks throughout the building, as well as several program instruments for sounding various schedules of bells in the different departments.

"Through the medium of a special transmitter, minute impulses are sent to the tower-clock mechanisms on the twenty-sixth floor, keeping them in exact synchronism with the master clock; and at each quarter-hour electrical impulses are transmitted to the electric hammers on the forty-sixth story, and simultaneously are heard the notes of the old historic Cambridge chimes, composed by Handel.

"Simultaneously with the illumination of the hands and dials at night, an automatically actuated switch lights up a great electric octagonal lantern, 8 feet in diameter, located at the top of the tower, from which powerful electric flashlights, marking the hours in the evening, may be seen for a great distance, far beyond any possible transmission of sound, the time being signaled therefrom as follows:

"Each of the quarter-hours is flashed in red and the hours in white light. One red flash for the quarter, two red flashes for the half, three red flashes for three-quarters, and four red flashes for the even hour—these latter flashes followed by a number of white flashes marking the hour."

The lower floors are heated by an indirect system in which all of the air entering the rooms is drawn by fans from the court through screens and forced over and through the concealed radiators, furnishing a constant supply of pure, dustless air at all times. The upper floors are heated by a direct-radiation two-pipe exhaust-heating system into which live steam is automatically admitted whenever the demand exceeds the exhaust-steam supply. Circulation in the heating system is maintained by the vacuum pumps in the engine room.

As the modern office building is intended to be adapted to almost any kind of business a great variety of service is re-

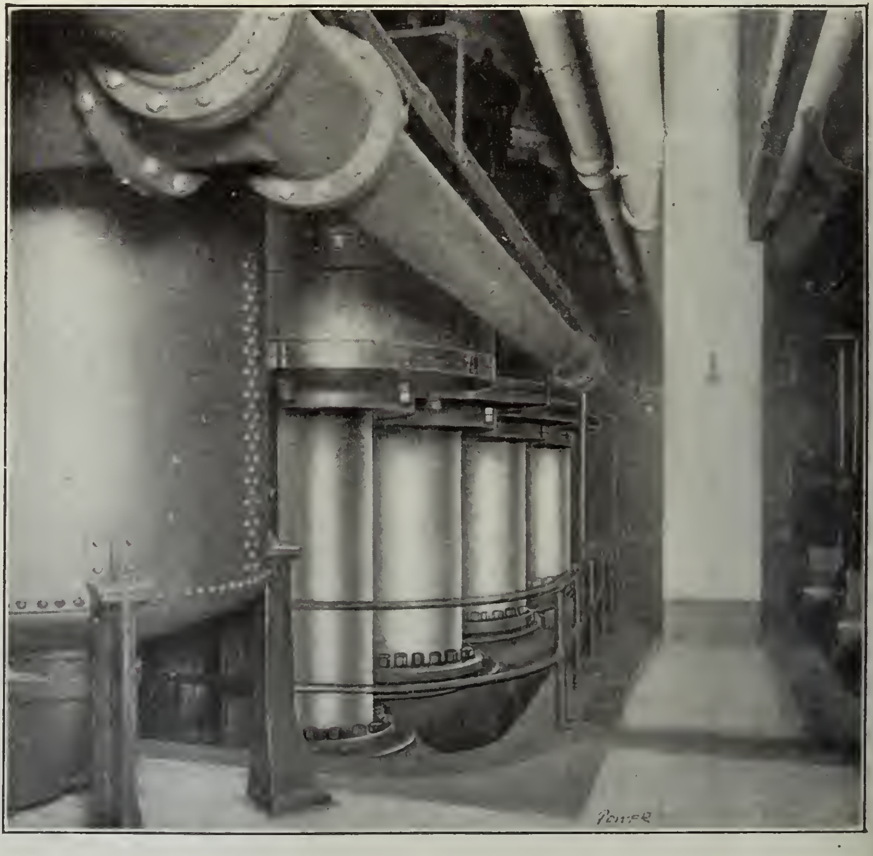


Fig. 6. A Few of the 80-ton Accumulators

quired, and the Metropolitan is no exception. If it is different from others of its class, it is in the greater extent to which the diversity of service is carried.

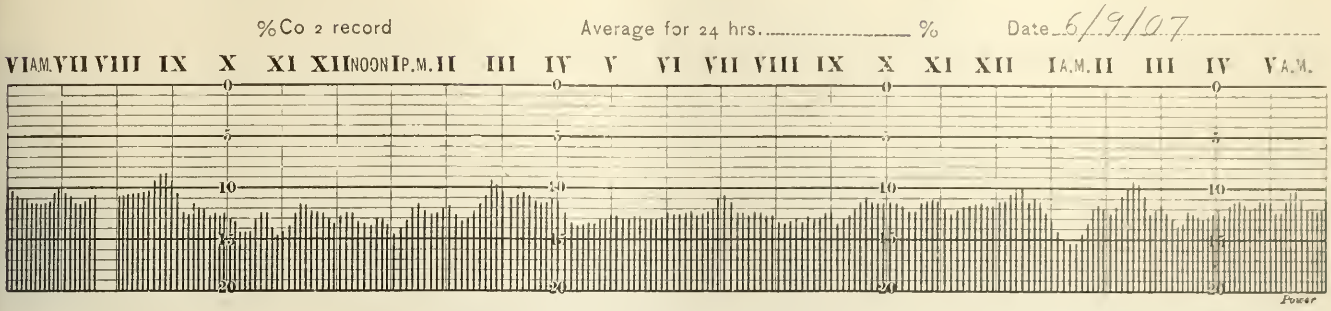


Fig. 7. Chart from Ados CO_2 Recorder

The United States mail service requires compressed air for mail transmission to the extent of 250 horsepower. The restaurant and lunch rooms, where 2500 officers and employees of the company working in the building are furnished their daily noon-hour meal, demands steam, hot water, cold water and ice. The laundry has its unique equipment and a part of the work of the 40-ton refrigerating plant is the preservation of cold-stored furs from the destructive attacks of moths.

In plain sight of the firemen is an automatic Ados continuous CO_2 recorder.

One other detail of the plant that arrests the attention of the visiting engineer is a large steam-driven Richardson force-feed oil pump attached to one of the pillars in the ice-machine room with oil pipes supplying all of the cylinders in the refrigerating department. Other pumps of this type with single and double connections are used throughout the plant on all of the pumps and engines but this is the only one independently driven and supplying the requirements of a whole department.

Unlike many other engine rooms, visitors are always welcome, and for those who care only for a general or passing view, a visitors' gallery is provided which is reached by a short stairway leading from the arcade on the first floor. This gallery is a plate-glass room about 12x15 feet near the middle of the engine room. Through its transparent sides a good view may be had of the show part of every power plant—the engines and pumps.

Of the 38 passenger elevators 30 are hydraulic, with the cable-operating cylinders located on the floor of the sub-basement, and eight are electric with the winding drums high up in the tower.

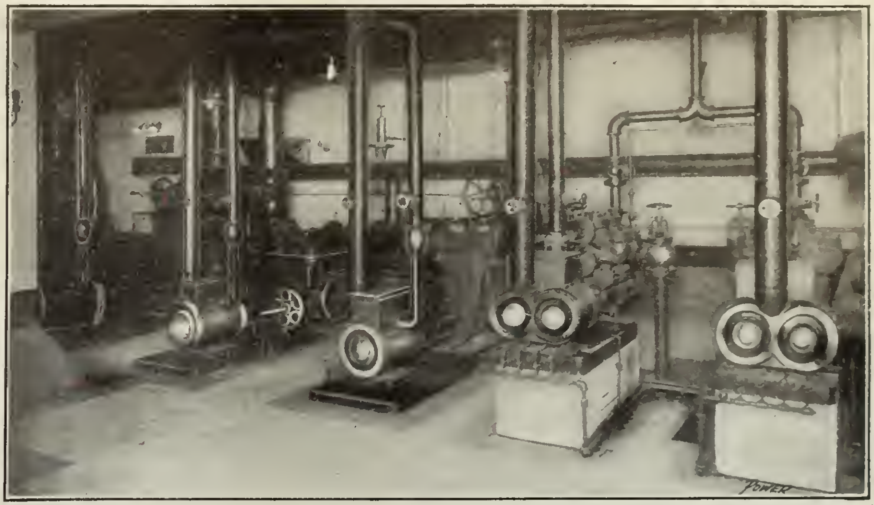


Fig. 8. Air and Vacuum Pumps for Heating System

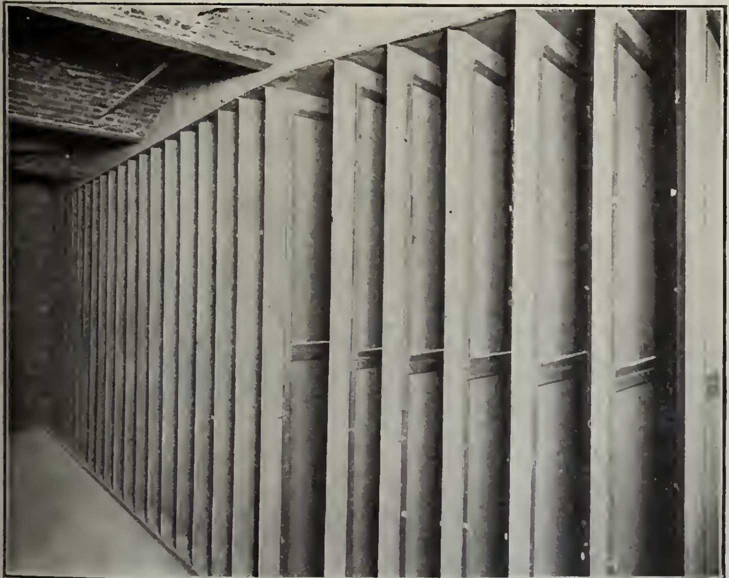


Fig. 9. Air Straining Screens

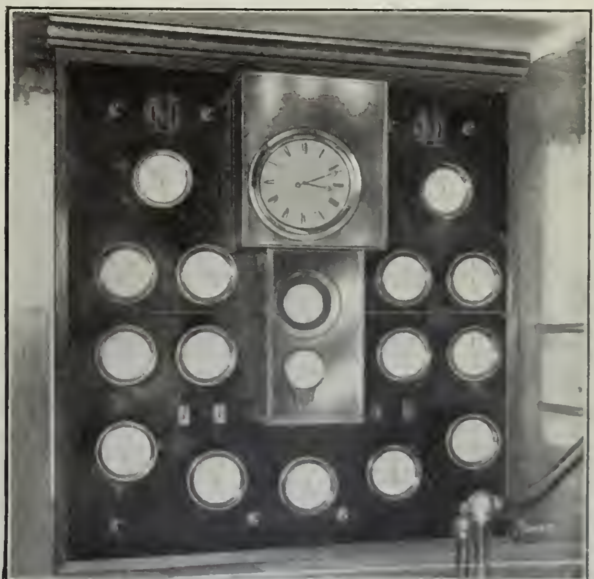


Fig. 10. Gageboard

Setting Valves of a Williams Engine

By Hubert E. Collins

The valve used on the high-pressure cylinder of a Williams engine combines features of both a balanced slide valve and a piston valve. Its operation is illustrated and directions are given for its proper setting, together with the governor adjustment.

The valve herein described has some novel features which make it of prime interest to those interested in valves and valve motions.

It is a six-ported valve having four steam ports and two exhaust ports and is also balanced, working under two pressure plates, one at each end of the valve. Another feature is that it is an indirect valve, admitting steam over the inside edges and exhausting over the outside edges. The two end members of the valve are secured together by tubes which act as supplementary steam passages for admission; in other words, it combines the features of both a balanced slide valve and a piston valve.

This type of valve is adopted on Williams single-cylinder engines and on the high-pressure cylinders of the compound engines of that make, there being more of the latter than of the former in use. On the low-pressure cylinders the valves are of the gridiron type. These latter are similar in action to the McIntosh & Seymour type which has been frequently described. It is the purpose of the writer to treat only of the single valve used on the high-pressure cylinders.

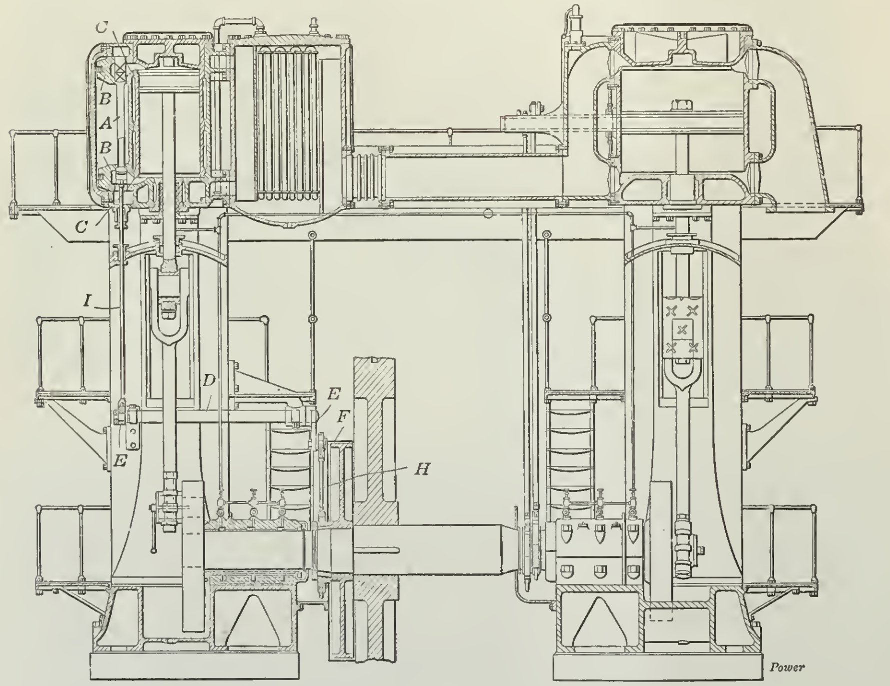


Fig. 1. Front Sectional Elevation

The front sectional elevation of a Williams engine, Fig. 1, and the end elevation, Fig. 2, show the location of this valve and its driving mechanism.

In Fig. 1 the connecting tubes of the valve are shown at *A*, the ends of the valve under the pressure plates at *B*, the cylinder ports at *C*, the rocker shaft at *D*, the rocker arms at *E*, the governor casing at *F*, to which is attached the governor eccentric *G*, the eccentric rod at *H* and the valve stem at *I*.

Fig. 3 shows this valve and gear on the end of the shaft of a medium-speed engine, the same letters designating the corresponding parts shown in the first view.

Fig. 4 is a skeleton sketch with the valve in central position. The eccentric position is shown central at point *K*, and the eccentric positions for maximum and minimum cutoff are also indicated when the crank is on the top center.

The eccentric is in position to give maximum cutoff when the weight arm of the governor is on the inner stop, and minimum cutoff when the weight arm is at the outer stop. It will be noted that when the valve is central, the bottom exhaust edge is on a line with the port, while the top edge has a clearance or negative lap. *This peculiarity should be carefully noted.* In this, and succeeding figures, the valve is represented in a horizontal position for the purpose of convenience, but it should be remembered that it actually works in a vertical position.

Another peculiarity of this valve is the steam lap, which can be considered in two ways. For convenience we will call these lap *M* and lap *N*. The former is the distance the outside steam edge of the valve overlaps the steam edge of the cylinder port, while the latter is the amount the inside steam edges overlap the edges of the valve seat and pressure plate. It will be seen that the steam passes these alternate edges simultaneously. That this is an indirect valve and indirect motion is shown by the fact that the eccentric is advanced ahead of the crank and the valve travels in an opposite direction to that of the piston while admitting steam.

Fig. 5 shows the valve open for lead on the top center. The valve is open to the top port for the passage of steam direct from the steam chest and at the same time the bottom member of the valve has cleared the pressure plate and seat enough to admit steam through the tubes to the top port. The direction of steam flow is denoted by arrows, the exhaust flowing as indicated by the arrows on the bottom end.

Fig. 6 shows the valve full open to steam on the top end and exhaust on the bottom. In this position the eccentric has reached the extreme bottom of its travel while the crank is at 27 per cent. of its downward stroke.

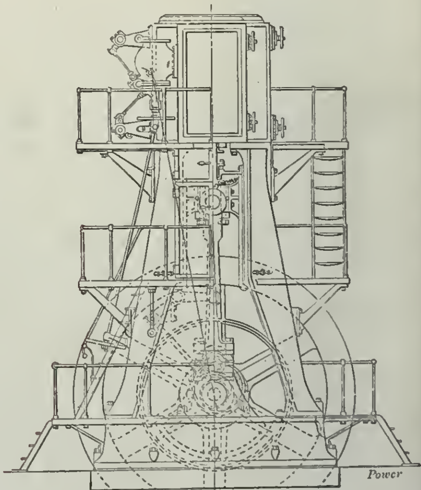


Fig. 2. End Elevation

Fig. 7 shows the valve at the point of cutoff on the top end. The crank has reached about 76 per cent. of its downward stroke and the exhaust on the bottom end is still open.

Fig. 8 shows the valve open for lead on the bottom crank center. It may be noted that on the eccentric circle the exhaust released on the top end at about 91 per cent. of the piston stroke and closed on the bottom for compression at about 93 per cent.

Fig. 9 shows the valve full open to steam on the bottom end and to exhaust on the top with the crank at 22 per cent. of its upward stroke.

Fig. 10 shows the point of cutoff on the bottom end while still exhausting at the top. The crank is at 76 per cent. of its upward stroke. A continuation of the operation will show that exhaust released on the bottom at 94 per cent. of the stroke and closed on top at 96 per cent.

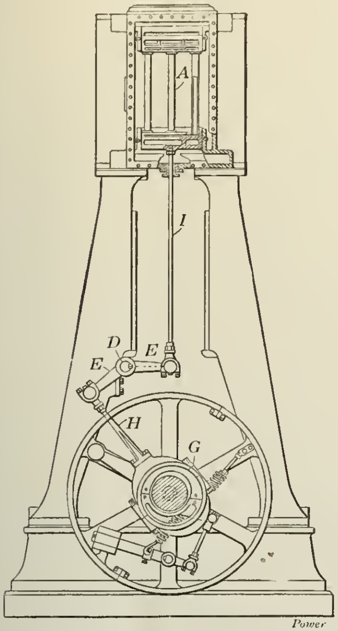


Fig. 3. Showing Governor and High-pressure Valve

Summary of Valve Action

To briefly summarize this valve action it will be noted that the lead is about the same on the bottom and top crank centers with the variation in exhaust action shown in the following table:

Valve Action.	Full Open, Per Cent.	Cutoff, Per Cent.	Exhaust-Release, Per Cent.	Exhaust-closure, Per Cent.
Top end....	27	76	91	96
Bottom end.	22	76	94	93

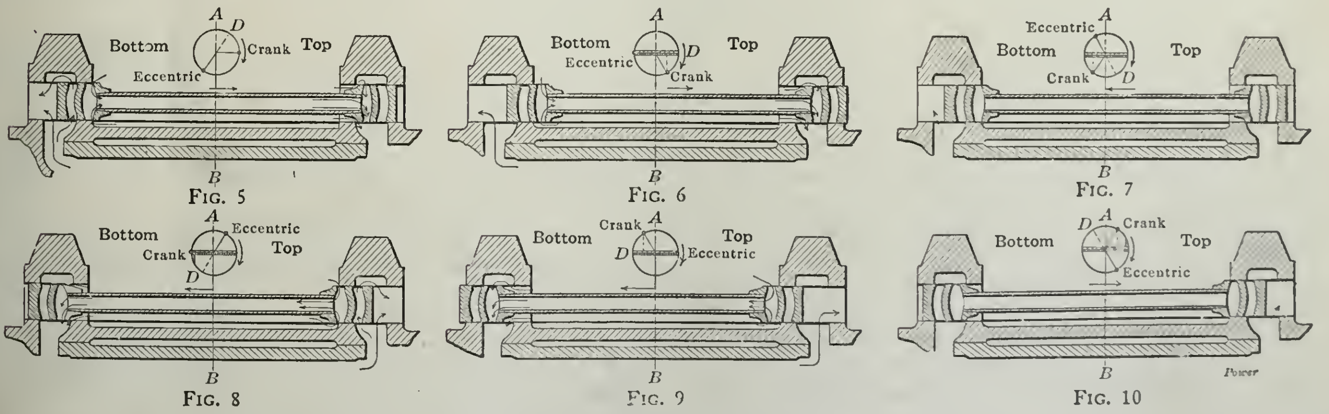


Fig. 5 Fig. 6 Fig. 7

Fig. 8 Fig. 9 Fig. 10

Various Positions of Valve

Owing to the type of valve motion, the valve can have even steam lap and negative exhaust lap on the top end to give the desired steam distribution and balance the inertia of the moving parts.

To Set the Valve

The governor eccentric which operates this valve is fastened to the governor wheel or casing, and the casing, in turn, is fastened to the shaft with a friction key. By slacking on the key set screws, hub and rim bolts the wheel with the eccentric can be revolved around the shaft.

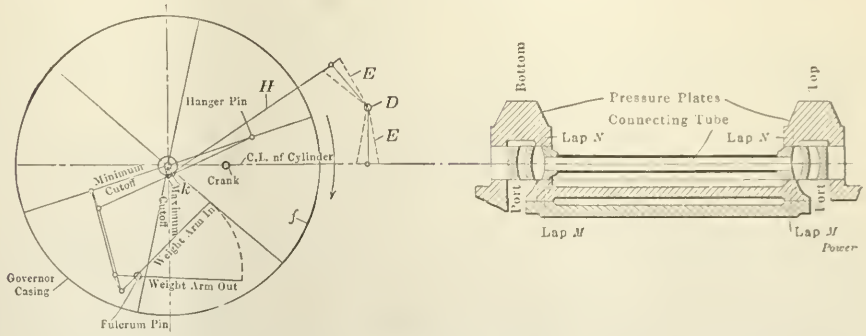


Fig. 4. Mid-position of Valve and Corresponding Position of Eccentric.

It should be borne in mind that the preliminary valve setting should be performed with all valve motion connected and the governor weight arm on its inner stop or in a position to give maximum cutoff. For convenience, remove the steam-chest cover and pressure plates. Then test the valve for lap by turning the governor around until the bottom edge of the valve comes on a line with the lower edge of the bottom cylinder port. Then note if the steam edges of the valve are equidistant from the edges of the valve seats nearest the cylinder center. There will be exhaust opening on the top and the valve should be in the position shown in Fig. 4. Again test the correct length of the valve by moving it up until the bottom edge on the top of the valve is in line with the bottom edge of the top cylinder port, and note if the top edge of the bottom steam port is in line with the top edge of the bottom valve seat.

Next, test the valve for equal travel, up and down. As the bottom exhaust edge of the valve is on a line with the lower edge of the bottom cylinder port when the valve is central, it is plain that this exhaust edge should travel an equal distance above and below this port edge during one revolution of the eccentric. Also, this valve travel should be equal with the rocker arm, to which the valve stem is attached, traveling an equal distance above and below the center line of the rock shaft as shown by the dotted lines in Fig. 4.

Now place the engine on one center and set the valve for lead on that end and then on the opposite end, making the lead the same on both ends. About 1/8-inch lead is correct on sizes of 300 to 800 horsepower.

After setting the valve, secure the governor casing in place by taking up on the hub and rim bolts and finally the key set screws. When the governor casing is in place turn the engine to the top center and note the position of the governor in relation to the crank. It will be noted that the eccentric leads the crank by about 90 degrees. The hanger pin from which the eccentric is suspended will be between the eccentric rod and the crank center nearest the eccentric rod as in Fig. 4.

Remove the governor spring and block the weight against its outer stop and while turning the engine a complete revolution note that the valve does not open for steam admission at any point of the revolution. This can best be observed with the pressure plates in position. No opening should show between the bridges of the valve nearest the steam side and the edge of the pressure plates. After this point has been reached in the process of valve setting, everything should be marked so that any slippage or changes can be noted.

GOVERNOR ADJUSTMENT

The Williams governor is centrifugal in type and is of the class that throws the eccentric across the shaft center, varying the length of valve travel. The governor has two weight arms, one on each side of the spoke in which the pivot pin has a bearing. The force of the weights is opposed by two springs, one to each weight arm. The springs are fastened to the weight arm with a movable clip so that more or less purchase can be given the springs on the weight arm. The weights can also be moved along the weight arms to give more or less centrifugal effect. To move the weights in toward the pivot pin has the effect of less weight; to move the weights out has the effect of more weight. To move the spring clips on the weight arm in toward the pivot has the effect of more weight; to move the spring clips out has the effect of less weight.

To obtain more speed on the engine, give more tension on the springs until the racing point is reached, then to get more speed take away the weight, and to decrease the speed, add weight. Changes on the springs affect the speed and sensitiveness. Changes in weights affect the speed principally. Make all changes alike on springs and weights.

A Centrifugal Pump Experience

By W. N. Wing

In a certain plant two centrifugal pumps refused to pick up the water. Every expedient was tried to make them work, but the pumps would not throw a drop until they took the notion, and then of their own accord.

Some time ago I was sent to a small plant a few miles from the large city where I was located. It consisted of two centrifugal pumps, direct-connected to two vertical engines which were run at about 300 revolutions per minute. The size of the pumps was 10-inch discharge and 12-inch suction. Each pump was connected by a separate suction into the main suction pipe of 16 inches in diameter, which extended both ways from the engines, about 300 feet each way. From this main, short branches about 30 feet apart were connected to driven wells, about 20 in all. The water level in the wells, was, according to test wells situated at different points on the line, about 14 feet from the surface. The plant was shut down and it was stated that the pumps would not pick up the water.

I describe the layout of the plant to show the arrangement of the piping. The two pumps were side by side, about 12 feet apart, with the back sides of the pumps, which were connected with side-entrance suctions, next to the wall. The discharge pipe extended vertically from the pump about 9 feet and then turned with a long-sweep elbow and extended through the wall and into a wooden trough about 15 feet away. where it discharged. On the end of each discharge pipe was a leather-seated clapper valve. Out of the top of each pump was a 4-inch pipe with a valve, and these pipes had a rise of about 2 feet and then turned and came together with a tee in the center, looking up. From this tee another riser was put in, about 4 feet long, and at the top of this was another tee, one end of which was connected to a 2- or 2½-inch siphon and the other to a discharge pipe for the siphon. The method of operation was to close the suction valve of one pump and then open the siphon, which would close the clapper valve and remove the air from the suction line, thus forming a vacuum and causing the water to rise in the pump. It should be said that only one pump was operated at a time.

The engine was started and put up to speed and the trick was supposed to be done, but, on the contrary, the pump refused to work. Then came the hunt for trouble. First we had a tryout of every joint in the engine room, and the clapper valve, by closing the suction valve and starting the siphon, and everything proved to be tight after a close examination. Nothing remained then but the main suction line and the branch lines to the wells, and these, after starting the siphon, were carefully examined. A small leak was discovered in one of the joints, and immediately made tight, but no cause was found to prevent the pump from working. We tried again but the pump absolutely refused to throw water. The engineer made a few extemporaneous and (as I thought) well chosen remarks to the pumps and we tried again. This time we started the other pump, but it wouldn't throw water, so we tried first one and then the other, and then both together, with the siphon on both, and while the siphon threw a 4-inch solid stream of water, those blessed pumps wouldn't throw a drop.

It was discouraging and I was about ready to give it up, as we had examined the impeller in the pump as about the last resort, when it occurred to me that some one of those wells might be dry or the pipes broken. The main suction had a valve between the two pump suctions and this was closed and the siphon put on the one side, and then each well was tried separately on both sides of the pipe line. Another hope was gone, as they all proved to be all right. In the meantime all these trials and suggestions had taken a few days. The water level in the wells had risen about a foot.

One morning I decided to "try her out" once more, so the engine was started after the siphon had picked the water up, and we stood around to see what the result would be, but there was nothing doing. Then, not knowing what to do, I opened the valve to the siphon on the pump that was standing, and closed it on the one running, when to my surprise the engine seemed to be getting the water, and upon opening the petcock on top of the pump the water squirted about two feet into the air, and yet no water was discharged at the end of the pipe. Now then, said I, we have it. The pump wants more speed. So we shut down and moved the regulator weights (it was a flywheel governor) for more speed, and started again and she would not throw a drop of water. Then we put her back to her regular speed, and tried again, and opened the siphon first on the running pump and then on the idle one, then both together, and then one and then the other again, and I had just opened it on the idle pump and closed it on the one running when the engineer called me to the window. While we were talking there came a thud from the engine. We turned and she was "on the job," throwing a solid 10-inch stream of water. We just looked at each other, and the engineer exclaimed: "Well, I'll be ———." He expressed my feelings to a nicety. Well, we let her go at that, concluding that she had decided to go to work.

This happened a few years ago, and I have been told that they always have the same trouble in starting those pumps. They are never sure whether they can start them, and engineers have been there who never could start them. It has always been an unanswered question in my mind what the trouble really was, and I have mentioned it to a great many engineers, all of whom had suggestions to make, and yet none was ever made that we didn't try; so it has never been as I said at first, settled to my satisfaction. It was a case where a prescription from a P. D. (pump doctor) that would effect a cure would be worth several silver dollars.

Blowoff Cocks for Steam Boilers

By Charles J. Simeon

> *Originally the plain plug cock was used for blowoff purposes, but it proved unsatisfactory for such service. The developments that have been made in improving its design are illustrated herewith.*

When the first boiler was made some device was needed to allow the blowing out of mud and loose scale left by the evaporation of the water. In the beginning, steam pressures were low, and steam and water fittings as they are known today, were primitive and clumsy or nonexistent. Since that time the demands for high pressures and the better understanding of the requirements have directed a great deal of inventive ability toward the design of special appliances suited to the conditions. These have to handle not only steam and water, but loose scale, mud and such other matter as enters the boiler with the feed water, including boiler compounds and homemade scale solvents.

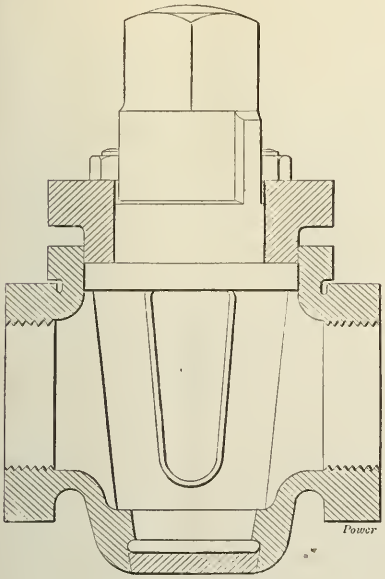


FIG. 1. FAIRBANKS

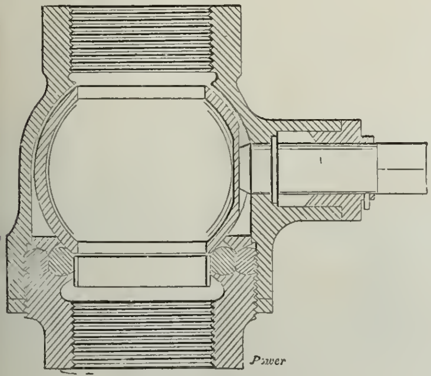


FIG. 2. HUXLEY

No feed water is pure, and all impurities not deposited and adhering to the shell and tubes must pass out through the blowoff in a more or less concentrated form. The solid portions made up of grit and pieces of scale are liable to adhere to the working faces of the cock and to choke or clog restricted passages. On account of their usually isolated and somewhat inaccessible location the blowoff is often more neglected than any other detail in the steam-power plant, despite the fact that, owing to the severe nature of the work done, it is more apt to need attention than any other appliance.

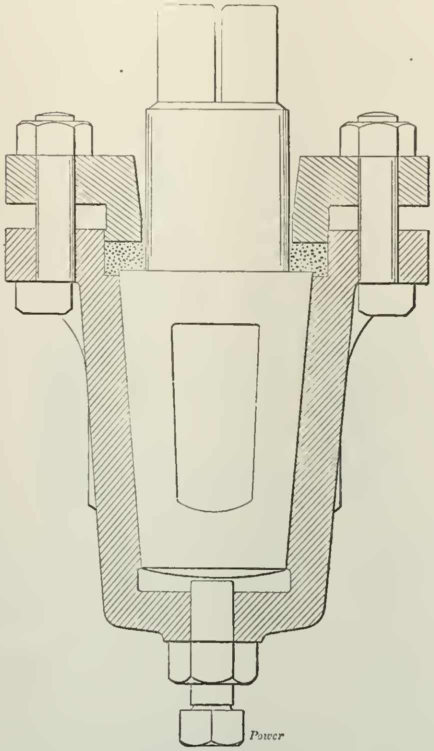


FIG. 3. WICKES

Most men will handle steam apparatus with care because they have a healthy respect for live steam under pressure, but many will be careless in operating water valves and cocks. If one sticks the first impulse is to resort to the use of a hammer or a long wrench, with the possibility of personal injury or damage to the fitting, if this impulse is not checked. It is axiomatic that violence should never be used in operating any steam or water apparatus, but it is especially true of the boiler and all its connections. Too rapid opening or closing of the blowoff may be a fruitful source of trouble. A water-hammer action may be set up which will strain or break the pipe or some of the fittings or a bit of hard scale may be caught in such a way as to cause cutting and subsequent leaking, while if care is exercised, the obstruction may possibly be felt and coaxed out by patient manipulation.

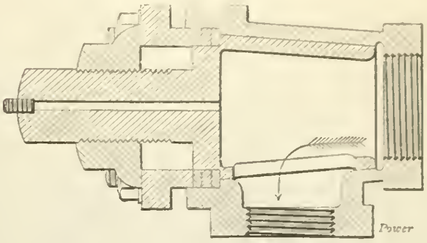


FIG. 4. SHAW

Some appliances, excellent for other service, fail when used in the blowoff because the design or the material used is unsuitable. A valve or cock that is easily choked or the working faces of which are exposed to the cutting action of the flow is wrong in design, and one which may be affected by the impurities in the water or is not tough enough to stand the cutting action of the stream of dirty water passing through it, is made of the wrong material. In many designs the aim to overcome these difficulties is shown in ample clearances, unobstructed passages, massive proportions and in the use of the strongest and toughest material available.

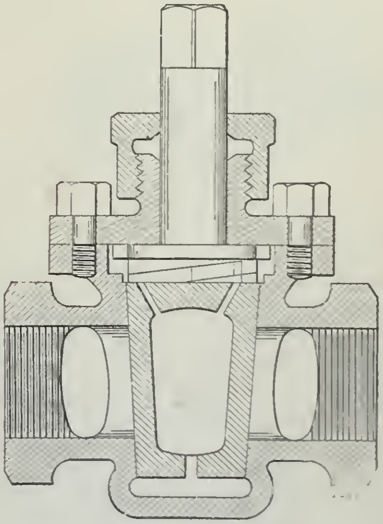


FIG. 5. HOMESTEAD

The first appliance used for the blowoff was the ordinary brass plug cock, because it was the only thing available. It is still used by many engineers on account of its simplicity and cheapness, but by others it is considered unsuitable because mud and scale come to rest against and adhere to the plug, and when it is turned they are carried with it, cutting the faces of both plug and body, causing it to stick and leak. Also when it is closed, the sides of the plug and the body nearest to the boiler are exposed to a higher temperature, which causes an unequal expansion, warps both members and makes the plug stick in the socket.

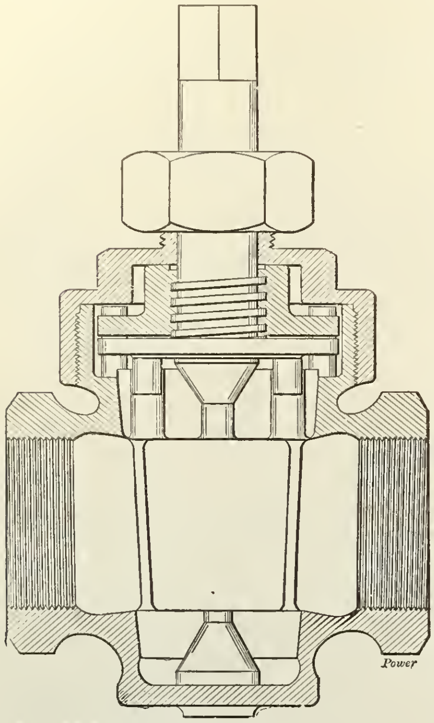


FIG. 6. CADMAN

Developments of the plug cock have been numerous and along different lines. The first variation took the form of an iron plug in a brass body which was less liable to stick from uneven expansion than one made entirely from one metal. This, however, was not a great success, owing to the rusting of the iron plug and the high cost of the brass body. One made with an iron body and a brass plug though cheaper, was impracticable for the very reasons that made the other a partial success. Then one was tried in which both the body and the plug were of iron, given an anti-rust treatment which reduced the tendency to stick and cut. This form is still used to a large extent.

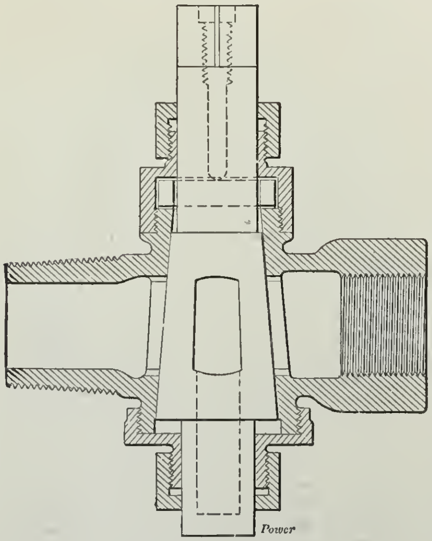


FIG. 7. ANDERSON

The next step was, in addition to the anti-rust treatment, to make the seat of some material more pliable than iron or brass. The Fairbanks asbestos-packed cock, Fig. 1, is an example of this type in which a barffed-iron plug works in a socket of specially prepared asbestos forced into position by hydraulic pressure. It has a vulcabeston ring on the shoulder of the plug for a top packing.

The Huxley, Fig. 2, is somewhat similar in character. It is a plug cock in which the plug takes the form of a hollow sphere while the seat is formed by a layer of soft packing that is pressed into intimate contact with the spherical surface of the plug by means of a screwed gland.

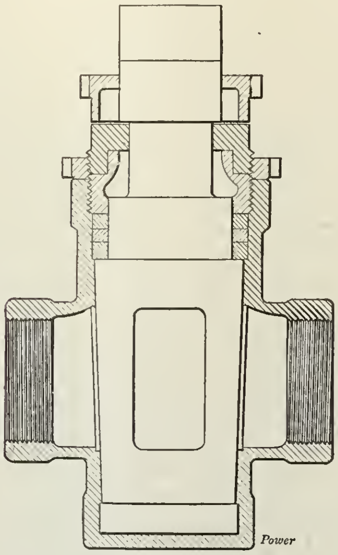


FIG. 8. BORDO

The next successful cocks are of the type in which a mechanical device is used for loosening the plug in its socket, thus allowing it to move with ease and yet be tight when closed. The Wickes, Fig. 3; Shaw, Fig. 4; Homestead, Fig. 5; Cadman "Split-plug," Fig. 6; Anderson, Fig. 7; Bordo, Fig. 8; the Wiltbonco, Fig. 9, and the Ashton, Fig. 10, are examples of this type.

The Wickes is provided with a set screw and lock nut which permits adjustment of the plug to any degree of tightness desired or readily loosening it if it should stick. The Shaw has a large nut working on the stem of the plug for the same purpose. In these two the adjustment of the plug must be deliberately made.

The Anderson, Homestead, Bordo and Cadman also provide for adjusting the tightness of the plug in its socket, but the loosening or tightening is done automatically in the act of operating. In the Anderson this is effected by means of two rollers that revolve on a shaft passing through the stem of the plug and run up inclined planes cut in the body. A movement in one direction loosens the plug in its seat and locks it open, while a reverse movement tightens it and locks it closed. The Homestead attains the same results by means of a spiral cam. The Shaw is an example of a cock in which an attempt is made to keep the gritty sediment contained in the water from resting on the working surfaces when closed. To attain this the inlet is made at the small end of the plug and the outlet at one side, pointing downward. When so placed, any sediment from the boiler collects inside and is blown out when opened, instead of coming to rest against the ground surface of the plug.

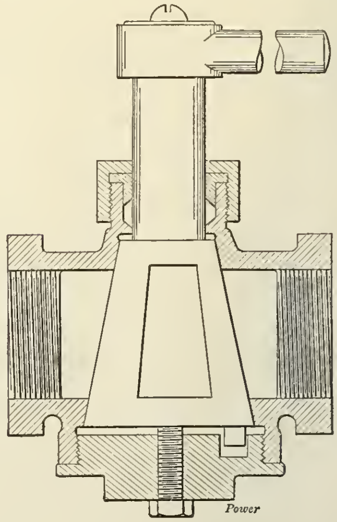


FIG. 9. WILTBONCO

The above steps trace the development of the plug cock, which was the first piece of apparatus used for this purpose and is by many engineers still preferred to any other, owing to its simplicity and cheapness.

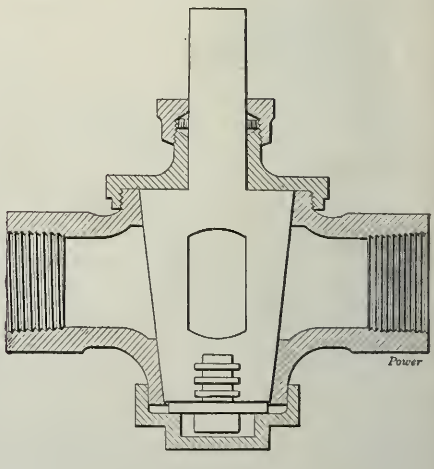


FIG. 10. ASHTON

Operation of Compression System

By F. E. Matthews

The cycle of operation is thoroughly explained, beginning with the introduction of the ammonia into the expansion coils, following it through the compressor up to the condenser and its return by way of the liquid receiver to the expansion coils.

While it is impossible to show in a single illustration all the details entering into the mechanical construction of the complete refrigerating system, the diagrammatic representation presented is more complete than the elementary diagrams already shown, and will suffice for the explanation of the cycle of operation of the compression system.

The cycle begins with liquid anhydrous ammonia conducted to the expansion coils through a liquid line and regulated by appropriate expansion valves. The source of the liquid ammonia may be either a liquid receiver, which, as already shown, is one of the essential members of the system in practical operation; or it may be an ammonia-shipping drum from which ammonia is introduced into the system by the process of charging. In either case the first function of the refrigerant is to enter the expansion coils in the compartment to be refrigerated and there to evaporate (boiling at a temperature dependent on the suction or "back pressure" within the coils) and absorb heat from the surrounding objects. If the back pressure be 46 pounds gage, or less, the temperature of the boiling liquid will be 32 degrees Fahrenheit, the freezing point of water, or below, and the pipes containing the refrigerating medium at these temperatures will soon be covered with a coating of frost. If, on the other hand, the back pressure in the coils is above 46 pounds gage, the temperature of the boiling liquid will be above 32 degrees Fahrenheit, and no frost will be formed. Refrigeration will be produced whenever the temperature of the expansion coils is lower than that of the air in the cooler, regardless of whether frost is formed or not and, according to the amount of atmospheric humidity, uncongealed moisture may or may not be precipitated on the pipes.

Having vaporized in the expansion coils, the ammonia vapor enters the return header which conveys it back to the suction side of the compressor. This return line is usually fitted with a "scale trap" constructed similarly to a simple steam separator. The function of this trap is to prevent any scale from the inside of the pipes, or other foreign substance, from entering and damaging the compressor. From the scale trap the gas passes through the suction valves into the compressor.

On leaving the compressor the hot, high-pressure ammonia gas passes through the discharge valves and out into the two legs of the main discharge pipe, which come together in a "T" just below the large hand-operated discharge valve at the left-hand side of the compressor cylinder. Leaving the discharge valve, the gas passes into the side of the pressure-tank head, and is given a spiral motion as it descends into the tank. The centrifugal force produced by this spiral motion is intended to aid in precipitating any entrained oil which the gas may hold in suspension, against the outside of the tank. The gas then passes up through the hot gas line, through a "check valve" (sometimes omitted) located a little above the level of the top of the condenser, and down into the header connecting the bottom pipes of the several stands of condensers. The loop is placed in the hot-gas line to prevent condensing liquid from running back down into the pressure tank when the compressor is shut down.

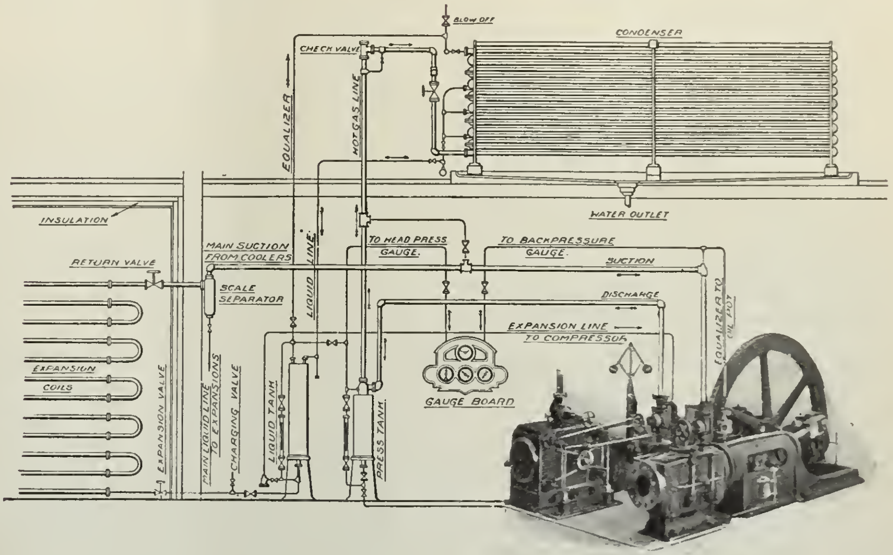


Diagram of Complete Ammonia Refrigerating System

The gas entering the bottom pipe of the condenser passes up through successive pipes while the water distributed over the top pipe trickles down, producing the desirable countercurrent cooling effect, in which the hottest water encounters the hottest gas at the bottom and the coolest water the coolest gas at the top of the condenser.

As fast as the ammonia is condensed in the pipes of the condenser it is conducted away through smaller liquid-drip pipes connected with a common liquid header. The outlet from this header also rises to form a short "goose-neck" which is intended to keep the header always full of liquid and prevents gas from being drawn down to the tank through the liquid line. In trying to ascend through the column of descending liquid in a small liquid line, bubbles of gas offer no inconsiderable resistance to the passage of the liquid. The obvious remedy for this difficulty is the installation of lines of liberal diameter.

If the gas is carried down into the liquid tank it can escape by going up the equalizer line into the top of the condenser. The ammonia previously liquefied in the condenser under a pressure of from 135 to 200 pounds, according to the temperature of the cooling water, is conveyed first to the receiver. This consists of an appropriate cylindrical containing vessel which acts as a storage tank in which the liquefied ammonia is collected and from which it passes as required into the expansion coils. The flow of this high-pressure liquid into the expansion coils is regulated by expansion valves, which are virtually nothing more than convenient mechanical devices for accurately varying the opening through which the liquid ammonia must pass on its way to the expansion coils. As stated, the word "expansion" has been erroneously applied to these valves and coils, because of the idea, also erroneous, that the liquid ammonia vaporizes or expands immediately when the pressure is relieved, as it passes the regulating valve and enters the cooling coils. As a matter of fact, before it is possible for a pound of ammonia to change from the liquid to the gaseous state it must be supplied with about 555.5 B.t.u. of heat.

In practice, not all of the heat-absorbing capacity, or negative heat, of a pound of anhydrous ammonia available at zero Fahrenheit can be utilized for useful cooling work, on account of the cooling work which must first be expended on the ammonia itself in order to reduce its temperature from that of the condenser to that of the cooler. This may be illustrated by a similar process in which water is the medium. The amount of heat that must be abstracted from one pound of water at 32 degrees Fahrenheit in order to freeze it, is 144 B.t.u. On this basis, a ton of ice would represent 288,000 B.t.u. of negative heat. In practice, the expenditure of this amount of cooling will not freeze a ton of water, because it must first be reduced from its natural temperature, or, in crystal-ice systems, from the temperature of the distilling tank to 32 degrees Fahrenheit. This involves a further expenditure of one negative B.t.u. per pound per degree of temperature.

If the 555.5 B.t.u. were absorbed at the expansion valve, which its immediate vaporization assumes, there would be no further heat-absorbing capacity in the ammonia, and its introduction into the expansion coils would be useless.

Besides the principal pipe circuit just described, the compressor is provided with a set of bypass connections for reversing its operation so as to draw the ammonia from the condenser and discharge it into the expansion coils, as well as other so called "pump-out" lines through which ammonia may be pumped out of other parts of the system in case it becomes necessary, as when making repairs.

In the figure is shown a small liquid line running from the liquid tank to the compressor cylinder. This line is provided with an expansion valve through which ammonia may be admitted to the compressor, to prevent abnormal heating of the piston and packing when starting up, or at any other time when the ammonia returning to the compressor is not sufficiently cold to insure satisfactory operation of the compressor. Another small pipe line connects the lubricating system on the compressor with the pressure tank. Through this line oil passing over with the discharged ammonia gas and separated out in the pressure tank, may be blown back into the lubricating system. On entering this line the oil passes first through a small strainer which intercepts any scale or foreign substances that might otherwise return to the compressor.

Details of construction of the various parts of a compression system are too numerous to warrant an attempt to fully describe them here. Since the ammonia compressor is so important a member of the refrigerating system, a subsequent article will be devoted to defining and illustrating the two principal types, namely, the horizontal double-acting and the vertical single-acting compressor.

An Improved Hatchet Planimeter

By Frank S. Bunker

The general theory and operation of planimeters are explained and a description is given of an improved form of hatchet planimeter whereby the plane of the tracing arm is kept constantly at right angles to the surface of the area to be measured.

Among the instruments used by steam engineers the planimeter is second in importance only to the steam-engine indicator. In the offices of designers, architects, civil engineers, the planimeter has a place as honored and as essential as the oil can has in an engine room. By its use the designer is able to quickly obtain the cross-sectional area of any irregular object, and thus obtain the contents and weight without involving tedious calculation. The civil engineer uses it on his maps and obtains areas which would require a long time to calculate.

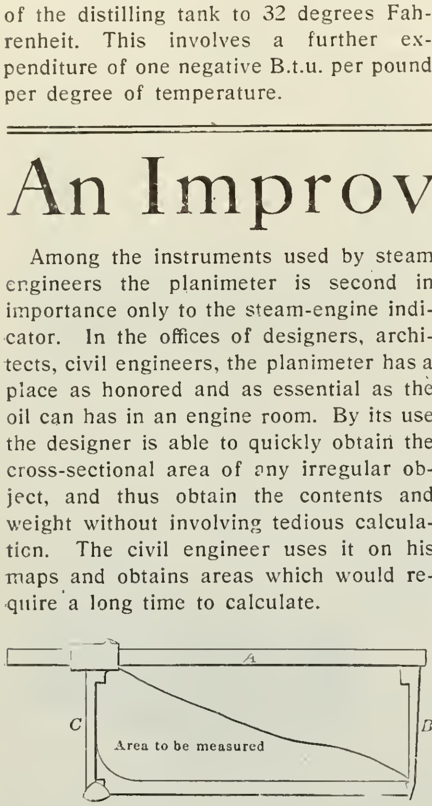


FIG. 1. SIMPLE FORM OF HATCHET PLANIMETER

The designs of planimeters are almost as varied as their uses. Some styles are better for certain fields of application than others, but the underlying principles of all types of planimeter are the same and one general theorem applies to all.

THE HATCHET PLANIMETER

One of the simplest forms is the hatchet planimeter, often called the Stang-Prytz-Goodman planimeter. This is shown in Fig. 1 and consists of a bar *A* at the extremity of which is a rigid arm *B*, the latter being the tracer. Sliding along the bar *A* is another arm *C*, the lower edge of which is shaped like a hatchet, the edge corresponding to the cutting edge being in perfect alinement with the tracer *B*.

The arm *C* is adjustable along the main bar and is clamped in place by a spring. When in use, the adjustable arm is shifted along the bar to a position at which the two arms just encompass the length of the figure to be measured. The center of gravity of the area to be measured is roughly ascertained and a line is drawn from it to the edge of the figure, preferably parallel to its longest dimension or parallel to the atmospheric line in an indicator diagram; see dotted line *E E'*, Fig. 2. With the tracer at *D* place the instrument approximately at right angles to the line *DE*. Press the hatchet end into the paper so as to make a depression at the starting point *F*. Now move the tracing point along the dotted line *DE* and trace around the area, returning to the starting point *D*. It is preferable always to trace diagrams in a clockwise direction at the first operation. Now press the hatchet again as at *G*. If the point *D* were the exact center of gravity then the distance between *F* and *G* would be the mean hight of the diagram, but the point *D* was selected according to the operator's judgment and is therefore only approximate.

Still holding the tracer at *D* turn the diagram through approximately 180 degrees and retrace the area in the opposite direction (counter clockwise) and when in the final position press the hatchet as at *H*. A point *J*, midway between *H* and *G*, is now found, and its distance from *F* is the average or mean hight of the figure. If the figure be an indicator diagram, this mean hight multiplied by the scale of the spring will give the mean effective pressure.

The distortion of the correct positions of the points *H* and *G*, due to not holding the plane of the instrument at 90 degrees with the surface of the area being measured, led me to design and construct the instrument shown in Fig. 3.

This instrument carries an arm *K*, at the extremity of which is a small roller *L*, which is free to swivel in any direction and prevents the instrument from slipping sideways. I also substituted a wheel *M* having a knife edge in place of the hatchet end, and attached a small pointer *N;* this when pressed gives the

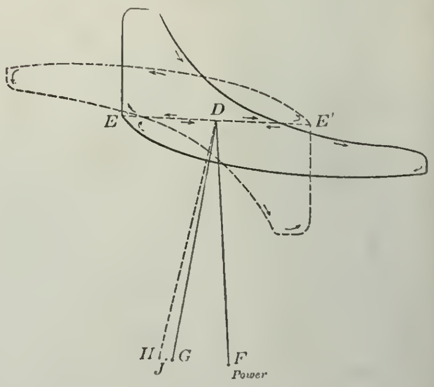


FIG. 2. OPERATION OF HATCHET PLANIMETER

required location of the starting and finishing positions *F*, *G* and *H*, Fig. 2. The small disks shown holding the card and on the instrument are used as weights to add to the stability of the instrument when in operation.

I have tested this planimeter with known areas and have found it to be as accurate as any of the more complicated and expensive planimeters.

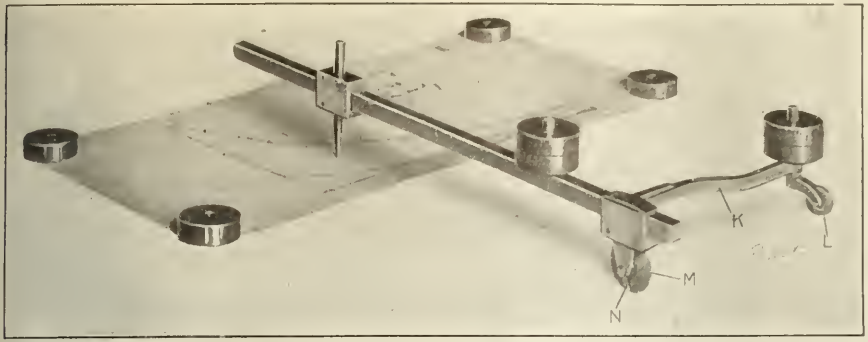


Fig. 3. Improved Hatchet Planimeter

Theory of the Planimeter

To Professor Henrici we are indebted for a discussion of the general theory upon which all planimeters are based, namely, that the net area swept over by the tracing arm equals the difference in the areas traced by the extremities of the arm. Hence in Fig. 4 the tracing point moves over the outline of the closed curve *O P Q R*, while the opposite end of the tracing arm moves over the outline of the lower closed figure *O′ P′ Q′ R′*. Let the starting position of the tracing arm be *Q Q′* and let the tracing point move around the outline *Q P O* from right to left; then the other end of the arm will follow the outline of *Q′ P′ O′*. Continue until the tracing point comes back to the starting point *Q* and the follower will be at its initial position. The arm has now swept over the entire area *O P Q Q′ R′ O′*. The area *O R Q Q′ P′ O′* having been twice swept over by the tracing arm, once in a counter clockwise direction and again in the opposite direction, will cancel and there is left the area *O P Q R*, swept over in a positive direction, and *Q′ P′ O′ R′* swept over in a negative direction. Hence the net area swept by the tracing arm equals the difference between the areas swept by the ends of the arm, that is, *O P Q R Q′ P′ O′ R′*.

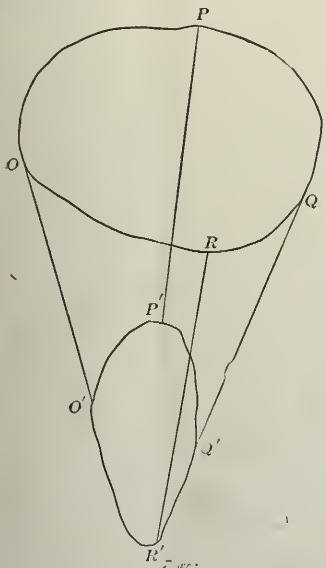


Fig. 4. Area Passed Over by Tracing Arm

With the Coffin and the Amsler planimeters the follower passes along a line without inclosing an area; therefore the difference between the areas swept over by the extremities of the tracing arm is the area of the figure traced. Following the same reasoning with the hatchet planimeter, the hatchet either goes over the same line in two different directions or sweeps the area partly in a positive direction and partly in a negative direction, and the net area swept over by the arm is the area of the figure traced.

The motion of the arm is one of translation along the arm and of rotation about the point of contact of the hatchet. As the former does not sweep over an area, the whole area moved over by the arm is therefore due to angular motion. In other words, a line the same length as the tracing arm, rotated about one of its extremities as a fixed point and always kept parallel to the tracing arm, will sweep through an area equal to that swept through by the tracing arm. Hence the area swept through by the tracing point is equal to the length of the tracing arm multiplied by one-half the length of the arc swept through by the tracing point.

Now proceed with a closed figure, as *S T U*, in Fig. 5. Let *U V* be the initial position of the tracing arm, *U* being the tracer and *V* the follower or hatchet. Trace the curve from *U* to *T* and *S* around to *U*. The follower will have traced the path outlined by *V W X*. Now draw *V U′* parallel to the final position *X U* of the tracing arm. The area swept by the arm is equal to the area of *U′ V U* or one-half the length of the arm multiplied by the circular measure of the angle *U′ V U*, which is the same as angle *V U X*. But the tracing arm has been set to the exact length of the figure to be measured; hence arc *V X* must equal the average or mean hight of the area.

To make sure that the area of the figure traced by the hatchet is zero, the point *W′* may be selected as the center of the area, and through *U* draw *U W′*. If when the tracing arm was in its first position *U V*, the tracing point had been moved from *U* to *W′* the hatchet would have followed the line *V Y* to *Y*, and the same may be said of the final position of the tracing arm *U X;* move the tracing point and the hatchet will follow the curve *X Z*. Now place the tracing point at *W′* and the hatchet at *Y* and move the former along from *W′* to *U*, then the hatchet will move from *Y* to *V*. Trace the figure as before, ending at *U* with the final position of the arm at *UX*. Move *U* along *U W′* and the hatchet will follow to *Z*. Swing the hatchet through the arc *Z Y* and if the center *W′* has been properly selected the area of the figure traced by the hatchet is zero, being generated half positively and half negatively. When this is so the net area whose outline was traced by the tracing point will be equal to the area swept over by the tracing arm, which is equal to *W′ Y* × *Y Z*. If *W′ Y* equals the length of the figure then *Y Z* must equal the mean hight.

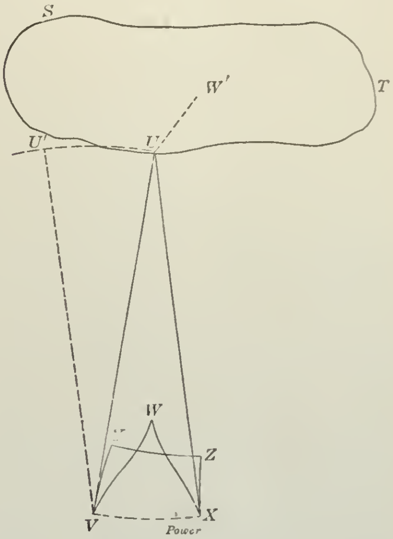


Fig. 5. Diagram Illustrating Proof of the Theory upon Which the Planimeter Is Based

To reduce the error due to a faulty location of the center of gravity after having once traced the area to be measured, turn it through 180 degrees and retrace in an opposite direction; then take the average between the two points as described in connection with Fig. 2.

Provided that care is taken in obtaining perfect alinement of the hatchet and the tracing point, the results are as satisfactory as those obtained by the most expensive planimeter.

The Right Angled Triangular Weir

By Charles N. Cross

> *This type of weir is better adapted to the measurement of small and variable quantities of water than the rectangular weir, and besides the quantity of water flowing becomes the function of only one variable—the head of water.*

The word weir was first used in an engineering sense to designate an artificial obstruction placed in the bed of the stream. The weir was used to raise the level of the stream for purposes of irrigation, navigation or for power. Later, when it became the custom to specify the amount of water to be used at certain places, its measurement in large quantities was necessary. A rectangular hole was cut in the wall and the amount of water roughly gaged by the size of the opening. These rectangular notches were calibrated in various ways as the specifications became more exacting, and the openings were called weir notches and later simply weirs.

In 1858 James Thomson, professor of civil engineering, Queen's College, Belfast, suggested the use of the right-angled triangular notch with its apex pointing downward, to take the place of rectangular weirs, because the latter were not adapted to the measurement of small and variable quantities of water. Another advantage of the triangular notch given by Thomson is, that the quantity of water flowing becomes a function of only one variable, viz., the head of water; while in the rectangular notch it is a function of at least two quantities: the head and the horizontal width. Of course, the application of the triangular weir at the present time is rather limited. For some kinds of laboratory testing and for measuring small flows in irrigation work, it is very convenient.

TABLE 1. CONSTANTS FOR WEIR DISCHARGE.

$q = c\,h^{\frac{5}{2}}$	$Q = C\,H^{\frac{5}{2}}$	$Q = \frac{8}{15}\,K\,H^{\frac{5}{2}}\sqrt{2g}$
$c = 0.317$	$C = 2.635$	$K = 0.6162$
$c = 0.306$	$C = 2.544$	$K = 0.5948$

Thomson reasoned that the flow must be proportional to the 5/2 power of the head, that is, the discharge is proportional to the product of the velocity of the water times its cross-sectional area. In Fig. 1 let

$h =$ Head of water,

$b =$ Width of notch at level of the water,

then the area of the notch filled is ½ $b\,h$; but in a right-angled triangle

$$b = 2h,$$

therefore,

$$\tfrac{1}{2}\,b\,h = h^2.$$

Theoretically the velocity of a body falling through a vertical distance h is,

$$\sqrt{2\,gh},$$

where g is the acceleration due to gravity and is about 32.16 at a latitude of 45 degrees. Therefore the theoretical discharge is,

$$q = h^2\sqrt{2\,g\,h} = h^{\frac{5}{2}}\sqrt{2\,g}.$$

Thus the discharge is proportional to $h^{\frac{5}{2}}$. This reasoning of Thomson's was substantiated by more rigorous mathematical development later.

Now it remained to determine by experiment whether there was some constant which multiplied by $h^{\frac{5}{2}}$ would give the quantity of water discharged for all values of h, that is,

$$q = c\,h^{\frac{5}{2}}.$$

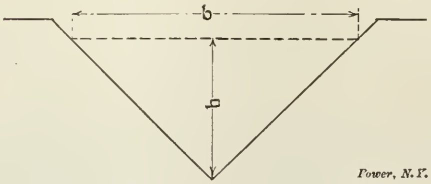


FIG. 1. TRIANGULAR NOTCHED WEIR

After a limited number of trials and with h varying from 2 to 4 inches, the following expression was arrived at:

$$q = 0.317\,h^{\frac{5}{2}},$$

where

$q =$ Cubic feet discharged per minute and

$h =$ Head measured vertically in inches from the still-water level of the pool down to the vertex of the notch.

This is the formula which Thomson submitted to the British Association for the Advancement of Science in 1858. In his report he states that it is only a temporary formula subject to amendment depending upon more perfect experiments covering a wider range of heads. In 1861 Thomson presented his final report before the association, and it is published in the records of the society for that year. This report was the result of experiments extending over a period of two years and with heads of from 2 to 7 inches. The average of all these determinations for the coefficient was 0.3064. The formula was thus made to read,

$$q = 0.306h^{\frac{5}{2}}.$$

Thomson himself gave it as,

$$q = 0.305\,h^{\frac{5}{2}},$$

because he gave somewhat more weight to certain experiments. But 0.306 would seem to be the logical value.

It is the present custom to measure all heads of water in feet, and weir discharges are usually measured in cubic feet per second, thus

$$Q = C\,H^{\frac{5}{2}}.$$

The coefficient reduced to give the result in these units will change the equation to,

$$Q = 2.544\,H^{\frac{5}{2}}.$$

Another form of this same equation is often used. It comes from a different theoretical treatment, and is as follows:

$$Q = \tfrac{8}{15}\,KH^{\frac{5}{2}}\sqrt{2\,g},$$

where K is the experimental coefficient; the value of K corresponding to 0.306 above is 0.5948. For convenience of comparison a table of these coefficients is given.

TABLE 2. WEIR CONSTANTS GIVEN BY VARIOUS AUTHORITIES.

Name of Author.	Title of Book.	Coefficient by Author.	Coefficient Corresponding in Thomson's Formula.
W. J. M. Rankine.	Manual of Steam Engine, 3d edition, 1866, page 95.	2.645	0.3181
W. J. M. Rankine.	Manual of Steam Engine, 8th edition, 1875, page 95. Revised by E. F. Bamber.	2.54	0.3055
Mansfield Merriman	Treatise on Hydraulics, 1889, page 127.	2.54	0.3055
S. Dunkerly.	Hydraulics, Vol. I, 1907, page 20.	0.62	0.319
J. A. Ewing.	Steam Engine and Other Heat Engines, 1902, page 211.	2.54	0.3055
Chas. H. Innes	Centrifugal Pumps and Turbines, 4th edition, page 6.	0.595	0.3061
John Goodman	Mechanics Applied to Engineering, 1904, page 561.	0.60 to 0.61	0.3087 to 0.3138
Henry T. Bovey	Hydraulics, 1895 edition, page 56.	0.617	0.3174
R. G. Blaine	Hydraulic Machinery, 1897 edition, page 76.	0.317	0.317
Irving P. Church	Mechanics of Engineering, 1893 edition, page 676.	0.595	0.3061
L. M. Hoskins.	Text Book on Hydraulics, 1907 edition, page 153.	0.592 or 2.54	0.3045 or 0.3055
Herbert M. Wilson.	Irrigating Engineering, 1907 edition, page 86.	2.54	0.3055

All of the values given in Table 1 and some besides have been used by different authors in referring to Thomson's result, as may be noted in Table 2 which contains a partial list of those who have mentioned this type of weir. An inspection of this list, and it was selected without any particular discrimination, shows that nearly one-half of the authors who quote Thomson either directly or indirectly use coefficients in the neighborhood of the value determined in his preliminary experiments, while the other half use values in the neighborhood of his final result.

It is remarkable that a constant derived in a set of experiments by one man can have so many variations. However, there may be some reason for these seeming discrepancies. Thomson arrived at the conclusion that the coefficient must be very nearly a fixed constant because the form of the cross-section of the water discharging over the weir for any head is similar to that for all other heads, that is,

$$\frac{b}{h} = constant,$$

while this is not true for rectangular notches. When it is observed that the flow of water over a weir is the result of the action of gravity modified by various secondary causes, such as velocity of approach, resistance of the air, temperature, adhesion of the water to the edge of the weir and other influences, it is hardly to be expected that a single constant will suffice for all the possible conditions.

That there should be two values given widely differing from each other can probably be explained by the fact that Professor Rankine, who is largely quoted as an authority, was the first to refer to Thomson's work. It was in the first edition of his "Manual of the Steam Engine," which was published in 1859, that he mentions Thomson's formula. This was before Thomson had completed his work. Rankine continued to give 2.645 as the value of the constant in at least three editions of his book, or till 1866. For lack of the intervening editions the writer has been unable to determine just when the change was made, but at least in the eighth edition published in 1875 the proper value is given. This was after Rankine's death, which occurred in 1872.

TABLE 3. WATER DISCHARGED THROUGH RIGHT-ANGLED WEIR NOTCH.

Head in Feet.	Discharge.		
	Cubic Feet per Second.	Gallons per Hour.	Pounds per Hour.
0.05	0.001422	38.29	319.3
0.10	0.008047	216.67	1,806.8
0.15	0.02217	596.95	4,978
0.20	0.04551	1,225.40	10,220
0.25	0.07950	2,140.80	17,852
0.30	0.12542	3,377.00	28,164
0.35	0.18437	4,964.50	41,403
0.40	0.25745	6,930.81	57,810
0.45	0.34548	9,302.21	77,500
0.50	0.44977	12,111.00	100,980
0.55	0.57060	15,365.0	128,138
0.60	0.70927	19,098.0	159,280
0.65	0.86650	23,333.0	194,580
0.70	1.0430	28,086.0	234,220
0.75	1.2392	33,369.0	278,280
0.80	1.4562	39,210.0	327,000
0.85	1.6946	45,628.0	380,510
0.90	1.9550	52,641.0	439,000
0.95	2.2380	60,261.0	502,540
1.00	2.5440	68,502.0	571,300

In Table 3 is given the weir discharges for heads varying from 0.05 to 1 foot computed from the formula,

$$Q = 2.544\, H^{\frac{5}{2}}.$$

Except for raising H to the power indicated the computing was done by means of the Fuller slide rule. The greatest error is probably not more than 1 in 1000; in many cases it is less than 1 in 10,000. The work has been checked over; still it is not impossible that a mistake may be found. In the computations one cubic foot of water was taken as 7.48 United States gallons, and one United States gallon equal to 8.34 pounds at 60 degrees Fahrenheit.

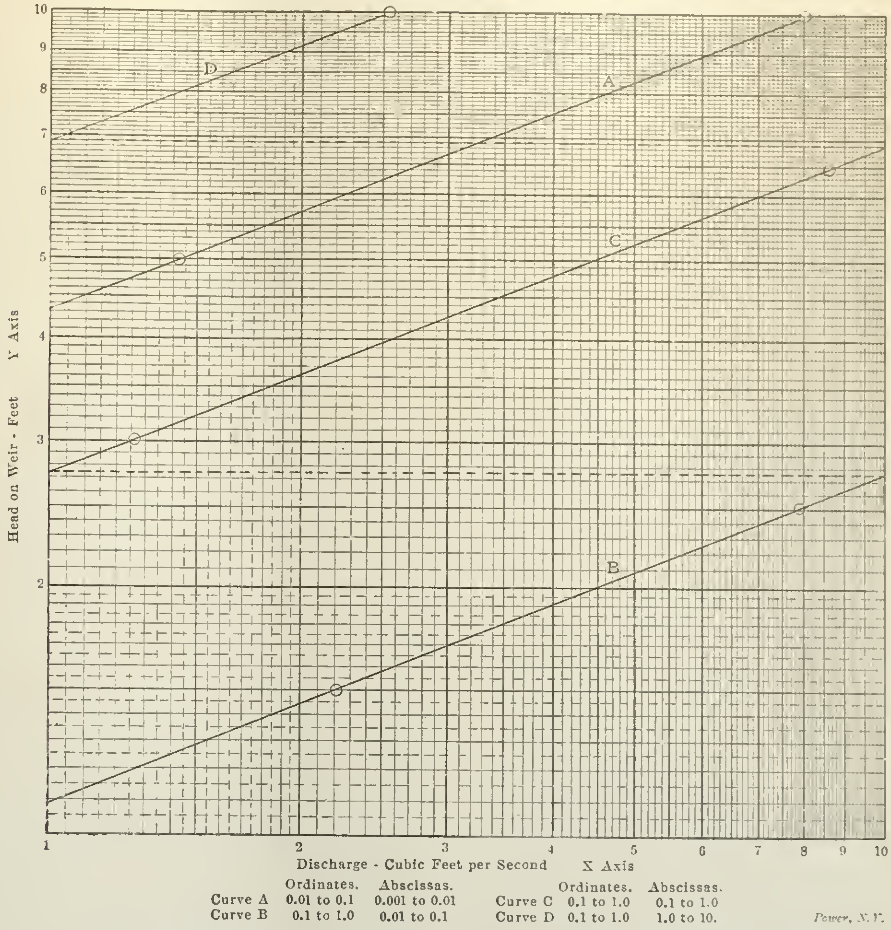


FIG. 2. DISCHARGE CURVES PLOTTED ON LOGARITHMIC PAPER

It is often convenient and sufficiently accurate to use a curve in reducing a large number of observations. When it can be done an enormous saving in time is effected. The expression which gives the discharge of the triangular notch is an exponential equation. It can be most readily plotted on logarithmic paper, as shown in Fig. 2, as this particular paper is ruled in such a way that the distance between lines is proportional to the logarithms of the numbers which they represent; that is, the distance between 1 and 2 represents the interval between the logarithm of 1 and the logarithm of 2.

In Fig. 2 curves B, C and D have the same scale of ordinates but different scales of abscissas. Curve A has a different ordinate scale as well as different abscissa scale. By this use of different scales on the same sheet a wide range of values may be plotted. To illustrate the use of the curves, suppose the head on the weir was 0.22 foot. The scale of ordinates which will contain 0.22 must be between 0.1 and 1, so beginning at 0.22 on the vertical scale follow across the sheet horizontally to the B curve, then vertically down to the X axis. The reading there is 583, and as B is laid off with abscissas from 0.01 to 0.1, it will be known that the reading must be 0.0583.

A boiler needs washing at least once every month and more often if the feed water is bad. Do not allow mud or scale to accumulate in the boiler and do not allow oil or grease to enter the boiler; these foreign substances may cause overheating and burning of plates and result in a bag, rupture, or violent explosion. A solution of soda ash or a small quantity of kerosene fed continuously into the boiler is often beneficial in preventing or softening scale. If kerosene is used, great care must be exercised to see that it is entirely gotten rid of by allowing a draft to blow through the boiler before bringing a naked light to the boiler or allowing anyone to enter the boiler, as an explosion may ensue or the fumes may overcome the person entering.—The Fidelity and Casualty Company of New York.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

An Unlucky Induction Motor

By R. H. Fenkhausen

A 20-horsepower induction motor in the plant under my charge met with so many mishaps and required such unusual repairs that it became noted as a sort of mechanical Jonah. The motor was installed in a small frame building situated on the wharf, and drove a group of machine tools used during the overhauling of a large steamer. One night the building was completely destroyed by fire and the planking of the wharf was so badly burned that the motor fell through it and went to the bottom of the bay. After the underwriters' inspection was over and some of the wreckage cleared away, a tackle was rigged up and at low tide the motor was raised from its watery bed and taken to the shops for examination and, if possible, repair, although it was thought that the sudden cooling of the machine by plunging into the water while almost red hot had warped the cast-iron frame beyond repair.

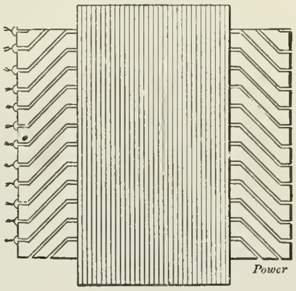


Fig. 1. Core and Winding

The motor was completely dismantled, and to our surprise the stator bore calipered perfectly round and true, probably due to the stiffening action of the heads, which were of course in place. The stator winding had been of the formed coil type laid in open slots, and as only bare copper remained, the winding was removed and the slots thoroughly cleansed of all rust and dirt.

As our electrical shop was unusually busy at the time, the job of making a new set of coils was given to an outside repair shop, and the stator frame was sent there with instructions that the coils be placed in the slots, but not connected up. This was done to make sure that the coils were not too thick for the slots, as trouble from this cause had been met with before.

While the coils were being made the rotor bars were reinsulated, the shaft trued up, the bearings re-babbitted, and the rotor balanced, as the lead balance weights had been melted out in the fire. The night before the motor was to be returned, the down-town repair shop burned down and the stator winding was again destroyed; but the owners of the shop secured temporary quarters, made new coils, and in due time delivered the motor frame with the coils in place.

Upon the arrival of the stator we again calipered the bore and found it to be so badly warped by the second fire that the rotor would not enter, but this had not been noticed by the repair shop. The absence of the heads during the second fire probably was responsible for the distortion, as the first fire was much worse than the second and no distortion had occurred.

After rejecting several schemes as impracticable, it was finally decided that the complete set of laminations with all the coils in place (Fig. 1) would have to be removed from the cast-iron frame. It was considered advisable to leave the coils in place because there is always danger of damaging the insulation in taking the coils out of the slots and, moreover, it was thought that the coils would help to hold the laminations together and to keep the teeth in line.

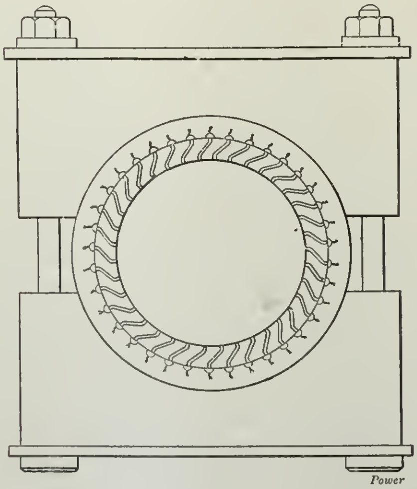


Fig. 2. Core Clamps

The laminated core was driven out with a light sledge and drift, care being used not to injure the coils. A pair of holding clamps (Fig. 2) was then made and in these the laminations were pressed back into shape. The bore was calipered and found to be perfectly round and axially true; then the rotor was tried in the stator bore and the air-gap tested. The winding was tested for grounds, crossed phases and open circuits and was found to have been unharmed by the straightening process.

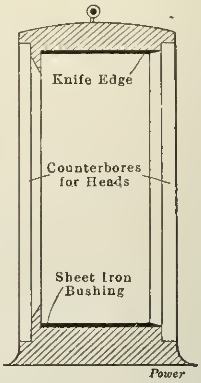


Fig. 3

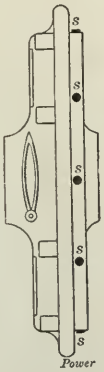


Fig. 4

The cast-iron stator housing was found to be ⅛ inch out of round, so the frame was trued up on a boring mill; both the internal counterbores for the heads and the main bore for the laminations were tooled out ⅛ of an inch larger than the original diameters, which were ascertained by measuring the laminations and the heads. A piece of 1/16-inch boiler plate was then rolled into a bushing to make up the ⅛ inch difference between the external diameter of the laminations and the bore of the frame, and this bushing strip was cut a quarter of an inch short of the length required to make a complete circle in order to provide a way for the key which held the laminations in alinement. One edge of the bushing was filed to a knife edge to facilitate the entrance of the laminations, and it was shoved into place as shown in Fig. 3. The laminations were then driven home and the key and locking ring inserted. Owing to the fact that the frame counterbores had been enlarged, the heads were of course too loose, so eight ⅜-inch studs were screwed into the heads as shown in Fig. 4, and turned off in a lathe to fit the frame.

The coils were connected up next, and as they tested clear the motor was assembled and started up. It was found, however, that the motor would not run over 850 revolutions per minute, whereas it should have run nearly 1800, being a four-pole 60-cycle machine and unloaded. After losing some time in vain attempts to locate the trouble, the rotor was removed and a pole to pole test of magnetic polarity made with a pocket compass while the stator windings were excited with direct current from a few dry-battery cells. This test revealed that all four poles in each phase (it was a two-phase machine) were of one polarity. The connections were gone over and found to be normal, as shown in Fig. 5. The explanation for this apparent paradox was revealed when the man who made the coils was questioned. It developed that he had wound alternate pole groups in opposite directions to shorten the connecting wires between pole groups, but had neglected to inform us of the fact. The stator winding was reconnected as shown in Fig. 6, and the motor ran perfectly.

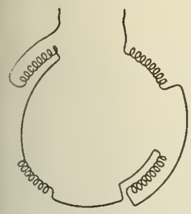


FIG. 5

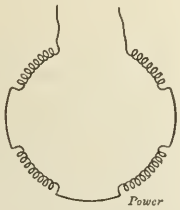


FIG. 6

After this unusual series of mishaps the motor changed its luck and has run without a mishap ever since.

Equalizing the Voltages of a Multipolar Generator

By C. G. Carlson

I have found that it is essential with some of the dynamos in my charge to get the armature electrically centered in order to prevent poor commutation, sparking at the brushes and constant blackening of the commutator. The usual way of centering an armature is to use wooden wedges, and this method, as a rule, is considered near enough to accuracy. I have found it better, however, to center the armature magnetically, although the process is longer and more tedious.

Before beginning the adjustment of the field magnet, I make a simple diagram like Fig. 1, on which to record the results of successive adjustments. The diagram here shown is for a six-pole generator; the lines represent radial center lines through the brushes of the machine, and the letters indicate the magnet poles. The armature is centered, or, more correctly, the field magnet is adjusted, to obtain equal magnetic strengths under the pole faces. The armature is driven at normal speed and the voltage of each section of the winding is measured by putting down the two sets of brushes corresponding to that section, lifting all the others off the commutator. These voltages are written on the diagram in the spaces between radial lines corresponding to the armature sections, as in Fig. 2. The figures here shown mean that in the first test the voltage between brushes Nos. 1 and 2 was 222; between Nos. 2 and 3 it was 225; between Nos. 3 and 4, 224; between Nos. 4 and 5, 204; between Nos. 5 and 6, 195; between Nos. 6 and 1, 199. After this first test the machine was shut down and the field magnet shifted and shimmed with a view to reducing the distances between the armature core and the magnet poles *A*, *B* and *C*, and increasing the airgaps under the poles *D*, *E* and *F*. Then another test was made, with the results shown in the second circle from the center, in Fig. 3. This diagram shows the results of each test and the subsequent adjustment; four tests were made before approximately equal voltages were secured between all neighboring pairs of brush studs. It is advisable to have the upper magnet poles slightly stronger than the lower ones to relieve the bearings of part of the armature weight.

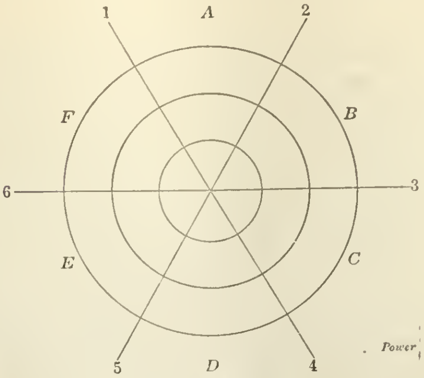


FIG. 1

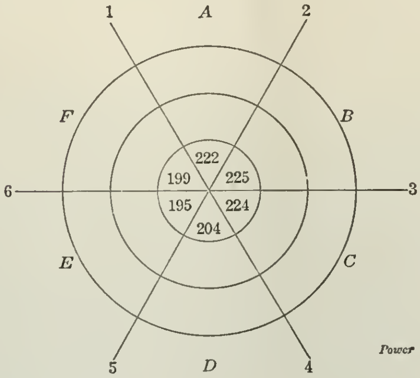


FIG. 2

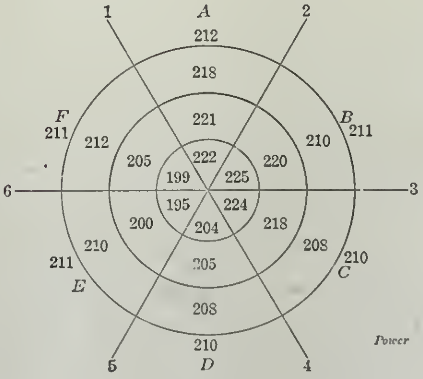


FIG. 3

LETTERS

Three Wire and Two Wire Generators

In the August 23 issue of POWER, Mr. Smith praises the three-wire generator, but from his remarks I feel sure that he is operating under conditions most favorable to this class of machine.

The chief argument in favor of the three-wire machine is that the system is operated from a single unit which can be started and operated like a two-wire machine; this is one of the advantages which Mr. Smith points out. Another reason he gives for favoring the three-wire unit is its ability to regulate the voltage of the system for fluctuating loads, but this can also be done by a two-wire generator operated with a balancing set or by the old arrangement of two dynamos each carrying one-half of the load.

My experience with the three-wire machine is that it will not regulate the voltage on either side of the system independently to compensate for unbalanced load; that is, if the load becomes heavier on one side of the system than on the other, the increased current through the series field winding raises the voltage on both sides alike, which makes that of the light side of the system abnormally high, and this unbalancing of the voltage increases as the unbalancing of the load increases. Adjusting the shunt field rheostat will not remedy the fault, since such regulation has a like effect on both sides. For systems designed and operated so that the unbalancing is slight this type of machine is well adapted, but when the system is unbalanced due to large two-wire service connections not properly divided between the two sides of the system, or to shifting of the load from one side to the other at irregular intervals, the three-wire machine will give poor regulation.

I have used balancing sets in connection with regular two-wire generators and have found that they will automatically regulate for equal voltage on both sides with unbalanced load and retain all the advantages of simplicity of connections and operation of the straight two-wire system. One balancing set will serve several generators, though for reliability of service it is best to have two sets. Actual tests have shown the combined efficiency of the generator and motor balancer to be higher than that of the three-wire generator and balancing coils.

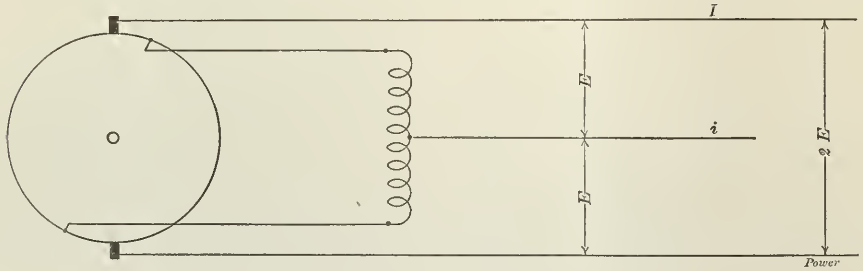
The connections of the three-wire generator are complicated, requiring two sets of series field coils, two equalizers, an extra circuit-breaker, switch and ammeter, and an extra busbar with supports. Considering the special design of the generator and the additional apparatus required, it is doubtful if the complete installation has any advantage in first cost or in maintenance.

LEWIS C. REYNOLDS.

Willard, N. Y.

The Three Wire Generator

Mr. Smith's article on the three-wire generator, in the issue of August 23, contained potent remarks on a type of machine with which many of the readers of POWER are probably more or less familiar, many no doubt having such machines under their immediate supervision. The three-wire generator combines two machines in one, resulting in higher economy of operation and of space; requires less care, lessens commutation troubles and costs less than any other equivalent equipment. The balancing coils do not necessarily require floor space and need no more attention than ordinary transformers.



ELEMENTARY CONNECTIONS OF THREE-WIRE GENERATOR

Perhaps some men operating three-wire machines have wondered why the balancing coils were so small compared with the size of the generator. Consider a machine with one coil, as represented by the accompanying diagram. If the voltage of one side of the system $= E$, giving $2E$ across the outside, the power rating of the generator equals $2EI$, I representing the current in the outside conductor. The voltage impressed on the balancing coil is alternating, and has a maximum value of $2E$, giving an effective value of $1.414\,E$. The neutral current divides on entering the coil; therefore, the current to consider is one-half of the neutral current, which may be expressed as $0.5\,i$, neglecting the small magnetizing current. The volt-ampere capacity of the coil equals $1.414\,E\,0.5\,i$, which is equivalent to $0.707\,E\,i$. Then the ratio of the volt-ampere capacity of the coil to that of the machine would equal

$$\frac{0.707\,E\,i}{2\,E\,I}$$

which reduces to

$$0.354\,\frac{i}{I}$$

Now, it is evident that if the neutral current i equaled the current I in one of the main conductors, or in other words, if only one side of the system were loaded, the coil would have but 35.4 per cent. of the generator capacity, and with two coils, as is generally the case, the capacity of each would be approximately 18 per cent. of that of the machine. As the coil is an autotransformer with a one-to-one ratio, it would have to be but one-half the size of a transformer of the same volt-ampere capacity, or 9 per cent. of the capacity of the generator.

But the neutral current, instead of being equal to the current in the outside conductor, is seldom if ever more than about 25 per cent. of it, so the coil would need to be the size of a transformer of but 2.25 per cent. of the generator capacity, which would be very small.

Three-wire generators under ordinary conditions of load regulate very closely and require no more care than the ordinary type of generator. They are paralleled as easily, no account having to be taken of the balancing coils, and for the average installation using three-wire direct-current distribution they are hard to equal for all-around satisfaction, especially to the man who has the care and the worry.

A. C. KERR.

Ft. Monroe, Va.

Unbalanced Airgaps on Induction Motors

Mr. Orr's experience with an induction motor which would not run with an unbalanced airgap but ran all right as soon as the airgap was equalized around the rotor agrees with a number of experiences I have had along the same line. A number of years ago a 5-horsepower 110-volt single-phase induction motor came in the shop as having been "burned out." It did not look very bad, so we connected it up and threw in the switch; it refused to move—just stuck and roared—and I could not move it by dragging on the pulley with both hands, although without the current it revolved freely. I took it apart but an examination showed no burned coils. We tried everything we could think of and finally one of the bearings was raised a little before the bolts were drawn down tight and the motor started off like a new machine. We had accidently found it. We shimmed up the bearings to equalize the airgap and heard no more from that "burned-out" motor. A short time ago I had a similar experience with a ½-horsepower motor driving a vacuum cleaner. This was also "burned out." I threw in the switch and it showed all the symptoms of an unsymmetrical airgap; then I raised one end of the shaft and it immediately started. A new bearing was secured and the motor has given no more trouble.

The airgap on induction motors is very small and "finicky"; much more so than on direct-current motors, and successful operation cannot be obtained with an unequal airgap. In neither of the two cases I have cited was the rotor touching at any point; it seemed to stick as if a powerful magnet were holding it at one point, and that is practically what takes place. Most manufacturers of induction motors send out an airgap gage with each of their motors; this is merely a thin strip of steel ground to the same thickness as the airgap and it should be possible to insert it at any point of the airgap; if the gap is too small at any point the bearings should be adjusted to permit the gage to enter.

I have had three different cases with 50-horsepower motors caused by melting the babbitt in a bearing and I always caution the machinist about the necessity of having the rotor revolve exactly in the center of the space and not merely to move freely.

G. S. SPRAGUE.

Geneva, Neb.

The induction-motor trouble experienced by Mr. Orr, as outlined in the issue of August 23, was probably due to the bearing having been worn away on one side to such an extent that the rotor was pulled against the stator when current was passing through the stator winding, although it did not quite scrape when the current was off.

Very often an induction motor with loose cap square bolts, especially if the motor is of small size, will refuse to run until the bolts are tightened down, due to the fact that the rotor, being somewhat above the center, is pulled against the stator by the magnetism when the stator winding is energized. Mr. Orr's trouble was probably similar.

A. C. KERR.

Ft. Monroe, Va.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

An Interesting Gas Power Factory Plant

By Osborn Monnett

Gradually the gas engine is making its presence felt in the larger industrial institutions in competition with other forms of motive power and is beginning to hold its own in fields where steam was the only thing thought of a few years ago. The case under consideration is an interesting example of the encroachment of gas power upon the field of steam, and the methods employed to insure the same character of service, not only for power but for heating the buildings and supplying hot water for various purposes, are notable as indicating the probable trend of development along this line.

The manufacturing establishment for which this plant furnishes the power is that of the A. O. Smith Company, of Milwaukee, maker of automobile parts. The power plant occupies one end of a brick and steel fireproof building, the other end of which is devoted to the blacksmith shop, annealing ovens, etc., the two departments being separated by a fire wall. Electrical distribution is used exclusively. The generating unit is an Allis-Chalmers tandem double-acting gas engine, with 24x30-inch cylinders, direct connected to a 400-kilowatt direct-current Allis-Chalmers generator and running at 150 revolutions per minute. The engine, as is common practice in larger sizes, has an intermediate crosshead between the two cylinders and a tail rod supported by a crosshead at the rear, both running in bored guides resting on the foundation and designed to support the weight of the pistons and piston rods so that they may travel without imposing weight on either the walls of the cylinders or the stuffing boxes.

The inlet valves are located at the top and the outlet valves at the bottom of the cylinders, as in most double-acting engines. The frame is of the open type, which gives easy accessibility, with the slipper crosshead used in the larger engines of this make. The jaw in which the main bearing shells are held is especially heavy and rigid and the bottom bearing shell is water cooled. This part of the frame, which must receive the full thrust

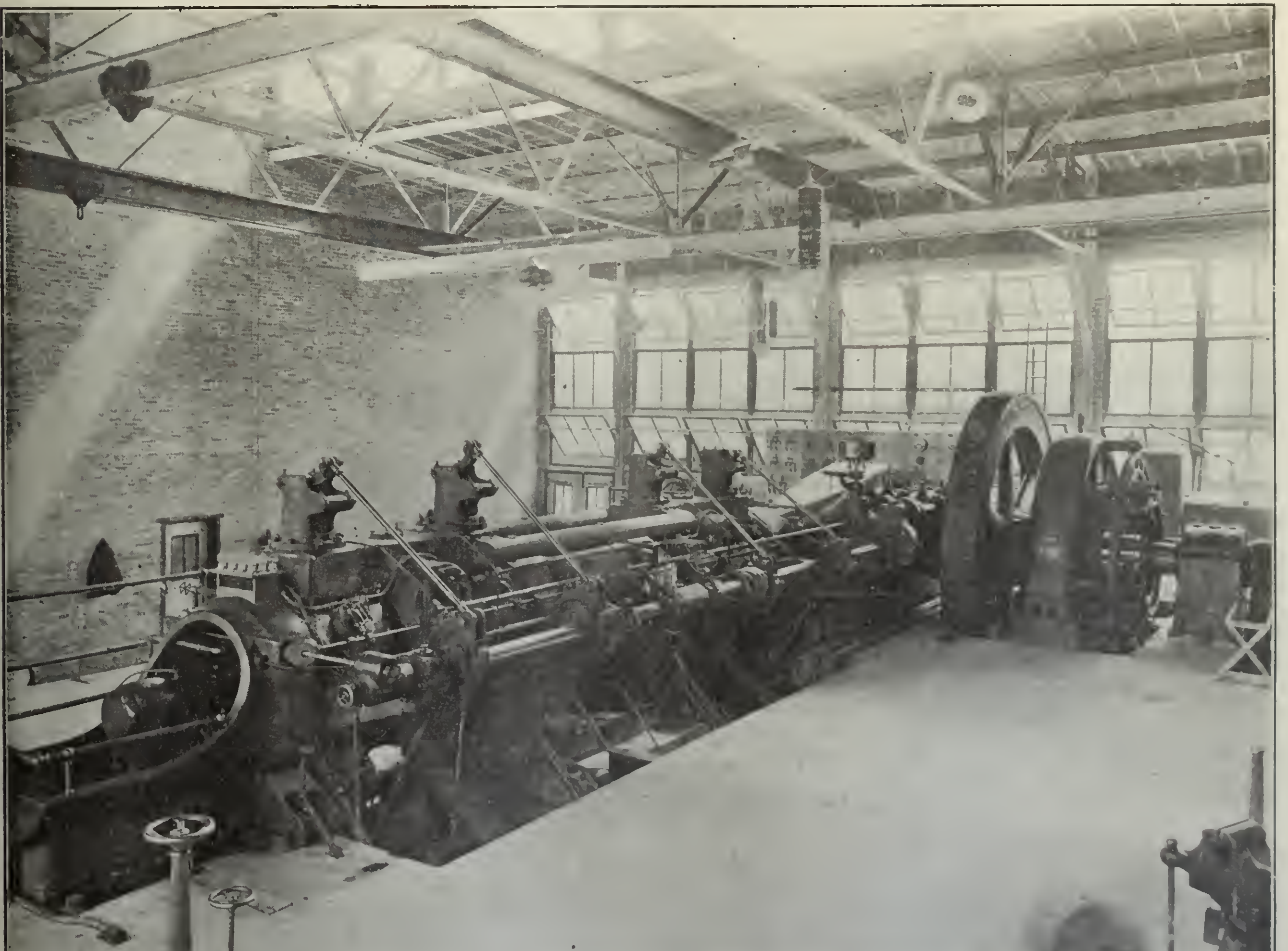


Fig. 1. Generating Unit of the A. O. Smith Factory Plant

of the explosions is further strengthened by two steel tie bolts extending across the jaw above the bearing. A feature of the engine differing from most of the other makes is that the two cylinders and the distance piece between them are cast in one piece. The cylinder casting proper is subjected to no wear, as the working bore is fitted with a hard cast-iron liner to take the wear due to the motion of the piston.

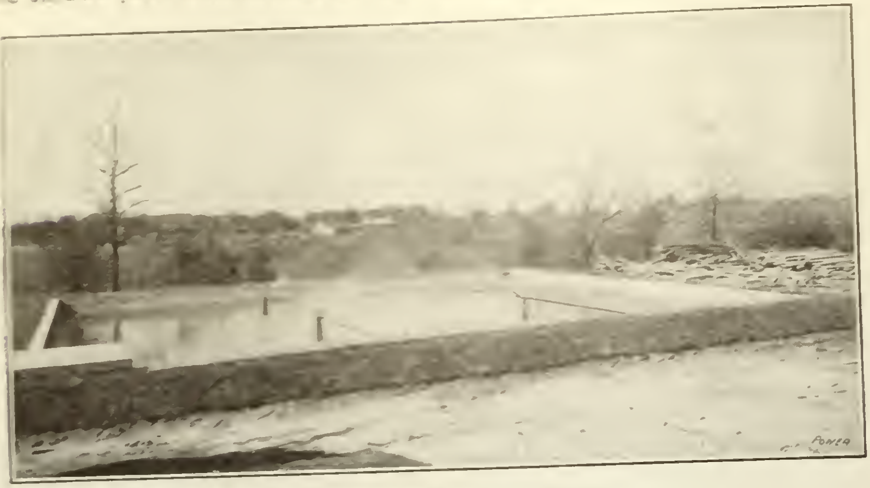


Fig. 2. Cooling Pond and Sprays

Auxiliary Details

"Quantity" regulation is used, the admission of mixture being controlled by a fly-ball governor driven from the valve-gear shaft. The gas and air inlet conduits are cast integrally with the cylinders and the valves themselves are of the simple poppet type, mounted in removable cages. Mechanical make-and-break ignition is used, and two igniters, located at well-separated points, are provided in each end of each cylinder. The electrical ignition equipment consists of a 30-volt dynamo mounted on the main engine, operating in conjunction with a storage battery, and a coil for each working cylinder connected in series multiple with three low-voltage lamps which can be connected or disconnected to vary the strength of the spark.

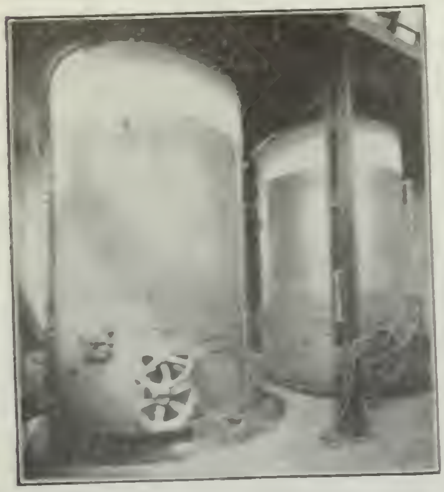


Fig. 3. The Gas Generators

The engine is provided with an air-starting gear which, as usual, is normally disengaged and inoperative. On opening the air throttle the air pressure automatically throws the starting valve gear into engagement with cams on the valve-gear shaft which open and close the air valves in proper time to start the engine.

Lubrication is effected by means of a gravity system, a pump on the valve-gear shaft returning the oil to an overhead tank after it has passed through a White Star filter. The excess oil handled by the pump overflows from the storage tank back into the pump suction. For internal lubrication Richardson sight-feed oil pumps are provided, the oil being introduced at two points in the cylinder timed to give the maximum benefit.

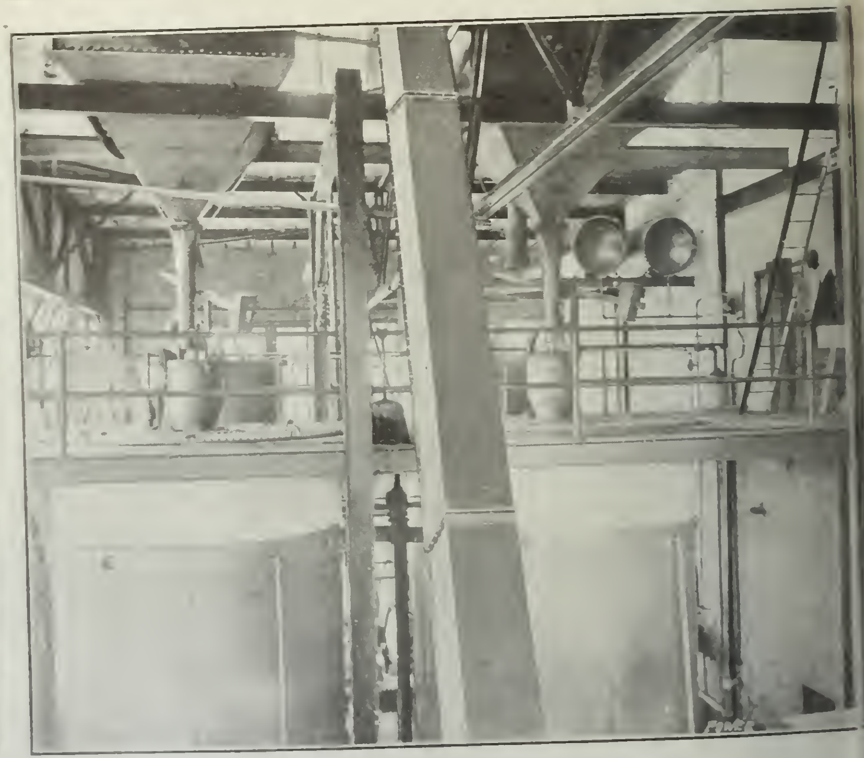


Fig. 4. Producer-charging Platform and Coal Elevator

Cooling Arrangements

The cooling system has been desig d with a view to minimizing the consu tion of water, which must be bought f n the city. A concrete cooling reserv , approximately 50x60 feet in size d holding about 88,750 gallons of wa , has been provided from which to d w the water supply from the engine. Loca d in the basement and belted to the n n engine shaft is a geared pump which livers water to the engine jackets s needed. The cooling water passes thro h the piston, exhaust-valve cages and st s separately from the cylinder jack , and valves are provided to regu e the flow to the different parts indepe ently. The overflows are placed in p n view of the operator and the discha e water is collected in a tank in the ba ment from which it flows by gravity o Koerting spray nozzles over the coo g reservoir, shown in Fig. 2.

Producer Equipment

In the producer room are two horsepower Wood producers, the erators of which are equipped with sh ing grates through which the ashes e delivered to a water-sealed pit and h vaporizers in the top for furnishing e necessary steam. The gearing thro h which the grates are shaken by m of a hand crank appears in Fig. 3. 5-horsepower motor-driven blower nishes an air blast directly above grates, the producers being of the p sure type. Scrubbers of the usual al type are provided after passing through which the gas goes to a centrifugal scrubber and tar extractor (Fig. 5) driven by a 25-horsepower motor; from this machine it is delivered to a small gas holder outside of the building, whence it is drawn by the engine as needed. Coal is unloaded from cars into a bin of 100 tons capacity, into which dips the leg of a bucket elevator operated by a 5-horsepower motor; this outfit raises the fuel to small individual hoppers over each producer (see Fig. 4).

On the operating platform of the producers (see Fig. 4) is located a gage board carrying U gages which register the gas pressure in the collecting main leading to the gas holder and the air pressure under the grate and above the fire in each producer. There is also a U gage one leg of which is connected to the air blast and the other to the steam generator, the gage indicating the excess pressure of the blast over the pressure of the steam which is generated in the vaporizer.

Several kinds of fuel have been tried in this installation, the first being coke breeze at $1.75 per ton, followed by pea size coke at $2.50 a ton. There was found to be a surprisingly large amount of tar in this fuel.

The large volume volume of coke which had to be handled was also an objection, and the fuel finally decided upon was pea anthracite, which costs $2.60 per ton delivered on the side track at the plant. It is stated that one ton of the anthracite coal has been found equal to 3½ tons of the pea coke.

WASTE HEAT UTILIZATION

One of the most interesting features of this plant is found in the arrangement for utilizing the waste heat of the exhaust gases from the engine. In the main exhaust line of the engine is inserted a "transverse current" water heater containing 334 square feet of heating surface, from which the exhaust line continues to an exhaust muffler outside the building and thence to the atmosphere. An interesting detail of this equipment is that the entire exhaust pipe, water heater, muffler, etc., which extends in a straight line from the engine foundation, rests on rollers and is free to expand or contract under the varying influence of the exhaust heat. Where the exhaust line parallels the foundation it is supported on arms projecting at an angle and resting in ball-and-socket joints embedded in the foundation.

The heater, a close view of which is shown in Fig. 6, is the Rinie apparatus which was described in the issue of April 19 in connection with some brief tests made in connection with a Westinghouse gas engine.

No attempt has been made to economize the heat absorbed in the water jackets but a test of a heater alone, placed in the exhaust line, has, on a smaller installation, shown a return of over 32 per cent. of the total heat supplied to the engine. It is unfortunate in this case that so far the hot water supplied by the heater has been only partially utilized, being circulated in the shop radiators and the lavatories. For the present the excess hot water is returned to the cooling pond and used over again in the engine.

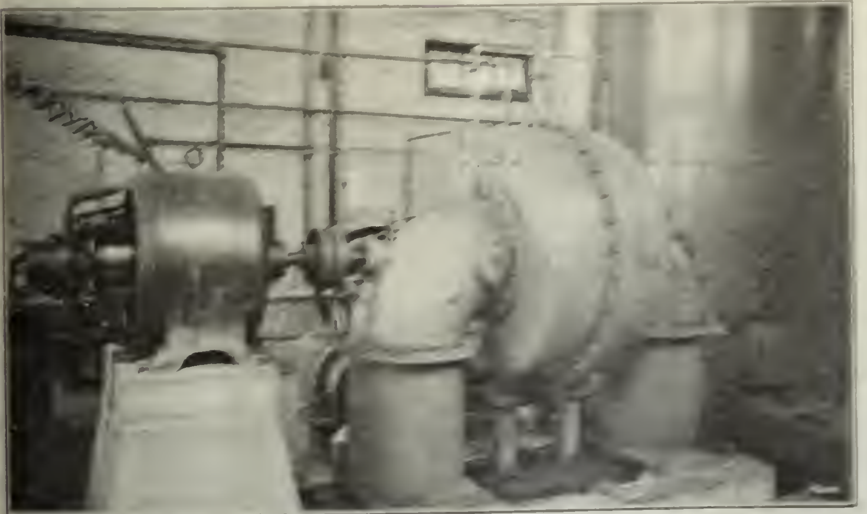


FIG. 5. MOTOR-DRIVEN TAR EXTRACTOR

HEATING PLANT

One of the carefully worked out details of this plant is the installation for heating the shop which is 285 feet wide by 1050 feet long, all under one roof without partitions. The heating is to be done with hot water, as intimated above, there being installed two Kewanee gas-fired boilers of the locomotive type, specially designed for this plant. In each side of the firebox is a circular flue, one of which is to receive the hot waste gases from the annealing ovens in the adjoining part of the building and the other to receive the exhaust of the gas engine after it has passed through the hot-water heater. Some objection has been urged against this utilization of the gas-engine exhaust on the ground that the sulphur contained might have a deleterious effect on the boilers. However, it must be remembered that this waste gas will, in the firebox, come in direct contact with the hot waste gases from the annealing ovens and possibly a bright coal fire on the grates which will aid in preventing the condensation of any sulphurous gases on the heating surface and flues of the boilers. As far as the heater in the exhaust line is concerned there are only two water joints to be made and if these are kept tight it is improbable that sulphuric action will occur in this piece of apparatus, inasmuch as the entire structure is of cast iron.

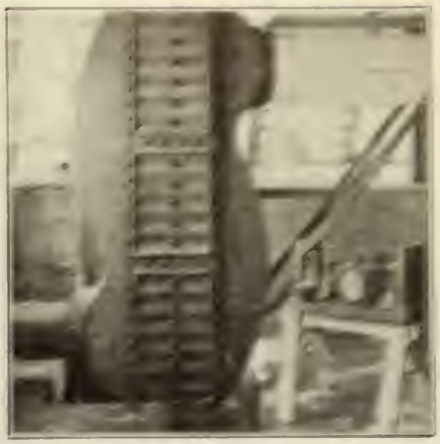


FIG. 6. WATER HEATER

Working these possible combinations it is expected that a large amount of the factory heat will be supplied this winter without appreciable extra fuel expense, which will materially increase the overall economy of the installation. The Cooper hot-water heating system has been installed in the shop. This consists of the necessary equipment of radiators through which the hot water is circulated by a motor-driven rotary pump. The temperature is automatically controlled by thermostats.

The arrangement for using the hot-water heater in series with the boilers in cold weather and taking advantage of the waste gases of both the engine and the annealing ovens was worked out by Jesse C. Cooper, of Milwaukee, who installed the heating system. It is conservatively figured that the waste heat available will be ample to heat the factory satisfactorily during at least four months of the total heating season of eight months.

SOME OFFICIAL FIGURES

This plant was put in under keen competition from steam-plant manufacturers because of the belief that the economy over steam would justify its installation. It has been operating ten hours per day for five months and according to numerous running tests one kilowatt-hour is being produced for one pound of pea-size anthracite coal containing 11,500 B.t.u. per pound. The total cost of the producer and gas-engine installation was $50,000 as against a proposed noncondensing steam installation costing $25,000 for 750 horsepower rated capacity of boilers and engines. On this steam plant the builders would not guarantee less than three pounds of coal per horsepower-hour running noncondensing and exhausting to the atmosphere. The hot-water heating system, consisting of the two locomotive-type boilers and 40,000 square feet of direct coil radiation, was installed at a cost of $19,000, which carries the cost of the entire installation up to $69,000. If the noncondensing steam plant had been installed it was estimated that a vacuum steam-heating system would have cost $30,000, making the total steam installation cost $55,000, or $14,000 less than the present plant investment.

The power plant is now consuming 2100 pounds of coal for ten hours' work which, at $5.60 a ton, amounts to approximately $5.90 per day, or a total of less than $1950 per annum for fuel. Adding to this the estimated consumption by the heating system, during the coldest months of the year, of 900 tons of steam coal at $3 a ton, makes a total fuel cost for heating and power of $4650. These figures show a saving of over 60 per cent. per annum in fuel over the estimated performance of a steam plant. Moreover, it was figured that if a condensing plant had been installed to realize better steam economy, the additional cost of a cooling tower, which would have been necessary because the water is bought from the city, would have materially increased the investment charges for the latter type of plant.

Some Reminiscences of Early Days

By A. E. Walker

A number of years ago, when trying to start an engine that had been just installed, I had a very narrow escape from being knocked out by it. The lay shaft for driving the valve gear had not been properly marked by the makers and we got it in gear wrongly with the main shaft, with the result that when we started up, the engine tried to run backward.

We shut off the gas supply to the cylinder, leaving the ignition-tube burner alight to save time in starting again and pulled the wheels round two or three times to make sure that no gas remained in the cylinder. To adjust the lay shaft, we had to jack up the main shaft until the gears cleared, and we had just done this, and I was kneeling on the floor in front of the engine turning the lay shaft round, when suddenly there was an explosion in the cylinder and the shaft and flywheels came rolling forward (it was a single-cylinder horizontal engine).

What saved me from being run over by one of the ½-ton flywheels was the fact that the shaft wasn't packed up high enough to clear the studs on the main bearings, which all got badly bent; the only other damage was a broken crank-pin oil wiper. The only explanation I could think of was that the gas-stop valve leaked, so we took the plug out and reground it, but I never worked on a gas engine again with the ignition tube hot.

One night the man in charge of an engine and dynamo for lighting a jeweler's shop came round and asked us to send someone at once to see what had gone wrong with his engine, as the cellar was full of smoke though the engine was still going. When I got to the shop the smoke was pouring up through a ventilator under the window and driving people off the sidewalk. Upon investigation I found that the floor had been cut out to clear the flywheel, and the hole had been allowed to fill up with waste-bearing oil; when the engine was started the wheel threw this oil over the hot exhaust-valve box.

At one small repair shop where I worked once we had a 2-horsepower gas engine solely for driving a dynamo for lighting the shop. This was located in a cellar, and as usual, the ignition-tube burner was tapped off a separate pipe not in connection with the supply to the engine cylinder, the idea being to avoid fluctuations. This burner happened to be supplied from the same pipe as a few gas jets we had in the shop and thereby hangs a tale.

One day, the boss had gone out, and the apprentice who usually looked after the engine and dynamo had gone below to start up, as it was getting dark. Some bright specimen of humanity upstairs suggested waiting until he got the engine started and then blowing down the gas pipe and putting the ignition-tube burner out. Everyone fell in with the scheme and we soon had a man at each gas bracket.

By and by the electric lamps began to glow and then at a given signal every man put his mouth over a gas burner, opened the tap and blew for all he was worth. Two or three spare men listened for the fun on the stairway. It was a short pipe and a few moments of blowing put the burner out; after a few big bangs, backfires and misses, the engine shut down. After relieving his feelings the apprentice lighted the burner again and when the tube got hot, put the starting handle on again and turned her over while the wind-bag brigade upstairs got ready to send another cyclone down.

The trick was repeated several times and when the apprentice had given the engine up in disgust, the old man came down and wanted to know what was wrong. The boy told him he could not get the burner to keep alight, and the boss suggested trying again, and of course it went all right to the boy's utter amazement, and he never found out why that engine did stunts.

CORRESPONDENCE

Setting Small Gas Engines

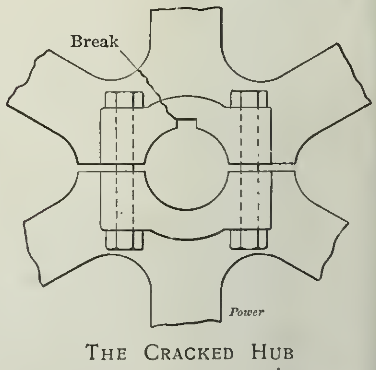
I read with much interest Mr. Wilson's description, in the issue of September 13, of his method of setting gas engines. The theory is all right, but I should like very much to know what parts of the engine the jacks are under when it is let down on the soft surface grouting, and how the base is kept truly level and the shaft in line with the driven shaft while the cement is setting. The process strikes me as a marvelous acrobatic stunt if the engine weighs more than a few hundred pounds.

Geo. W. Malcolm.

Brooklyn, N. Y.

Flywheels Broken by Preignition

A few months ago a 15-horsepower, single-cylinder, center-crank engine with two flywheels met with a peculiar accident. The engine had been running on a heavy load for some hours when it was found necessary to shut down for a few minutes on account of some trouble in the factory. Before the engine had come up to speed, after starting again, an explosion occurred on the compression stroke which brought it to a standstill. Another start was made but the engine pounded so badly that it was shut down. Upon examination it was found that the hubs of both wheels were cracked from one corner of the keyway through the hub, as shown in the sketch. Both hubs were broken in almost exactly the same place.



The Cracked Hub

Frank E. Booth.

Toronto, Can.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Self Ignition of Coal

Probably all engineers have had trouble at one time or another with self-ignition of the coal pile, and many will be able to call to mind occasions when this has been the cause of considerable expense and loss through the deterioration of the coal. The heat generated is at times very great and sufficient to give rise to serious fire risk. I came across one case in which the steel plates covering the bunker floor were warped and twisted completely out of shape. Generally speaking, self-ignition arises from improper methods of storing and failure to appreciate one or two fundamental principles. The root of the whole matter lies in the oxidization of the coal which is set up by moisture or heat or the two combined. It is slow but cumulative in effect, since the warmer the coal gets the more it oxidizes, and given the favorable conditions which so often obtain in warm boiler rooms, ignition is only a question of time once the heating has begun.

To prevent trouble from this source it is only necessary to take one or two simple precautions. If the coal is delivered wet it should not be heaped if this can possibly be avoided, but should be spread over a large area to dry and heaped afterward. This means extra expense, but is better than a fire or deterioration in calorific value. Coal should not be stored in a warm place and it should never be heaped more than 12 feet deep. The bunkers should be of iron or concrete, and should not be placed, as is sometimes the case, near to flues or boilers. If the heap is exposed to the weather the floor should be of concrete or steel with a decided slope so as to allow water to drain off rapidly. The chief precaution to take, however, is to arrange for efficient ventilation of the heart of the pile by means of spaces around the sides and at the bottom. This is seldom done, and when attempted the means taken are often quite inadequate. For instance, it is of no use to place air tubes in the heap unless they are sufficiently large to allow enough air to pass to carry off the heat generated. Otherwise, they simply assist oxidization by conveying air to the center of the pile. Such passages should be large and arranged according to a proper system of ventilation which will cause a constant current of air to flow. Wicker tubes have been employed with success, the coal being heaped around them after they are placed in position.

Bituminous and semi-bituminous coals are most liable to overheat. The more water, sulphur and volatile matter they contain the greater is the risk. A proportion of water exceeding 4.75 per cent. is sure to result in trouble unless precautions are taken, and if at any time the temperature of the center of the pile reaches 140 degrees Fahrenheit, ignition may be anticipated. To be on the safe side it is necessary to watch the temperature of a coal pile carefully. Perhaps the best means of doing this is to keep a few iron bars embedded in the heap right down to its core. It is only necessary to withdraw one of these and to attach a thermometer to it at the lower end by a handful of waste to ascertain the temperature of the pile. Experienced men can tell by touch whether the temperature is dangerous. It is better, however, to use a thermometer than to rely upon such rough and ready means.

If proper precautions have not been taken and the pile ignites, practically the only thing to do is to turn it over thoroughly, and when the heart is reached to pour water over it. It is useless to deluge the heap with water without first turning over, as probably only sufficient will penetrate to the core to make matters worse. If for any reason the pile cannot be turned over an effective alternative is to drive a good-sized pipe having holes at its lower end into the center right through the layer of coked fuel which surrounds the fire. A powerful stream of water should then be turned on. It is, of course, necessary to watch the heap very carefully after this treatment, as its damp state will be conducive to another outbreak.

FRANCIS H. DAVIES.

London, England.

Trouble with a Pumping System

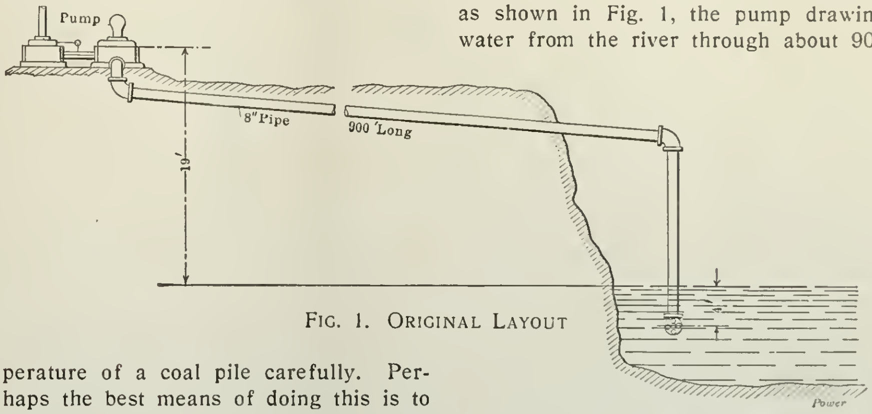


FIG. 1. ORIGINAL LAYOUT

Our pipe line was originally installed as shown in Fig. 1, the pump drawing water from the river through about 900 feet of 8-inch cast-iron pipe. We were unable to make the pump hold the water; it pumped all right for a few hours and then lost water and we were obliged to prime it. We finally changed the location of the pump to that shown in Fig. 2, and experienced no more trouble of this kind, although it would not furnish as much water to the tank as formerly.

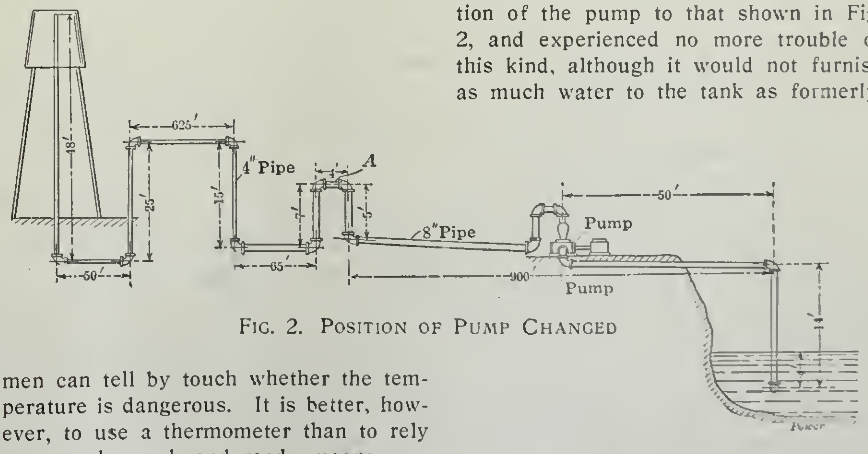


FIG. 2. POSITION OF PUMP CHANGED

The way in which the foot valve is incased is shown in Fig. 3. The rip rap extends out into the river for about 50 feet at the end of which is placed a box made of 3-inch plank, having a steel strainer on the side opposite to the direction of the current.

Some of the attendants claim that the trouble was due to the hight of the suction and the length of the suction pipe. I do not agree with them. I claim that if the pump had been in good condition and the pipe line and foot valve all right, the pump would have handled the water. Will some reader of POWER kindly solve the problem?

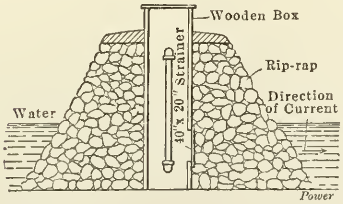


FIG. 3. FOOT VALVE CASING

V. T. KROPIDLOWSKI.

Winona, Minn.

Installing Valves

When installing globe, angle or gate valves or plug cocks trouble is often caused by the workmen placing the wrench on the hexagonal flange farthest from the pipe while screwing the valve onto the pipe. This springs the seat and will often distort the valve sufficiently to cause a leak. The wrench should be placed as shown in the accompanying sketch. At the opposite side of the valve is shown a nipple; when the fitting is to be screwed up on the nipple the wrench should be placed on that side of the valve to support it against the pressure.

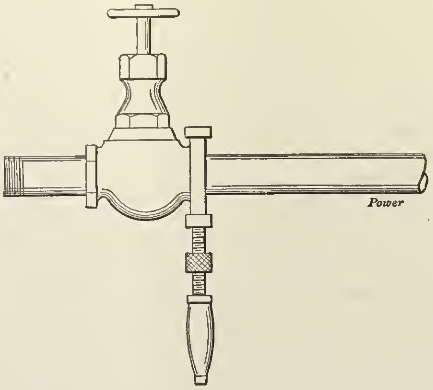
Many cases have come under my personal observation where the steamfitter has instructed his helper to place a pair of pipe tongs on the pipe at the other side of the valve while he screwed on a fitting. When the valve began to turn, the helper moved the tongs onto the hexagonal flange of the valve, marking it badly and leaving an unsightly trade mark of the man's carelessness.

It is good practice to leave a valve closed while screwing it onto a pipe. The plug should never be removed from a cock when screwing it on. Care should be taken to have the hangers as near as possible to valves on pipe lines, for it is poor practice to allow the weight of a pipe line to be carried by the valves; many leaky valves can be traced to this oversight.

I remember once watching a steamfitter who had a pot of red lead and a brush and was very careful to give the inside of every fitting a liberal coating of the paint. As I was to be in charge of the plant, I requested him to place the red lead on the outside of the pipe threads, explaining that some of the red lead, as he was using it, would probably catch on valve seats and then grit or scale would be collected at the oily point and probably cause a leaky valve. He took my request in good part and did as I advised, but I am sure the majority of careless pipefitters would have been too pig headed to take advice from a stationary engineer; for most steamfitters have a very poor opinion of the average engineer.

A steam valve should be closed tightly, for it contracts when it cools and a small amount of water passing through a valve will wear out the disk or seat much faster than a full stream. The reason for this is that a small leak will wear the disk and seat unevenly.

When a valve is closed and it continues to leak, do not use a wrench or extra force to close it. Probably there is a piece of scale under the disk. Open the valve quickly and allow a quick rush of steam through it; this often proves effective in removing anything which may have become jammed between the disk and the seat.



VALVE SUPPORTED BY WRENCH

I have seen valves spoiled by men carelessly screwing down the bonnet when the spindle was too far down. A valve should always be open when removing or putting on a bonnet. Another source of trouble is from cutting the pipe threads too long. When the valve is screwed on, the end of the pipe is forced against the seat and the latter is often sprung.

JAMES E. NOBLE.

Toronto, Canada.

Aid in Cutting Pipe Threads

The following is so simple a trick that doubtless it is a chestnut to many; however, it costs me nothing to relate it and the information may prove of value to some.

Recently I had occasion to cut a couple of 1¼-inch pipe threads with the old-style solid die. The only dies of this size that were at hand were so badly worn that no amount of persuasion would induce either of them to start a thread. I used a little kerosene instead of oil, in starting, and the old die took hold without extra effort, cutting a nice, clean thread. Perhaps turpentine would have answered as well, but kerosene is more generally at hand and works as well as can be desired.

G. M. JOHNSON.

Escanaba, Mich.

Pump Troubles

Pump troubles are numerous and about as varied as they are numerous. In a plant equipped with two single Blake feed pumps, I had considerable trouble with one. I did not have as much trouble to make it run as I did to find out just what was wrong with it. It would hang during the head-end part of the stroke whenever it was slowed down to one-quarter speed. It would run a few strokes at the desired speed and then stop at the head end. These pumps were new and I could see no good reason for a failure to operate. Thinking that the trouble lay in the valve setting, I changed the position of the tappets on the valve stem, shortening the stroke on that end. The trouble still continued. Having moved the tappet to the end of the thread on the valve stem until the piston struck the head of the cylinder on the opposite end, I moved the tappet on the other end of the valve rod and shortened the stroke from its greatest to its shortest length, thus making the stroke first uneven at one end and then at the other. As I observed no improvement, I concluded that the trouble was due to the finish of the castings. When I removed the steam chest, the trouble was plainly visible. On the outside edge of one of the ports a V-shaped piece had been broken out. The break was about 3/16 of an inch long and ¼ of an inch wide at the edge of the port. This allowed the admission of steam to the cylinder after the valve had traveled the required distance to close the port. The steam thus admitted, overcoming the compression at the opposite end of the stroke, caused the piston to strike the head at that end. The metal was so thin and soft that it was not advisable to attempt to set in an edge piece or even put in a flush-headed screw.

There were signs of other troubles, anc during the experiments the pump stoppec dead at the beginning of the stroke. I could not be moved with a bar. Takin off the head of the steam cylinder I founc the counterbore to be short enough t prevent the metal packing rings from getting caught in it. I then took off th cylinder head of the water end and removed the follower plate. I found tha the packing had become worn and loos and two rings had dropped into th counterbore, which was too long, and held the piston tight.

The pump had been partially spoiled by a careless machinist. It had been used for a number of months without the engineer or purchaser discovering its defects. During that time it had been paid for. As it would run at full or half speed it was hardly likely that the manufacturers would replace it, so the superintendent told me to do what I could to make it work better.

I shortened the stroke on this end and put a 3/16-inch leather washer ahead of the tappet and a ⅛-inch leather washer on the opposite end. In this way the tappets were in a position which allowed the longest possible stroke without causing the pump to stop. Should the leather washers get worn or accidentally come off, the pump would stop without the water piston overriding the counterbore, the counterbore being at the same end of the stroke as the defective port. To stop the thump of the piston against the cylinder head I had to partly close the exhaust valve. In other words, I operated the pump under heavy back pressure by regulating the speed by the valve on the exhaust pipe instead of the throttle valve. The pump is still running in this manner although the loss in economy would have more than paid the cost of counterboring and rebushing the water cylinder as well as the cost of a new port seat for the steam cylinder.

The leather washers could be removed while the pump was running and thicker or thinner ones put on to suit the speed required and quantity of water delivered.

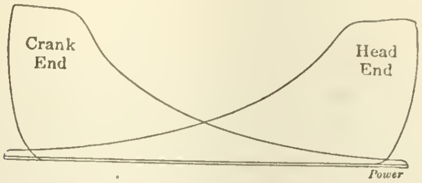
One other trouble I experienced with the pumps in this plant. One day one of them slowed up suddenly, coming almost to a stop but reversing at each end of the stroke sharply and evenly. After working over the pump for a few minutes I noticed as I tried all valves along the discharge pipe that the water level was rapidly failing in the gage glass on the boilers, which were of the water-tube type. The air cock on the discharge chamber showed that there was much pressure in the discharge line. At this point my thought turned to the check valve. The water in the boilers being at the lowest level and not having any other way of supplying them, we were forced to shut down long enough to remove the check valve. We found that it had lost the nut which held it into the hinge and had come out, turned over and wedged tight against the end of the pipe, shutting off the water almost completely. There being a combination stop and check valve and a stop valve on the line, we were able to repair the defective check without draining the boiler. The plant was running again in less than 30 minutes.

R. A. Cultra.

Boston, Mass.

Diagrams for Criticism

The accompanying diagrams were taken from a 14x42-inch Corliss which has been in service for 37 years and has never been rebored. It is evident that the work is pretty evenly divided between the crank and head ends. I should like the opinions of some brother engineers regarding the diagrams.



Harry C. Leavitt.

Lynn, Mass.

Insulating Flanges

The question of insulating flange unions on steam lines has arisen in a large plant recently. From the viewpoint of the firm and of the engineer in charge, it seems of doubtful economy.

In the first place, the cost of the labor and material required to do the work is equivalent to at least one-half of that required for all of the straight work. The work is often done before the joints have been thoroughly tested. With covered flanges bad leaks often grow from very small ones that might have been stopped at the start by drawing up on the bolts had they been accessible. When it becomes necessary to open a joint and put in a new gasket, the nuts are usually so rusted and the threads so corroded that many bolts have to be broken off before they can be removed.

I am of the opinion that a much more satisfactory joint can be made by simply tapering the pipe covering down to the base of the flange. Thus, the joint can be watched and kept right. The flanges may be coated with a water-color paint which will harmonize in color with the rest of the piping or a different color may be used, as preferred.

Edward T. Binns.

Philadelphia, Penn.

A Cracked Cylinder

We have a 14x42-inch Corliss engine which has been running for nine years without giving any trouble. A short time ago, when starting up in the morning, I noticed water dripping from underneath the cylinder; but later in the day it seemed to almost stop and dripped only occasionally. At first I thought the packing in the bonnets or the flanges around the valves leaked, as I could not see them on account of the covering.

I removed the covering and found the felt packing to be soaked with water and useless. Looking further, I discovered a circumferential crack at the head end of the cylinder, starting from a point at the lower left-hand corner near the exhaust port, and extending up 7 inches on one side. I was unable to determine how far it extended on the other side on account of the valve gear cutting off my view. When I notified my employer, he said, "I suppose that means a new cylinder," and told me to closely observe the crack.

Now, the question which confronts me is: What is the best way to remedy the trouble? Can it be remedied without getting a new cylinder? Also, is it safe to keep on running with the cracked cylinder? The pressure used in the cylinder is 75 pounds.

C. G. Harrison.

Chicago, Ill.

Ice Machine "Mystery"

The ice machine in a brewery was shut down by the day engineer at 7 p.m. on account of a boiler that had to be repaired, and as they brewed that night, it could not be started up (not having power enough for both kettle and machine) until the beer had been brewed. The night engineer let the water run over the condenser before he went home at 7 a.m. The day man tried to start the machine at 8:30 a.m. but could not do anything with it. The beer cooler would not freeze (that was the only part of the system in which the ammonia was turned on at the time). He had about 180 pounds condenser pressure in the high-pressure and about 60 pounds in the low-pressure cylinder.

A smaller machine was started, which answered the requirements. After the beer cooling, the big machine was started again to work on the cellars. It acted as fine as ever. What was wrong with the machine? My opinion is that the engineer allowed the steam pressure to get low and turned on too much ammonia so that the machine could not handle it all. The engineer still declares that the suction and discharge both were cool.

William L. Keil.

Philadelphia, Penn.

Provide a V-shaped brick pier or asbestos cover to protect the blowoff pipe from direct flame in the furnace. Have this pipe extra heavy with malleable elbow. Once or twice a day blow down about 2 inches of water. The best times to use the blowoff are the first thing in the morning and directly after the noon hour, on each occasion before starting up; the circulation has then been retarded for some time and the scale-forming matter has had opportunity to settle near the blowoff. Open and close the blowoff cock slowly to avoid water hammer.

Keep the boiler dry on the outside, as moisture will cause external corrosion.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Economical Fire Room Methods

In the article on the fire-room methods at the Wood worsted mill, in August 2 POWER, there are a few things that I do not understand. The first one is the statements of the gas analyses. For instance, on page 1370, it is stated that the CO_2 in one gas sample equaled 15.4 per cent., but on page 1374, where the results of tests of 12 hours' duration are recorded, in only one instance is the CO_2 as high as 10 per cent. Why is there such a difference? If those "automatic" stokers could produce such a good sample of gas for the chemist, why could they not do as well during the 12-hour test?

The temperature of the gases leaving the economizer also looks too good to be true. Of course, we must remember that induced draft is used and the temperature is taken on the cool side of the economizer. If Mr. Diman's economizer is as leaky as some of them are, then 237 degrees may be all right. But it is hardly fair to ask us to believe that 237 degrees is the temperature of the waste gases. It might easily be the temperature of the waste gases plus a lot of cold air that has leaked in. It would be absurd to try to figure the chimney losses from the temperature of the chimney gases. Mr. Diman seems to be pleased with the result, for he asks how we are going to beat it. My way to beat it would be to use forced draft and thus reduce the cold-air leakage. The results might not look as well on paper as far as the waste-gas temperature is concerned, but I believe that the coal dealers might raise a kick.

Even the temperature of the gases leaving the boiler looks too good. Take, for instance, the test in which a 200-horsepower boiler is developing 316 horsepower, the temperature of the gases leaving the boiler is 491 degrees; the steam pressure is 150 pounds; the steam temperature is 365.88 degrees; the excess air is 96 per cent. and the depth of the fire is 14 inches. There is no doubt that the 20-foot tube is a good thing, but to work a boiler at 58 per cent. over its rating and get a waste-gas temperature of only 491 degrees—well, I wish that I could do it.

I believe that Mr. Diman is right about pressure regulators. If the right kind of stokers are on the job and the right number of boilers are used, a pressure regulator is not necessary. A few years ago, after reading the stories of economy effected by the use of regulators, I got a good one but could not get any more pounds of steam per pound of coal and, as the engine did not require that the steam pressure be kept up to the limit, it was discarded.

R. McLaren.

Berlin, Ont.

Condensing and Noncondensing

In the August 2 issue on the page headed "Inquiries of General Interest" the following question and answer were given:

"Should the valves of a Corliss engine be set differently when running condensing from what they would be for running noncondensing?"

The answer given is,

"No, the valve setting should be the same for both conditions."

My opinion does not agree with this answer. It has always been my policy to use more compression in a condensing than in a noncondensing engine. That is, I adjust the valve gear so as to close the exhaust valves earlier than for noncondensing service, thus getting a larger compression curve in the diagram, though the maximum pressure produced by compression may not be as high as that produced by the later valve closing used in a noncondensing cylinder. There is no difference in the valve operation except the earlier closing of the exhaust valves. This may necessitate advancing the eccentric on the shaft; this will shorten the point of cutoff slightly if the valve gear is of the single-eccentric type. But if the valve gear is of the double-eccentric type, any changes made to the exhaust eccentric will have no effect upon the point of cutoff.

Some years ago I was shown a set of diagrams taken from a "Jerome-Wheelock" engine. They were excellent diagrams, but I remarked on the high compression. The man who took them was from the shop where the engine was built. His reply was that it was considered good practice to carry a higher compression in a condensing engine than in a noncondensing engine. I gave the point some thought and have followed this plan ever since and believe it to be correct.

Harry W. Benton.

Cleveland, O.

Oil versus Coal

On page 1394 of the August 2 issue, the statement appears, "that figures are current in California that the same amount of heat can be obtained from 2½ barrels of oil as from one ton of coal," but no figures are given to substantiate the assertion. While California oil is a very satisfactory and efficient fuel, I do not know of any such figures being current in California; in fact, it would be a very poor grade of coal of which 2000 pounds would be equivalent to only 2½ barrels of fuel oil in heating value.

California oils range in density from 10 to 35 degrees Baumé. The heating value ranges from about 17,500 to 19,000 B.t.u. per pound. The density has no relation to the heating value. I have a copy of some Bomb calorimeter tests, showing an oil of 11 degrees density as having a heating value of 18,257 B.t.u. per pound, one of 17½ degrees density having 17,805 B.t.u. per pound and one of 18½ degrees density having 18,410 B.t.u. per pound.

The heavier oils are generally used for road oiling and the lighter oils for gas making and refining. What is generally known as fuel oil runs from 13 to 18 degrees in density. Oil of 16 degrees density, containing about 18,000 B.t.u. per pound is usually considered standard and weighs about 8 pounds to the gallon, or 336 pounds to the barrel; and at 18,000 B.t.u. per pound would contain 6,048,000 B.t.u. per barrel.

Kent states that Youghiogheny coal, which is a standard grade of bituminous coal in the middle West, contains 12,941 B.t.u. per pound, which would be equal to 25,882,000 B.t.u. per ton of 2000 pounds; at which rate it would require about 4.28 barrels of oil to be equivalent to one ton of coal. Comparing Kansas coal, containing 10,506 B.t.u. per pound, with an oil containing 17,800 B.t.u. per pound, it would require 3½ barrels of oil to produce the same number of heat units as one ton of coal.

In burning oil it is considered very good work, if the steam used for atomizing the fuel does not exceed 3½ per cent. of the steam heated by the oil, to say nothing of the amount that is generally used for operating the fuel-oil pumps. This, of course, should be allowed for when the commercial amount of the heat obtained from the oil is considered. In practice the actual evaporation of 13 pounds of water per pound of oil under ordinary running conditions is exceedingly good, and somewhat better than the average, and about on a par with the tests cited on page 1374 of the same issue, which give the average evaporation of 8.89 pounds of water per pound of coal. Comparing this with an actual evaporation of 13 pounds per pound of oil, it would still require 4.07 barrels of oil to equal one ton of coal.

On page 1393 of the same issue, under the title of "Oil Fuel in the British Navy," the statement appears that 10 tons of oil were equal to 13 tons of the best Welsh coal. I presume this refers to the long ton of 2240 pounds, which, compared to California oil of the same weight and heat value as just mentioned, would require 5⅛ barrels of California oil containing 18,000 B.t.u. to equal one long ton of coal, or 4.57 barrels to equal one ton of 2000 pounds.

There is no coal burned in this section for generating steam, and it is difficult to give any exact figures as to comparative results, but in making comparisons, the heating value of both the coal and oil should be given. In a rough way, however, it is generally considered that approximately 4 barrels of oil are equal to one ton of coal. Oil has many advantages as fuel, which must be considered separately from the heating value.

G. A. Reighard.

Los Angeles, Cal.

Reading the Technical Paper

The editorial in the August 2 issue of Power is opportune; it is appropriate for the experienced engineer and the layman; for the student, for the apprentice, for the engineer, for the chief and for the proprietor, the tenet "Read your scientific paper in a scientific manner" should be stored in the mine of memory. The rule is applicable to any branch of engineering and is the corner stone in perfecting a broad knowledge along such lines.

Apropos of this doctrine should be the disposition to compile one's own handbook, to make intelligent extracts which pertain to one's work. The knowledge of what has been done by practical men, gleaned from such columns as those offered by Power, means economy in labor; items and kinks may not prove useful until months after perusal and then memory does not always hold in good stead. Using the pencil in "a scientific manner" will evolve a handbook of reference such as cannot be purchased at any price; its value to its owner and compiler lies in knowing where to find what is wanted at the right time. Thus, such a work should be properly and sufficiently indexed as it progresses, pages being left at the front or back of the book for the purpose. The notebook should be of a size easily carried on the person and of a volume of 300 to 400 pages.

Covering only fifty-two issues, a volume of Power becomes a large and bulky affair; it finds a place in your library and its equivalent worth for reference is indisputable. As a book, it is useless in the field. To be of ultimate value the meat must be extracted and must be placed available in convenient form and the self-compiled handbook Power, the cause and the effect, digested and noted intelligently, will prove a reference of illimitable assistance.

L. R. W. Allison.

Los Angeles, Cal.

Notes on Barometric Condensers

In the issue of August 2, Warren H. Miller contributes some notes on barometric condensers.

Several of the statements given do not conform with my experience.

Mr. Miller defines a barometric condenser as an ordinary jet condenser discharging through an ejector into a vertical tail pipe 34 feet long. This, to me, appears much too general a definition, as it would imply that there is only one kind of barometric condenser in service, which is certainly far from the truth.

On page 1401 Mr. Miller gives some figures regarding an installation of his which, evidently, he has not checked over.

He claims to have obtained a vacuum of 28.5 inches with 300 gallons of injection water per minute entering the condenser, which is presumably of the parallel-flow type, at 70 degrees and leaving the condenser at 118 degrees.

Now with this type of condenser it is seldom possible to get an injection discharge nearer than 20 degrees to the vacuum temperature. According to the figures published, Mr. Miller obtained an injection discharge temperature of about 20 degrees *above* that due to the vacuum temperature.

I figure that with the injection temperature given the vacuum obtained would hardly exceed 24.5 inches. Also, with the rise of temperature indicated, i.e., from 70 to 118 degrees, the ratio of injection water to steam would be in the neighborhood of 22 to 1. With this ratio and 300 gallons of injection water per minute, the quantity of steam handled would probably be,

$$300 \times 10 \times 60 \div 22 = 8200$$

pounds per hour.

With two engines developing 300 kilowatts each, the steam consumption per kilowatt per hour would equal 13.7 pounds, which would indicate an exceedingly efficient pair of engines.

Mr. Miller states that the fall in vacuum given was not due to air, but to the fact that the condenser was designed for 1000 horsepower on a 26-inch vacuum, and the load was far above this.

I fail to trace the overload mentioned, as the total horsepower of the four machines would only appear to be 1005, and it must be a very poor condenser that breaks down with an overload of 0.5 per cent.

On page 1402, Mr. Miller, in referring to the relative amounts of injection water required for surface and barometric condensers, states "it is more than likely that the actual figures of the former equal, if not exceed, those of the latter." This statement is incorrect. If the comparison were made between a surface condenser and a parallel-flow jet condenser, it would be approximately correct to say that the quantity of water for small condensers would be the same for both types of condensers, but with reasonably large condensers the surface type would require considerably less than the jet type.

If, again, we compared the relative quantity of water required for surface condensers and barometric countercurrent jet condensers, the difference would be still greater, but in this case in favor of the countercurrent jet condenser.

This is easily made clear when we consider that the "drop" or temperature difference between the vacuum and the injection discharge is usually about 6 degrees for the countercurrent jet condenser, 10 to 15 for the surface condenser and 20 degrees for the parallel-flow and ejector-type condenser.

It follows that the closer the injection-discharge temperature approaches the vacuum temperature, the greater will be the quantity of heat taken up by the water, and the smaller the quantity of injection water required.

The relative quantities of water required for different temperature conditions can easily be obtained from my figures published on page 918, November 30, 1909.

W. Vincent Treeby.

Goodmayes, Essex, Eng.

Packing a Feed Pump

In the August 9 issue of Power, George Kramer wants to know how to overcome his boiler-feed pump packing troubles. Five years ago, when taking charge of a small mill plant, I was told that the boiler-feed pump had to be packed every three or four days. Thinking that the pump was at fault, I opened it up and examined it, but found it to be in good shape. I carefully packed it with a good grade of hydraulic packing, only to have the pump "slipping" as badly as ever after a week's run. After using four or five of the best brands of packing with the same result, I started to look for the trouble elsewhere and found it in the feed pipe. The pump was located close to the deep-well pump in the engine room. Water flowed to the pump from an open heater in the boiler room. The water was discharged through a 1¼-inch feed pipe, which was about 70 feet long and had eight 90-degree elbows. The excessive friction in this long and crooked pipe made the load so great that the packing in the pump would not stand up under it. After moving the feed pump to the foot of the heater and coupling it up with a short pipe of the same size, I had no trouble at all with the packing.

I might also draw Mr. Kramer's attention to the fact that his feed pipe may be lined up with scale where it enters the boiler. This would have the same effect as a long feed pipe.

CHARLES FENWICK.

Duck Lake, Sask., Can.

Vibrations in Boiler and Piping

A few years ago I was troubled with vibrations in the piping and boilers, which were similar to that described by Mr. Davis in the August 9 issue.

In my case the vibration became more violent as the load on the engine increased, until the windows in the building rattled.

The vibration was noticed only when a cross-compound engine was running. This engine had cranks set 180 degrees and made 275 revolutions per minute. An examination of the valve setting showed it to be badly out of adjustment; the engine was doing most of the work in the head end of the high-pressure cylinder. The trouble ceased when the valve travel was equalized. It is very likely that Mr. Davis will find the source of the vibration to be at the engine and not at the boiler. Loose foundation nuts, or the nuts that hold the cylinder to the engine frame, will help to cause vibration.

J. W. BLAKE.

New York City.

Vacuum Cylinder Jacket

To determine whether there would be much of a decrease in the steam consumption of an engine if a vacuum were maintained in the cylinder jacket, a series of tests was conducted at the University of Wisconsin. The engine was a 50-horsepower compound machine.

An air pump maintained a vacuum in the jacket within a half inch of the barometer reading. If under these conditions an increase in economy were effected it was thought that in commercial work the vacuum could be maintained by a direct connection to the condenser. If there were an increase in economy only by the use of a higher vacuum it would certainly be impracticable for commercial use; therefore, the vacuum maintained in the tests was considered sufficient. The engine was run both simple and compound. Each of the tests was run with air in the jacket, with steam and with a vacuum. The six tests were each four hours long. The load was varied from one-quarter of the rated capacity of the engine to 25 per cent. above its rated capacity. Each load was carried for three-quarters of an hour and time was allowed between the runs for the engine to become adjusted to the new load. From the data obtained from the tests curves were drawn to show the economy of the engine under the various conditions.

In both the compound- and the simple-engine tests the greatest economy was secured when steam was used in the cylinder jackets. The economy curves of the tests during which air was used, a vacuum maintained in the jackets lay very close together. The vacuum secured slightly better economy at under- and overloads; the difference did not amount to more than 2 or 3 per cent.

Although these tests do not prove absolutely that a vacuum jacket around a steam-engine cylinder would not increase the economy, they do show that whatever effect it may have would not be great enough to be of practical importance.

RUBEN N. TRANE.

Milwaukee, Wis.

Engineers' Wages

The articles on engineers' wages that have appeared in the recent issues of POWER seem to cover the subject quite well; but, in considering this matter a very broad view is required because of the fact that in every plant the conditions are different and the payment of the operating force cannot be graded uniformly.

With many men the amount of time required to be put in is of more consequence than that of compensation. In any plant running 24 hours the matter to be disposed of is only the question of the length of the watches.

In the case of the factory plant where the usual running time is eight to ten hours, the engineer often has to run overtime and he may even be compelled to work on Sunday to do his repairing. In many instances he gets no extra pay for this work so that there is no encouragement to keep things up to the highest standard.

In addition to all this, if the engineer wishes to have a vacation or is prevented by illness from being on hand, the company deducts the amount from his salary, forgetting that by paying him they might give him such encouragement that he will become a more valuable man to them instead of a "sore head" who will leave as soon as he sees something better.

It is a curious fact that when the wages are only a small fraction of the operating expenses, business men will haggle over a difference of a few cents a day in the pay of the engineer when the burning of a ton, more or less, of coal to increase the production would hardly be thought of.

An employer is willing to lay out money to make a machine more efficient. I do not know of any better way in which to make a man more faithful than to let him know that he is considered necessary to his employers and to pay him a proportionate wage.

Many men think that because an engineer does not ask for more pay he does not want it. If they knew how some of the men feel about this they would realize that a raise gained by asking for it is considered as given grudgingly and because the employer feels that it is necessary.

Perhaps I have in mind an ideal condition of affairs but I am sure that if a more liberal policy were followed we would have better engineers.

To the employer I say, if you expect your engineer to work from 10 to 30 hours a week overtime doing piping, fixing belts and attending to numerous other jobs that rightly someone on the repair force should do, you expect too much. If you expect him to work all sorts of hours for a fixed salary, you expect too much. Please remember that you are dealing with men, not machines.

To the engineers I say that I think we should all take to heart the front-page editorial in the July 26 POWER for, having been through the same experiences myself, I can say that money is not everything. A position where everyone is pulling together is much nearer the ideal.

G. H. KIMBALL.

East Dedham, Mass.

Verdict on Canton Explosion

In reading the editorial in August 30 issue of your paper, "Verdict on Canton Explosion," I wish to say that in my opinion the only evidence that the cororner took into consideration as to the cause of the explosion was that furnished by people who were interested in covering up the real cause of the disaster. My reasons for saying this are, first, turning a stream of cold water into a red-hot boiler to cause a violent explosion of the same has been tried and proved a failure. In that case, where does Mr. Allen's evidence come in?

In all testimony heretofore given by all experts on violent boiler explosions that were caused by water hammer, it was distinctly stated that the water hammer was caused by suddenly releasing the steam pressure on a large body of heated water which contained an enormous amount of stored energy, and when the pressure was suddenly reduced the water flashed into steam and the safety appliance in the boiler would not take care of it. If this is true, how can anyone say that putting pressure on to the top of a body of heated water would cause a water hammer and cause the boiler with the lower pressure to explode? How can anyone claim that the exploded boiler was safe for a working pressure of 100 pounds when, if that boiler came under the head of the Massachusetts State boiler-inspection law, it would not be allowed to carry 60 pounds working pressure? Does it not look as if there is something wrong in Ohio in regard to the law covering boiler inspection?

What right has the coroner to throw out the evidence of the only man that knew anything about the explosion, and then take into consideration testimony that was only surmise? The evidence given by Louis C. Hughes, who swore that the exploded boiler was cut into the battery 20 minutes before the explosion occurred, was rejected. And in the face of that evidence, how can a man who holds the position of coroner "square" himself after such a decision when, if a fair test had been made it would have shown that the boiler was not safe for a working pressure of 65 pounds?

Who are the boiler-men who have written congratulatory letters to the coroner on his findings and verdict on the explosion and the real cause of it?

I hope they are not the men I know who write editorials for POWER.

MICHAEL H. HARRINGTON.

Fall River, Mass.

The verdict of the Canton, O., coroner on the cause of the explosion of the boiler at the American Tin Plate Mill last May is peculiar in that the sworn testimony of the fireman, Hughes, was thrown out in arriving at a conclusion. Hughes testified that the exploded boiler had been cut in, that the safety valve had blown and that the boiler was connected and in service when the explosion occurred. Unable to account for the affair he ventured the opinion of explosives in the coal as the cause, even as other men have in the past. In announcing his verdict the coroner sums up by deciding the explosion was due to water hammer caused by cutting in the boiler in question when the pressure was higher in the line than that in the exploded boiler. Had he reversed this by giving the exploded boiler the higher pressure his argument would have appeared more reasonable. The facts that the damaged stop valve was found two-thirds open and that the body of the unfortunate Austin was projected several hundred feet, seem to have been the main reasons for arriving at the water-hammer conclusion. Engineers, boilermakers, firemen and inspectors are interested in the matter. They live, move and have their being around steam boilers, and explosions touch them deeper than anyone else. Employees may hear of explosions and insurance companies grumblingly pay the freight, but they live by these storehouses of danger called steam boilers.

Being one of the craft, may I not examine and question the verdict of the coroner in this case? First, it seems the testimony of the fireman Hughes was disregarded solely on account of Austin's body being projected to such a distance that it was decided that he was on top of the boiler at the moment of explosion. Grant this fact, any man of experience knows how seldom the stuffing box on a stop valve on a boiler is packed. It is never thought of, as the valve is used only when cutting in and cutting out. The packing is usually burned out long before repacking is done. With a knowledge of this fact is it not probable that opening the valve caused the gland to leak and that, finding half an hour later that this leak had become worse, Austin went on top of the boiler with a wrench to screw up the gland and the explosion then occurred? Any engineer or fireman with experience in handling plants of this character will substantiate the logic of this view. Of course, the nonpractical man never had his attention called to this. That the valve was found two-thirds open proves nothing, for in practice this is the common position when not closed. Few open wide a stop valve over a boiler.

With regard to water hammer, it may be said, had the explosion been due to this cause the water would have been projected like a shot to the top of the shell and the metal would have ruptured in the solid plate from head to head along the top as in other explosions on record. That it did not do so in this explosion disproves conclusively the water-hammer theory in this case. Had this explosion been due to water hammer caused by quickly cutting the boiler in when under a pressure considerably in excess of that in the other boilers, then the energy instantaneously liberated would follow the path of outlet, the steam pipe, and the water would be carried upward, striking the shell with a force sufficient to disrupt it at the point of contact exactly as in the explosion at Swift's plant in Chicago in 1902. Water hammer in boilers of this type results in rupture at the point reached by the projected water.

What then caused the explosion? Here was a boiler about sixteen years old, constructed with one sheet on the bottom extending from head to head and a similar one on top, as described in Mr. Monnett's article. The upper sheet tore away from the lower sheet by shearing the rivets in the longitudinal seams on each side. This construction has been condemned by all experts for reasons that have been explained in POWER many times. The seams were of lap-joint double-riveted construction with the flexure common to that design. If the plate be the weaker, cracks occur. If the rivet section in shear be the weaker, then the bending action long continued will result in shearing at mid-section, and granting failure at that point it follows the remaining rivets being weaker than the net plate give way by shearing.

In view of the above the verdict of the Canton coroner may be set aside as of no value to the men operating boilers. The explosion in my opinion points out again the danger in operating boilers of this construction, together with emphasizing the weakness of the lap joint. We should have a factor of safety high enough for safety and a yearly water test of 50 per cent. in excess of the calculated pressure on the one-sheet boiler. A factor of safety of 6 is not too high, as the horizontal seams cannot be seen inside or out. No examination short of a water test with the walls removed to permit the seams being examined is of any real value on boilers of this design, and this test should be supplemented by trams over the seams and by steel tape around the circumference of the shell at several points. That about 1½ boilers explode every day in the United States is by no means creditable to those engaged in supplying us with reservoirs for stored energy. Nor should the Canton explosion blame be shifted on the shoulders of a dead man by the exceedingly peculiar views of the Ohio coroner.

JOHN DODD.

New York City.

In a recent editorial mention is made of the report of the coroner on the cause of the boiler explosion at the works of the American Sheet and Tin Plate Company, Canton, O., on the afternoon of May 17, which is a repetition of the "whitewash" so obvious in the account of the occurrence published in the issue of May 31. The coroner was assisted in arriving at his conclusions by the testimony of Francis B. Allen, Osborne Monnett and also by Edwin Whelan, whose testimony before the coroner varied widely from his remarks, as published in the Canton press, a few hours after the explosion. Mr. Allen said the explosion was probably caused by water hammer, a phenomenon liable to occur whenever a boiler was cut into a line with other carrying either a higher or lower pressure.

Mr. Allen's statements as to the probability of a water hammer may have fitted a case where the explosion occurred at the time of cutting in a boiler but are hardly fit in the case where the boiler had, as the injured fireman testified, been cut in with the others for some 20 minutes when the explosion came. He had personally attended to that matter and was sure of it.

While expressing regrets for his lack of technical education, the coroner states two impossible conditions and selects one as the cause of the explosion.

It is to be regretted that no effort is made to make the use of steam boilers comparatively safe.

Middletown, O. P. B. MARTIN.

German Municipal Stations

By F. R. Low

The people of Germany, of much of Europe in fact, seem to be able to do things for themselves to a greater extent than the "self-governed" people of America have done. The cities generally own and administer not only their water supply but their gas and electrical works, their street-car systems, etc., and are able even to provide amusement for themselves through municipal concert halls and gardens, subsidized theaters, etc., without any of the dire consequences symbolized by "bureaucracy" and "paternalism" with which we are threatened when such self service is proposed.

Frankfort-on-the-Main is one of the richest as well as one of the largest, territorially, of the European cities. Years ago it loaned the king a sum of money, secured by an extensive wood contiguous to the city, and the debt has never been redeemed, so the city still has the broad expanse of territory, and an appearance of being well to do. The municipal central station furnishes current for the street and commercial lighting and for the operation of the tramways. A visit to it is a formal affair to be preceded by a call at the bureau and the making of an appointment.

The effect is quite different from an informal dropping in. One is expected. Massive gates are thrown open, the uniformed *portier* is ready with every attention and an engineer is waiting to conduct one about and exhibit and explain the works.

The first impression is that of elegance and permanency. The buildings are impressive in size and architecture, the floors are tiled, the walls are wainscoted with colored ceramics, everything is clean, quiet and orderly. One would no more think of throwing anything upon that spotless floor, or placing his hand upon the polished bright work than he would of throwing a cigar stub upon a parlor carpet or putting his shoes upon a satin-brocade lounge. We have some stations with tiled floors and walls, and we use some metal polish and elbow grease, but I never saw a station where a bit of disorder would be so apparent, where a leak would be such a desecration as here. I suppose that all this elegance means high first cost; it may be expensive to maintain this "dress parade," but it must conduce to efficiency, effectiveness, longevity; and the moral effect and enjoyment of the nicer surroundings and conditions ought to be worth the difference in the cost.

The engine room contains five turbines of the Parsons type built by Brown-Boveri, each of 3500 kilowatts capacity and four 1000-kilowatt Sulzer engines. Surface condensers with motor-driven circulating and dry-vacuum pumps are used, the water coming from the river Main. The steam pressure was 154 pounds gage with superheat to 575 or 600 degrees Fahrenheit and the vacuum 28 to 29.25 inches of mercury. The incoming water is 64 degrees, the outgoing 73 and the hotwell 77. Under these conditions the turbines use about 12.76 pounds of steam per kilowatt-hour.

There are four types of boilers in the station, having a total heating surface of 107,600 square feet for an aggregate of 30,000 horsepower of engines. Narrow chain grates are used, two to a boiler. There is one chimney 246 feet high and 10.5 feet inside diameter at the top and two others 164 feet in hight and 9.2 feet inside diameter at the outlet. Each boiler has its own economizer with 2368 square feet of heating surface each, heating the water from 77 degrees to 194 degrees Fahrenheit. Centrifugal feed pumps are used, each capable of handling 2472 cubic feet per hour against the boiler pressure when running at 2600 revolutions per minute.

Ados and Pintsch CO_2 recorders are used, the records showing from 10 to 12 per cent. of carbon dioxide.

In 1909, the station turned out 34,000,000 kilowatt-hours at a station cost of 1.32 cents per kilowatt-hour, to which must be added 0.6 cent for standing charges. Coal costs about $4 per ton. The average attendant is paid $1.32 for a nine-hour day.

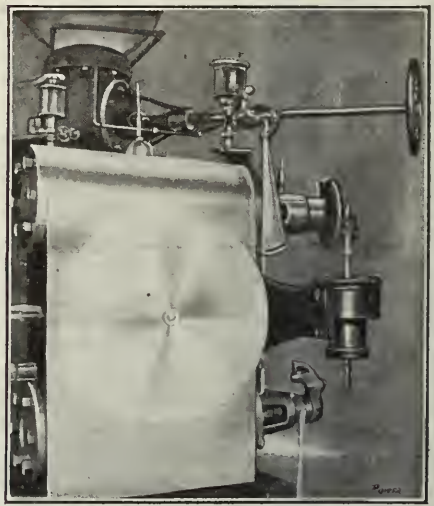
At Cologne there is even less evidence of industrialism in the exterior appearance and approach to the combined water works and electric stations than at Frankfort. If it were not for the chimneys one would never imagine that the handsome buildings standing in a park filled with flowers, trees and shrubbery, surrounded with an ornamental iron fence and entered through an impressive portal, were devoted to pumping water and generating "juice." No adequate idea of the installation can be conveyed without photographs, and these with a description of the station by one of the engineers we hope to be able to present to our readers later. There are two electric stations, one with four engine-driven units of 1000 kilowatts each and one turbine of 2000 kilowatts, and in another building three turbine units of 3000 kilowatts each. The pressure is 160 pounds gage; the temperature is 660 degrees Fahrenheit at the boilers and 540 at the engines. They use water from the city mains for condensing, the city water works alongside being thus their circulating pump; and, using a Brown-Boveri dry-air pump, get a 95 per cent. vacuum. The circulating water is thrown away after passing through the surface condensers, although it must be collected from wells. There are four chimneys, each 246 feet in hight and 13 feet in outside diameter at the top.

The pumping station is a high-vaulted room with gothic windows and stained glass like a cathedral. There are five large horizontal pumps made by Sulzer Brothers.

In neither of these stations did I detect a leak and neither would own to any trouble with the piping, notwithstanding the use of superheated steam. They never had had any trouble. They used steel pipe expanded into flanges, avoided the use of fittings wherever possible by using bends instead, and used cast steel when fittings were required as well as for valve bodies.

Position of Throttle

The illustration shows the arrangement of the throttle valve on a Corliss engine found running in a small steam plant at Bristol, Va. The throttle is so set that the stem comes a trifle inside the corner of the cylinder on the valve-gear side. The engine has two eccentrics, and the advantage of the valve-stem extension is at once apparent to the man who is familiar with this type of engine.



SHOWING POSITION OF THROTTLE

The throttle is conveniently placed for the engineer to operate in stopping the engine at the proper point for starting. It is certainly easier to handle the valve gear with a throttle of this design than if it were of the short type set with the valve stem midway between the valves, which would necessitate the engineer reaching over to catch the throttle wheel in order to get steam into the cylinder, either to push the crank over the center or prevent it from passing the quarter.

Some Interesting Gasket Kinks

By R. O. Richards

The author tells how to "lead" a gasket, how to cure curling and gives a number of useful kinks concerning gasket troubles.

In Fig. 1 is shown an ordinary way of connecting the three drums of an Aultman & Taylor boiler. The arrangement represents such an inflexible construction that when one of the gaskets *A* or *B* blows out, it is a most difficult matter thoroughly to clean the flanges without disturbing some of the other gaskets. The quickest, and in the end the cheapest, way to renew the packing in this case is to break the joint *C* and remove the elbow. If the gasket at *C* has been thoroughly graphited, only the packing of the affected joint need be renewed. The best way to "lead" a gasket is to mix the graphite with a little water, and not with oil, as often oil besides attacking the rubber will, when dried, cause the gasket to stick.

If a joint such as that shown at *D* leaks, it will be necessary to slacken on the bolts of the flanges at *A, B, E* and *F* sufficiently to allow for prying open the joint *D* with thin wedges. The task of removing the old packing, owing to the limited opening, is arduous and vexing. An effective tool for such work is shown in Fig. 2. Here *A* is a flat piece of steel 1/8 of an inch thick. The entering edge is ground to a one-sided sharp edge. By manipulating the nuts *B*, *A* is forced between the flanges, and removes the old gasket more satisfactorily than would an old saw.

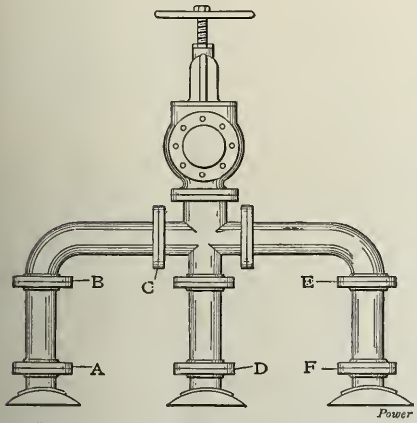


Fig. 1. Rigid Drum Connection

It is often difficult to be certain that the flanges have been thoroughly cleaned. If a piece of brown paper, lightly graphited, is inserted in the joint, and then the flanges are allowed to pinch it, the high spots are easily located. If the old gasket has any asbestos in its make-up, to soak it with water will often expedite its removal. When inserting the new packing a difficulty often met with is caused by the curling of the gasket which insists on dropping down the pipe opening rather than remaining between the flanges. The common remedy is to insert a sheet of tin in the joint and slide the gasket over it, but often the opening is so small that the tin causes the gasket to bind. A good method is to cut the gasket with two fairly long ears, and cement, with a little shellac, a portion of one ear to a large piece of strong brown paper. The gasket can then be coaxed through with but little trouble. When in place the cemented part of the ear is cut off, and the paper removed. In confined places, however, the use of metallic gaskets will save much time and annoyance, and when trouble arises they are very easily removed. When flanges are covered, often a joint will leak considerably before it will show through the nonconducting material to any extent and the result is scored flange surfaces. If this proves serious a length of pipe should be removed and a metallic cement applied to the grooves. When replacing the pipe care should be taken to prevent the furrows from again falling in line with one another. A gasket which proved successful in a troublesome joint that failed to line up properly is shown in Fig. 3. A ring of holes was punched in an old, flattened-out copper gasket, then a three-sixteenths inch spool packing composed of several strands of good, strong material was woven through the holes, so as to make a continuous line. As the weaving progressed, the packing was gradually thinned down by cutting off some of the strands, so that diametrically opposite to where the weaving started, the packing is less than 1/8 of an inch in thickness. A ring of the same material, gradually thinned down in the same manner, is then cemented with shellac close to and on the inner side of the woven packing. The whole is now carefully beaten down so that it will enter the joint; the thinnest part, of course, was placed on the side where the flanges are nearest together. As the packing encircles both sides of the copper sheet, it allows of considerable lack of alinement in the flanges. The gasket must be "followed up" for a few days until the packing sets.

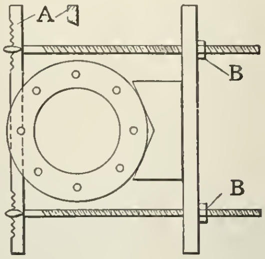


Fig. 2

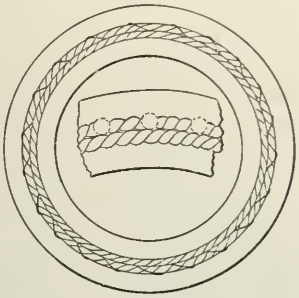


Fig. 3

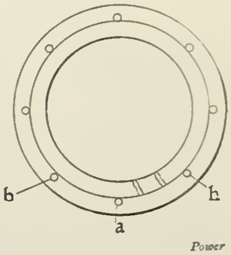


Fig. 4

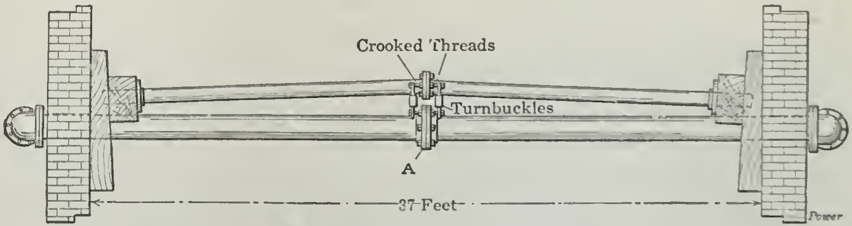


Fig. 5. Supporting a Run of Pipe

A gasket containing much rubber in its composition should be tightened up very evenly. To squeeze hard on the bolts the first time round will result in wrinkling the packing. This is graphically shown in Fig. 4. If bolt *a* is tightened hard, followed by the tightening of the others in consecutive order, the rubber will stretch and wrinkle between bolts *h* and *a*.

In some plants considerable trouble is experienced with gaskets from lack of proper support for the pipes. The following two examples are given to illustrate:

At a certain mill it is necessary to pass a 4-inch pipe carrying steam at 110 pounds pressure over a 37-foot roadway between two low buildings. No support could be placed under the pipe, and in consequence considerable difficulty was experienced in keeping the gasket *A*, Fig. 5, tight. The hight of the building does not allow the use of braces or guy ropes, and the pipe is at the lowest level allowable.

The problem was solved by taking two lengths of 2-inch pipe and cutting long crooked threads on one end of each. Securing flanges to these, the pipe was placed above the steam main, and set to rest on the brackets as shown in Fig. 5. The two single flanges at each end were bolted loosely but securely to these, allowing ample travel for contraction during the winter months. Four small turnbuckles connect two bolts of the middle 4-inch flanges, to two bolts of the supporting 2-inch flanges. In winter a couple of layers of tar paper cover the two pipes in such a manner that what little heat escapes through the covering of the 4-inch main will keep the support pipe warm, thereby preventing undue contraction in zero weather.

In another mill where a Climax boiler is in service, considerable difficulty was experienced in keeping the gaskets *A* and *B*, Fig. 6, from leaking. It was surmised that the piping was not sufficiently flexible, and that a long sweep should have been used in place of the 45-degree piece, to allow the boiler to expand upward, without undue strain on the flanges *A* and *B*. As it would entail considerable expense to alter the piping, it was decided to try a movable support, so that the boiler expansion could be thrown more on the long pipe leading to the engine. A 2-inch pipe in three pieces was set solidly in an upright position on the boiler-room floor. A slot was cut in the upper section for the two levers *C* to enter under the loose forked head *D*. Attached to the levers by the wrought-iron rods *G* are the weights *E* which, through the medium of the levers that have their fulcra at *F*, have been calculated to balance the weight of the steam main above them. Thus, as the boiler expands and raises the steam pipe, the forked head *D* will rise with it, and in a large measure maintain a uniform strain on the flanges at all times. The flange beneath the weights was thought necessary as a precaution against accident by the stripping of the rods *G*. Use was made of it by bolting to it hooks for the very convenient hanging up of the fire tools.

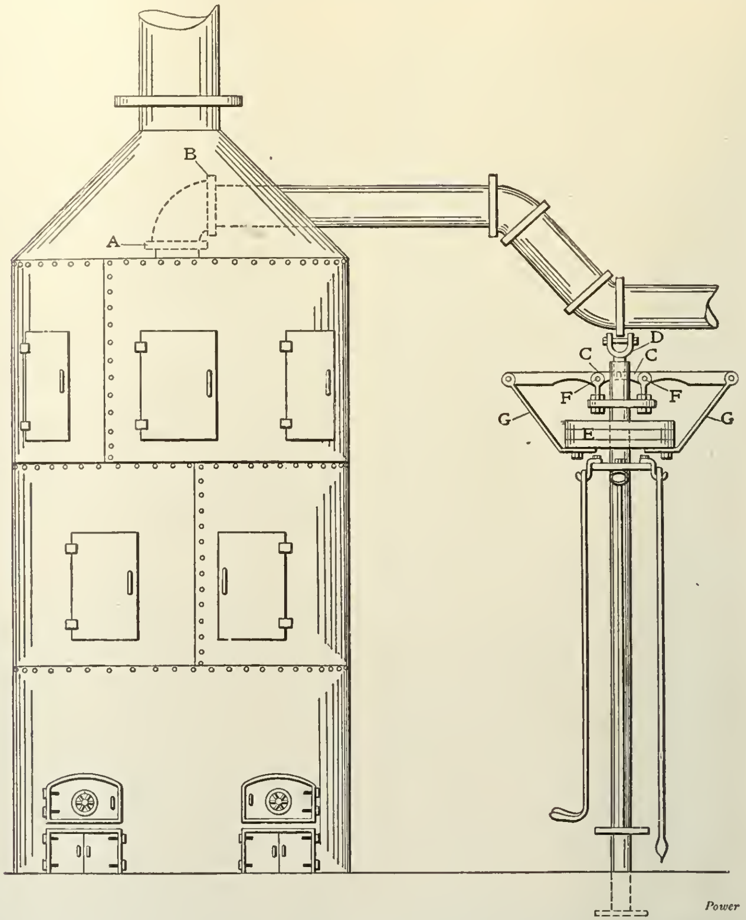
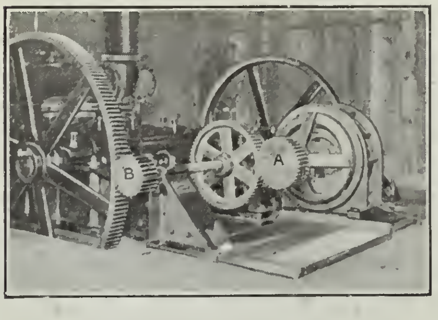


Fig. 6. Leaky Gaskets in Vertical Boiler

Rawhide Pinions Stop Vibration

Rawhide pinions are usually considered merely as a means of stopping noise on medium- and high-speed drives and by far the largest percentage of them are used for that purpose. It is a matter of engineering experience, however, that rawhide is beneficial on gear drives where there is vibration from irregular load, etc.



Location of Noiseless Pinions

This is well illustrated in the instance of the 1750-gallon quintuplex pump in the power plant of the Parral Power and Reduction Company, Parral, Chihuahua, Mexico. This pump, as shown in the illustration, is geared through double reduction to an induction motor and furnishes the circulating water for the condenser.

Originally the pump was driven entirely by cut-steel pinions and cast-iron gears. The pump was of so light a pattern that there was considerable spring in both shaft and frame. The result was constant annoyance from broken gear teeth and in addition to gear renewals, there was quite an expense due to the power loss of running the engines noncondensing while pump repairs were being made. The chief engineer had about decided to abandon the pump and replace it with another of different design when someone suggested the use of rawhide pinions. Accordingly, four "New Process" pinions were substituted for the four steel pinions. Two of these are shown in the photograph, one at *A* on the motor and the other at *B* meshing with the big gear at the left. The other two are on the other side of the motor and placed symmetrically to those shown. These rawhide pinions were put into service over two years ago and there has not been a moment's trouble with the gearing since. In fact, the pump has been out of service only long enough to repack the plungers and replace the rubber valves.

Rawhide where properly cured and machined makes up into a gear or pinion that is much the same as metal, except that it has no metallic ring and is more elastic. This elasticity is highly advantageous on motor and other drives in that it absorbs the shock as large gear teeth come into contact, and it cushions the irregularity of load due to the reciprocating movement of parts in machine tools or geared power-plant apparatus.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for August, 1910, *was* 170,000.

September 6........... 35,000
September 13........... 33,000
September 20........... 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

Engineering Calculations

Accuracy with some people is instinctive—they practice it in every thought and action; with others it has to be cultivated through severe mental discipline, and they have to be continually on their guard against carelessness. No one will deny that accuracy is a desirable attribute, yet there is always the chance of the exceedingly accurate person becoming absorbed in the details without giving due attention to the main points. This is especially true with engineering calculations; here it is almost as bad to use unnecessary refinements as it is to carelessly use a number of crude approximations. From this it should not be inferred that precision may be neglected in the purely mathematical operations; it means that forethought and descrimination are required in determining the necessary degree of accuracy. This involves an understanding of the relation which the calculation bears to the problem under consideration, in order that the desired result may be obtained without unnecessary labor.

Many problems involve assumptions or factors that cannot be accurately measured and it would be absurd to carry out the calculations to the third or fourth decimal place. For instance, in computing the load that a certain member would carry, the assumption is made that the point of rupture of that particular material is 50,000 pounds per square inch. Now, this material may fail at 45,000 pounds per square inch or it may withstand 55,000 pounds per square inch; hence, the futility of carrying the computations to a high degree of refinement.

If it is desired to find the size of pipe through which a certain quantity of steam would flow under given conditions, it is sufficiently accurate to use 3 1/7 instead of 3.1416 as the value of π in finding the cross-sectional area of the pipe, for the reason that a number of indefinite factors, such as the drop in pressure due to condensation, friction, etc., enter into the problem and make a really accurate predetermination of results impossible.

Again, in figuring the indicated horsepower of a steam engine from an indicator diagram, if a slide rule be used the result will be about as near the true horsepower as would be the case if the problem were figured out by long hand. For the error in finding the area of the indicator diagram and the fact that the steam pressure in the indicator may not exactly represent that in the cylinder, make the result, at the best, only approximate.

On the other hand, there are certain engineering calculations which require a high degree of accuracy. Consider, for instance, the problem of laying out a tunnel under a river. The civil engineer must use the utmost accuracy in establishing his lines in order that the construction, begun simultaneously from both sides of the river, may meet at the center.

The advantage of being able to size up a problem at a glance with the exercise of proper judgment as to the degree of accuracy required, increases with experience. In the computing departments of large firms a vast saving in time and labor may be effected by the intelligent use of approximations.

The Sacrosanct Boiler Owner

One of the proud privileges of the American citizen is that of destroying life and property without let or hindrance, provided certain lines of procedure are followed. He may only to a limited extent kill with adulterated and poisoned food stuffs; liquid exhilarants and soothing beverages may contain only prescribed percentages of poison, and children of all ages may not be indiscriminately ground into profits. But almost anywhere in this country, with few exceptions, a steam boiler with its potentiality for the destruction of life, maiming of limb spreading of fire and what not, may be installed and operated without question from any source whatever. If it were not so tragic in its consequences it would be ridiculous.

It is not necessary to call attention to the alarming rate at which the destruction of life and property is increasing as the result of boiler explosions in the United States to show that compulsory and intelligent inspection of all steam boilers, used for any purposes whatever, is fundamentally requisite for common safety.

If a building is to be erected or repaired, passers-by on the sidewalk are protected against danger from falling material by an arcade of heavy planking. But under the protected sidewalk or in the basement of the building there may be installed without question any number of steam boilers, even though they be known by the builder, the dealer and anyone else familiar with the subject, to be unsafe for the purposes intended, and the passer-by and the worker in the building are helpless while the owner, who is seldom or never on the premises and is the one person on whom rests all the moral responsibility for disaster, is hedged about by the protection of the custom that allows the unrestricted installation of such apparatus. When an explosion occurs, as a salve to the conscience of such owners, if they have any, the coroner usually renders an innocuous verdict which is arrived at in some cases by frankly ignoring evidence of criminal negligence which is submitted.

How long is this condition of affairs to continue?

The Inaccessible Boiler

It is an unfortunate circumstance that stationary steam boilers are so frequently placed in positions difficult of access, making it a hot and disagreeable task to properly inspect them when in operation. The top of the boiler, in this class of installation, is a place to be avoided except when one of them is to be cut in or out, and then on account of the intense heat no more time than necessary is spent in the operation. Leaking seams and flanges are sometimes allowed to blow for long periods in hot, inaccessible places over the boilers, as repairs are put off as long as possible on account of the disagreeable work the job entails.

In the comparatively cool and comfortable engine room the engineer can examine his engine, indicate it and at his leisure make adjustments, all of which are important from an economical standpoint. These adjustments, however, are for economy only, while the examination of the boiler is for safety, which is of much greater importance. Boiler seams are frequently embedded in the brickwork of the setting so as to be totally inaccessible for examination, while other parts of the shell are covered with brick or nonconducting material and left for long periods without inspection. Conditions surrounding the boilers should, as far as possible, be made comfortable and all parts should be easy of access to the end that examinations, whenever made, shall be so thorough as to lessen materially the chances for disaster.

Following the scalding to death of one man and the injuring of another while rolling tubes in the firebox of a locomotive boiler at the Chicago & Northwestern shops, the chief boiler inspector of the city of Chicago addressed a letter of protest to the officials of the company, calling it a crime and a cruelty to order any human being into a firebox to repair a boiler under pressure or containing water at or above the scalding point. This view is to be commended not only from the humanitarian standpoint but from that of cold-blooded self-interest on the part of the owners.

Comfortable workmen will do good work, and good work is what is required when boilers are under consideration. There is no apparatus about which the engineer should be more particular, and on the other hand, there is no apparatus which suffers the danger of being slighted more than the hot, crowded, inaccessible boiler. Whoever is responsible for the layout of a boiler room should by all means bear these facts in mind and as far as possible make accessibility for inspection and convenience of operation the leading features of the design.

Inches or Pounds?

One of the incongruities of modern engineering practice is the use of one unit of measurement in discussing the condition of steam or other media when dealing with pressures above atmospheric and the use of a different unit when the pressure is less than atmospheric. That is, we talk of steam at a hundred "pounds pressure" but speak of a vacuum of so many "inches of mercury." This is all right, of course, so far as vacuum in the abstract is concerned; when one is speaking merely of a vacuum it is just as convenient to measure it in inches of mercury, perhaps, as any other way. But in view of the intimate and indispensable relationship between vacua and steam, in power-plant engineering, it would seem more consistent to express degrees of vacuum in terms of pressure—that is, to stick to one kind of unit throughout the scale of pressures. "Twenty-eight inches of mercury" means absolutely nothing without the commonly-accepted interpretation, which involves the application of a conversion factor and an assumption of atmospheric pressure; but "one pound absolute pressure" means precisely what it says, "per square inch" being commonly understood. Furthermore, no inconvenience of speech is entailed. It is just as easy to say "one and a half pounds" as it is to say "twenty-seven inches," and the former is not only more descriptive but absolutely devoid of ambiguity.

By all means let us measure condenser pressures in the same units as boiler pressures. "Inches of mercury" have no place in a steam plant nowadays.

Do It Right or Quit

Did you ever hire a laborer to dig post holes eighteen inches in diameter and thirty deep, and obtain the pleasing result of an ill-assorted lot of cavities ranging from twelve to sixteen inches across the top, eight to twelve inches across the bottom and eighteen and twenty-four inches deep?

Or did you ever have a room papered by a man who manifested a cheerful disregard for pattern-matching at the seams and an optimistic faith in the ability of the old paper to hang on to the wall and sustain the weight of the new paper?

If you ever had either of these experiences or any others of a similar kind, you can readily imagine how your employer feels when you ignore the pounding of a loose box, or scrape a pair of brasses to a "near-enough" fit, or carry a fire bed two feet thick in spots, or try to stop a leaky stuffing box by jamming the gland in instead of repacking the box, or do any of the other hundreds of sloppy, shirking things that indicate either a half-baked mentality or a dishonest willingness to do as little as you can without getting "fired."

If you are too lazy to do things as they ought to be done, or you don't think you are getting paid enough to justify that much trouble, get out and let a real man have your job—you will never make anything of it or of yourself.

Smokeless Combustion

Smokeless combustion is a subject that has recently been given a good deal of attention and as a consequence many improvements have been made in the boiler furnace.

Engineers are no longer satisfied with the construction of side walls and bridgewalls and the setting of grates in the most convenient manner; they are realizing more and more that boiler furnaces should be built so that combustion will be complete before the gases strike the heating surface of the boiler.

Bituminous coal of average grades can be burned without smoke, providing proper equipment and firemen who will handle the fires and apparatus intelligently are employed. One thing is certain, a good stoker or furnace, improperly set, will not do as good work as an inferior stoker or furnace well set.

It is the opinion of many engineers that bituminous coal cannot be burned without smoke when hand-fired, because the fireman is altogether too variable a factor. This may be true in many small plants and perhaps is in some large ones too, but if proper supervision is maintained over the fire-room force, satisfactory results may be attained.

In small steam plants the problem of smoke prevention may be solved either by hiring capable and careful firemen or by installing suitable mechanical stokers. The best results can be obtained from hand firing only when the charges of fuel are frequent and the amount small. The success of the mechanical furnace is largely due to the fact that the fuel is fed continuously, and the nearer hand firing can be made to approach this condition, the better will be the results.

Firemen who are obliged to fire coals containing large percentages of heavy hydrocarbons have greater difficulties to overcome than if coals of other grades were burned, but with a well designed furnace burning an ordinary grade of coal there is small excuse for the fireman to offer if he is guilty of maintaining a smoky stack.

Inquiries of General Interest

Steam Required to Heat Water

How many pounds of steam at 64 pounds absolute pressure per square inch are required to raise the temperature of 300 pounds of water from 40 to 130 degrees Fahrenheit, the water and steam being mixed together?

L. J. K.

One pound of steam at 64 pounds absolute pressure per square inch contains 1178.2 heat units above 32 degrees, of which it will give up 1080.31 in cooling to water at 130 degrees. It will take 89.84 heat units to raise one pound of water from a temperature of 40 degrees to 130.

To raise 300 pounds of water through the same range will require

$$300 \times 89.84 = 26{,}952$$

heat units and will take

$$\frac{26{,}952}{1080.31} = 24.939$$

pounds of steam at 64 pounds absolute pressure per square inch.

Superheated Steam

Why does not superheated steam have a temperature due to its pressure, and does it not expand in superheating and thus increase its pressure?

F. L. J.

By definition superheated steam is steam having a temperature higher than that due to its pressure. In superheating it expands to a slight degree if the pressure is constant but if the volume is constant superheating will increase the pressure. In ordinary practice the superheat increases the volume so slightly that the rise in pressure is not appreciable.

Engineers' License Laws

What States have engineers' license laws, and what is required to obtain a license?

W. F. S.

Massachusetts, Minnesota, Montana, Ohio, Pennsylvania and Tennessee are the only States having laws requiring the examination and licensing of engineers and firemen. But several States have laws which permit municipalities to pass license ordinances. In Baltimore, Md.; Buffalo, N. Y.; Chicago, Ill.; Denver, Colo.; Detroit, Mich.; Goshen, Ind.; Jersey City, N. J.; Kansas City, Mo.; Lincoln, Neb.; Los Angeles, Cal.; Memphis, Tenn.; Mobile, Ala.; New Haven, Conn.; New York, N. Y.; Niagara Falls, N. Y.; Omaha, Neb.; Peoria, Ill.; Philadelphia, Penn.; Rochester, N. Y.; Santa Barbara, Cal.; St. Joseph, Mo.; St. Louis, Mo.; Sioux City, Ia.; Spokane, Wash.; Terre Haute, Ind.; Yonkers, N. Y.; Fulton County, Ga., and in the District of Columbia engineers are required to have licenses. To obtain a license as engineer, one must show that he has the necessary experience and intelligence to enable him to operate steam boilers and engines with safety.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Link Motion Valve Setting

I am running a reversing engine with the common link motion and would like to know how to set the valve if it should become necessary.

B. M. C.

Put the reversing lever in one end notch and see that the valve stem and eccentric rod are of the right length, then put it in the other end notch and see that the other eccentric rod is of the right length. Then with the engine on the center set the valve the same as in the case of a single eccentric, trying with the lever in full gear both backward and forward.

Overtravel of Valve

What is meant by the term "overtravel?"

U. C. G.

Overtravel is the amount of movement of the valve above that necessary to give full port opening. To give full port opening a slide valve must move twice the width of one port and twice the lap. If the valve travel exceeds this, it will overtravel.

Leather Belting

Should the grain or flesh side of a leather belt run next to the pulley, and why?

P. C.

Leather belts should be run with the grain side to the face of the pulley. This side is harder and smoother than the other and makes a better contact with the face of the pulley. This side is not as strong as the other and the wear, if any, should be on the weaker side. Belts with the grain side next to the pulley will transmit one-third more power, with the same tension, than those wit hthe flesh side next to the pulley.

Strength of Butt Joint

Where is a double-strapped butt joint the weakest?

A. S. M.

Usually along the line of the outer row of rivets.

Per Cent. of Condensation in Feed Water

Using an open heater and exhaust steam for heating feed water, what percentage, by weight, of the feed water will be due to the condensation of steam, assuming the original temperature of the water to be 80 degrees Fahrenheit and the final temperature 212 degrees?

E. L. W.

One pound of steam at atmospheric pressure contains 1150.4 heat units. In condensing to water at 212 degrees, it will give up

$$1150.4 - 180 = 970.4$$

heat units. In raising from 80 to 212 degrees, one pound of water will absorb

$$180 - 48.3 = 131.7$$

heat units. As each pound of steam will give up 970.4 heat units in condensing to water at 212 degrees, it follows that one pound of steam at 212 degrees will heat

$$970.4 \div 131.7 = 7.36$$

pounds of water from 80 to 212 degrees, making 8.36 pounds in all. As one pound of this is condensed steam the percentage is

$$1 \div 8.36 = 0.119, \text{ or } 11.9 \text{ per cent.}$$

Pitch of Rivets

In a boiler the plate is 3/8 inch, the diameter of the rivets is 13/16 inch, and the efficiency of the seam is 71.7 per cent. What is the pitch of the rivets?

C. G. F.

In a correctly designed seam its strength is that of the uncut plate in the outer row of rivets. This strength is expressed by the equation

$$Efficiency = \frac{Pitch - Rivet\ diameter}{Pitch}$$

Substituting the values given in the example the formula reads

$$\frac{p - 0.8125}{p} = 0.717$$

$$p = 2.84 \text{ inches.}$$

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

The Ray Automatic Feed Water Purifying System

This system consists of a settling tank arranged on the outside of the boiler setting at the rear. This tank is connected to the boiler and feed pump, as shown in Fig. 1. Details of the tank are shown in Fig. 2, where *A* represents a steel tank; *B* a pipe entering the boiler shell, if it is a return-tubular boiler, or the drum, if it is a water-tube boiler, just below the water level; *C* the feed-water pipe with a nozzled end which forces the feed water down into the tank, and *D* a drain pipe passing water from the machine to the boiler, usually through the boiler blowoff.

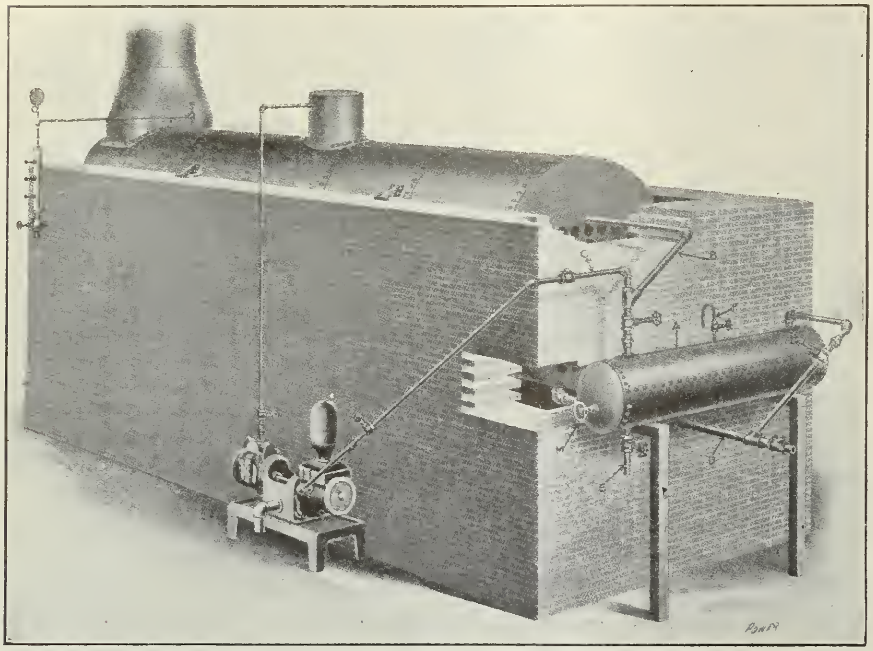


Fig. 1. Showing Settling Tank and Connections

The pipe *E* is the blowoff for the tank; *F* is an oil separator, air cock and water-test valve; *G* is a filter and strainer through which all water passes to the boiler; *K* is a copper elbow passing water to the oval end of the tank, creating an eddy and *M* is a scraper by means of which the tank is cleaned.

The action of the system is as follows: The rapid circulation of the water in the boiler sets up a current through the pipe *B* to the settling tank *A*, and back to the boiler through the pipe *D*. The velocity of the water which enters the Y through the pipe *C* acts in the manner of an injector and draws an increased amount of water through *B* into the tank *A*.

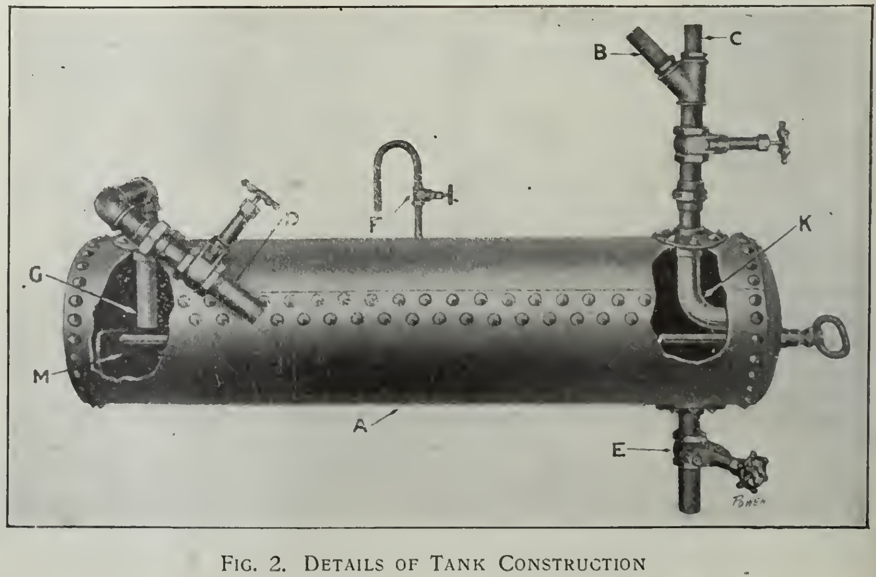


Fig. 2. Details of Tank Construction

The settling tank is so proportioned that the water passes through it slowly, and all matter held in suspension that is heavier than water, during the passage from the pipe *B* to the pipe *D* settles to the bottom of the tank, it is claimed.

Oil and the lighter particles rise to the surface of the water, where, due to the fact that the pipe *D* projects into the tank, it is held till drawn off through the valve *F*.

To remove sediment from the tank, the valves in both inlet and outlet pipes should be closed. The valve *F* should then be opened until the oil is blown off, if there is any, and then closed. The valve *E* is then opened and the greater part of the sediment is blown off and the remainder is drawn slowly toward the blowoff, by means of the scraper *M*, until it is all blown out. In refilling the tank the valve on the pipe *D* may be left closed and the valve on the pipe *B* opened, till the tank is filled, thus taking the water from the top of the boiler.

By taking a volume of hot water from the boiler just above the flues through the pipe *B*, a continuous surface blowoff without any waste of hot water is obtained, as it returns to the boiler with the feed water through the pipe *D*.

By reason of this hot water from the boiler mixing with the feed water from the pipe *C*, it is claimed that the reagent, if any is used, is instantly taken up and the temperature of the feed water is raised sufficiently high to drive off the carbonic acid gas, thereby liberating and depositing in the tank all the carbonates as sediment.

The operation of the system on all

types of boilers is substantially the same. The outflow pipe *B* is always connected to that portion of the boiler just below the water line, towards which the water on the surface flows. In the case of such boilers as the Stirling, where the water in the upper drum does not flow toward either end, perforated pipes are placed just below the water level in each drum, and connected to the pipe *B* extending the full length of the drum. The return pipe is usually connected to the boiler blowoff, especially in the case of return-tubular boilers.

This system is made by the Ray Steam Specialty Company, Kansas City, Mo.

The Wiechmann Turbine Tube Cleaner

The accompanying illustrations show the characteristics of a new line of turbine tube cleaners which are about to be placed on the market by the Clyde Machine Works Company, Thirty-ninth and Union avenues, Chicago, Ill. The styles shown are adapted for both water-tube and fire-tube boilers and different styles of heads are provided for scales of different character.

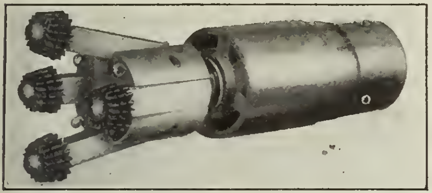


Fig. 1. Wiechmann Turbine Tube Cleaner

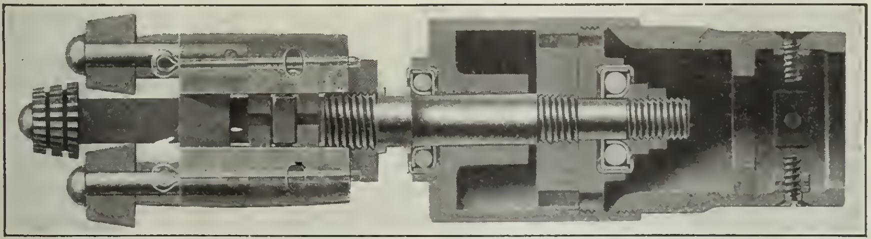


Fig. 2. Sectional View of Turbine Tube Cleaner

Fig. 1 shows an exterior view of the standard cleaner for water tubes. the feature of which is the simplicity of the cutter head and the ease with which the elements may be removed from the machine and replaced or repaired when necessary. All parts of the head are made of tool steel, tempered to withstand the hard usage, and, as far as possible, all wearing parts have been eliminated.

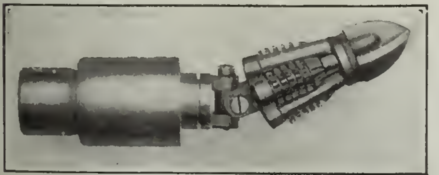


Fig. 3. Devil Head Tube Cleaner

Fig. 2 is a sectional view of the cleaner, showing its simplicity of construction. The cutter's arms are removed from the head by taking out a small brass cotter pin which holds the shaft on which the arm revolves. The cutter wheels may also be removed by withdrawing the key holding the cotter pin in position. There is no possibility of this key working loose and falling out on account of both sides of the main body of the cutter head working in close machine fit with the pin, as shown in Fig. 2. Another important feature of the head is the manner in which the cutter arms are interlocked, thus operating in unison, due to a projection on the two longer arms engaging a slot on the two shorter arms.

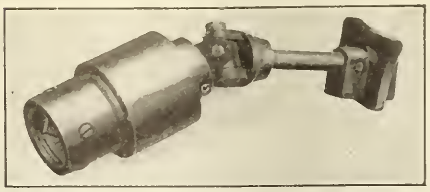


Fig. 4. Cleaner for Fire-tube Boiler

The arms are controlled by suitable stops in such a manner as to permit the opening up of the same only to the maximum of the inside diameter of the tube being cleaned, thus preventing loss of power by the cutter working on the metal of the tube itself. As the centrifugal force of the cutter heads reinforce each other, the machine has four times the momentum that one freely swinging arm would have in meeting hard sections of scale.

The machine operates on ball bearings which are easily removable and are adjustable at will. The motor consists of a single turbine wheel upon which water acts through suitable nozzles. The wheel is incased in a steel band which reduces leakage around the outside of the buckets and increases the efficiency of the motor. Fig. 3 shows an application of the cleaner to a different style of head.

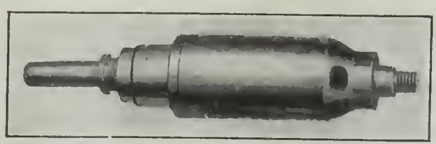


Fig. 5. Air Motor for Cleaning Fire or Water Tubes

Fire-tube boilers are cleaned with the head shown in Fig. 4. The vibrations are produced by the cutter head being hung in an eccentric position with regard to the turbine itself.

For operation with air or steam pressure, a specially designed motor containing rotary pistons is made, an exterior view of which is shown in Fig. 5. By using suitable heads for the duty to be performed it is claimed that greatly increased cleaning efficiency is obtained, due largely to the accumulative effect of the centrifugal force in the standard turbine head shown in Figs. 1 and 2, which enables it to work its way through any scale of varying thickness and composition which it is liable to meet in practice.

Jefferson Flange Union

It is claimed that this is the first three-part flange ever made that will always make a tight joint without the use of a gasket. Fig. 1 illustrates the general design.

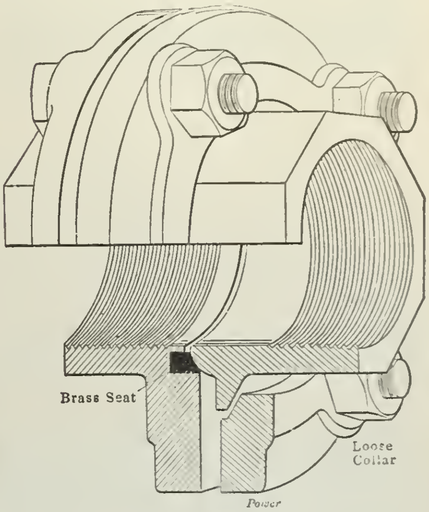


Fig. 1. Jefferson Flange Union

The body part is made of a particular grade of malleable iron. The seat is made from drawn brass tubing. The joint is spherical and thoroughly ground and the two flat disks do not come together. This

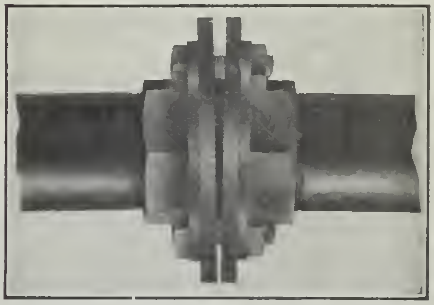


Fig. 2. Union on Pipe Out of Alinement

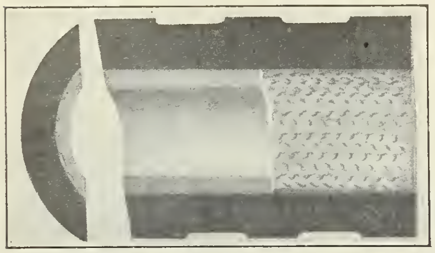
allows a tight connection to be made, even if the pipes are out of line, as shown in Fig. 2.

The loose collar allows the bolts to be applied at once. From this feature and the fact that no gasket needs fitting, much time is saved when using this style of flange, which is suitable for pressures up to 300 pounds.

It is made by the Jefferson Union Company, Lexington, Mass.

The Bollinckx Patent Bearing

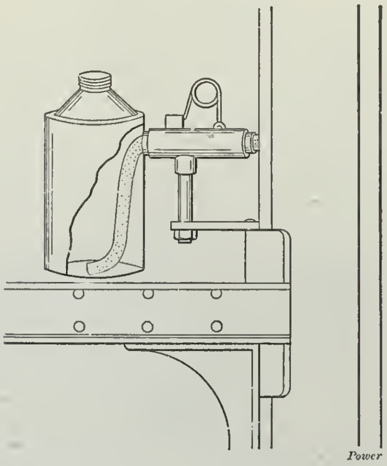
The accompanying illustration shows the method employed by H. Bollinckx, Brussels, in securing bearing metal in bearing boxes.



Bollinckx Patent Bearing

The sheet-steel box reinforces the alloy metal, and as the inner surface of the box is made with anchor cavities, the metal, when set, becomes firmly attached to the box portion, thus preventing working of the bearing metal from the box section.

The Perfection Elevator Guide Lubricator

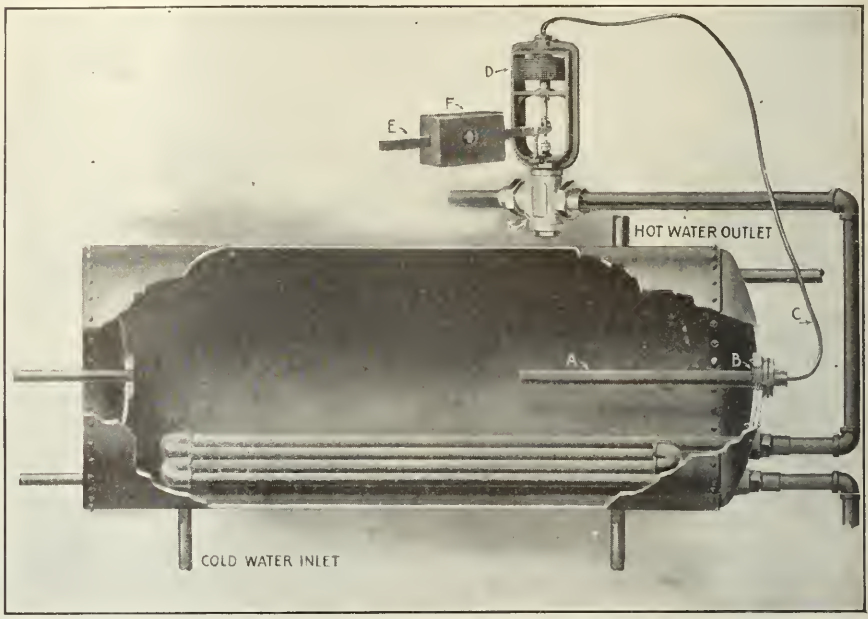


Perfection Elevator Guide Lubricator

This lubricator, which is illustrated and described herewith, consists of a can in which light machine oil is stored. A wick passes through a holder from the can and is held against the guide rail of the elevator by means of a spring as shown. The oil is fed by capillary attraction. There is no spattering or waste of the lubricant when the car is stopped or in operation. This lubricator is made by the Perfection Lubricator Company, 3 Linden street, Cambridge, Mass.

Ideal Sylphon Tank Regulator

The accompanying illustration shows a new temperature regulator for hot-water storage tanks heated by steam. The brass stem *A* screws into the storage tank through a 1¼-inch opening *B*, in either the end or the side of the storage tank



Sylphon Tank Regulator Adjusted to a Tank

A small flexible brass tube *C* connects the stem *A* with a metal bellows *D*, which operates the valve *G* controlling the flow of steam to the heating coil in the tank. This arrangement permits the valve *G* to be installed at any desired point in the steam line.

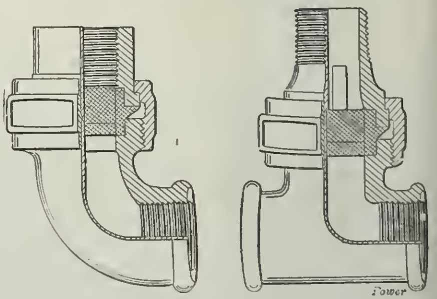
The stem *A*, tube *C* and bellows *D* are hermetically sealed, and confine a liquid which vaporizes at low temperatures. When the water in the tank becomes heated, vapor generated in the stem *A* exerts a sufficient pressure to force some of the liquid through the pipe *C* into the bellows *D*, which expands the latter by overcoming the counterpoise weight *F* and closes the valve *G*, thereby preventing further admission of steam to the coil in the tank. When the water in the tank cools slightly, some of the vapor in the stem *A* condenses and allows liquid to flow from the bellows *D*, through the tube *C* back into the stem *A*. This permits the bellows to collapse and open the valve *G*. By moving the weight *F* backward or forward on the arm *E*, the regulator is adjusted to operate at any desired temperature between 140 and 200 degrees Fahrenheit.

Extreme sensitiveness of action is claimed, due to the great power quickly generated by the volatile liquid and to the flexibility of the metal bellows or diaphragm. The outfit is completely self-contained, requiring no auxiliary power, and is made entirely of metal.

It is sold by the American Radiator Company, 282 Michigan avenue, Chicago, Ill.

New Dart Unions

The accompanying illustrations show two of a new line of unions now being made by the E. M. Dart Manufacturing Company, Providence, R. I. These are simply an adaptation of the regular union, having two bronze seats and used in connection with the regular forms of pipe fittings. Although only an elbow and a tee are shown, these can be made in any desired combination, the bronze joint being at any opening of the fitting, and the coupling either male or female, as desired. They have the strength of iron-pipe fittings in the pipe end, are made with strong nuts, substantial shoulders and in heavy patterns.



New Dart Unions

Have all defects and leaks repaired without delay. Don't wait until tomorrow.

50-Ton Crane Serves Coffee

Who ever heard of a crane being used to serve coffee? Nevertheless this is what actually happened at a banquet given on the evening of September 6 by the Nelson Valve Company, of Philadelphia, to the executive, operating and sales forces of the company. The dinner was a formal recognition of the completion of a new $1,500,000 open-hearth steel foundry. The crowning feature was the serving of the coffee, which had been arranged by Vice-president Carlisle Mason. This was done in imitation of pouring a heat. At a given signal the huge traveling crane began to move slowly on the overhead tracks until it reached the furnace. Then there was a flash and a red glow filled the room. Back swung the crane until it was just above the heads of the diners. The coffee was carried in a huge "bull ladle" suspended from the crane, and the coffee was poured directly into the cups from a spigot at the bottom of the ladle. The tables were turned on Mr. Mason, however, when the crane made a second journey and returned with a box containing a solid silver loving cup, 14 inches high, which was presented to the vice-president by the executives of the company.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

ROTARY ENGINE. John H. Zimmer, Omaha, Neb. 968,630.

GAS ENGINE. Baxter M. Aslakson, Salem, Ohio. 968,636.

ROTARY ENGINE. George H. Gross, Harrisburg Penn. 968,653.

INTERNAL COMBUSTION ENGINE. Theodor Reuter, Winterthur, Switzerland. 968,690.

INTERNAL COMBUSTION ENGINE. Georg Schimming, Berlin, Germany. 968,695.

EXPLOSION ENGINE. Martin Sears, Los Angeles, Cal. 968,702.

EXPLOSIVE ENGINE. Jay E. Woolf, Minneapolis, Minn., assignor of one-half to Ellis J. Woolf, Minneapolis, Minn. 968,725.

MOTIVE POWER SYSTEM. Alfred Wilstam, Los Angeles, Cal. 968,724.

ROTARY ENGINE. Frank Nearing, New York, N. Y. 958,786.

ELASTIC FLUID TURBINE. Raymond N. Ehrhart, Pittsburg, Penn., assignor to The Westinghouse Machine Company, a Corporation of Pennsylvania. 968,840.

ENGINE HAVING ROTARY PISTON. Friedrich Carl Kruger, Hanover, Germany. 968,859.

TIDE AND WAVE POWER. Ernesto Fuchs, Guadalajara, Mexico. 968,930.

ROTARY ENGINE. Craven Robert Ord, McAdam Junction, New Brunswick, Canada. 968,969.

ROTARY MOTOR. Herman Van Ormer, Hartford, Conn., assignor to Liberty Manufacturing Company, Pittsburg, Penn., a Corporation of Pennsylvania. 969,010.

ROTARY ENGINE. Walter Ball, Boston, Mass. 969,026.

COMPOUND ENGINE. Charles Benson, Lake City, Minn. 969,027.

ROTARY ENGINE. Roland E. List, Zanesville, Ohio. 969,070.

BOILERS, FURNACES AND GAS PRODUCERS

SMOKE CONSUMING FURNACE. Paul J. Kraetsch, Des Moines, Iowa. 968,570.

GRATE. David Boies and Joseph A. Waddell, Jr., Scranton, Penn., assignors to Spencer Heater Company, Scranton, Penn., a Corporation. 968,735.

OIL BURNER. Charles C. Wilson, Quincy, Ill. 968,825.

POWER PLANT AUXILIARIES AND APPLIANCES

TIMER FOR GAS ENGINES. Baxter M. Aslakson, Salem, Ohio. 968,635.

SPARK PLUG. Herbert F. Provandie, Boston, Mass., assignor of one-half to The Randall-Faichney Company, Boston, Mass., a Corporation of Massachusetts. 968,687.

FORCE FEED OILER. Gustave E. Franquist, New York, N. Y. 968,929.

WATER JACKET FOR EXPLOSIVE ENGINES. John F. Dodge and Horace E. Dodge, Detroit, Mich. 968,545.

CHECK VALVE. Enoch H. Heilig, Shelby, Iowa. 968,559.

CARBURETER. Joseph W. Parkin, Philadelphia, Penn. 968,597.

SPARK PLUG. Daniel M. Tuttle, Canastota, N. Y. 968,612.

TOOL FOR UPSETTING AND CALKING STAYBOLTS. George E. Wood and Edward C. Meier, Phœnixville, Penn., assignors to Heine Safety Boiler Company, St. Louis, Mo., a Corporation of Missouri. 968,625.

GUARD FOR PIPE UNIONS. Lester R. Fogg, Darby, Penn. 968,759.

STOP COCK AND DRAIN VALVE. William E. Peare, Cincinnati, Ohio. 968,794.

STEAM TRAP. William Mudd Still and Andrew George Adamson, London, England. 968,812.

SCREW PUMP. Erastus E. Bennett, New York, N. Y. 968,829.

PIPE END CLOSURE. William R. Jeavons, Cleveland, Ohio. 968,853.

CONDENSER. Ehregott T. Winkler, Kansas City, Mo., assignor of one-third to P. A. Hildebran, one-third to D. P. Gray, and one-third to John W. Roberts, Kansas City, Mo. 968,909.

VALVE CONSTRUCTION. Andrew Benner, Detroit, Mich. 968,916.

VALVE MECHANISM FOR GATE VALVES. George J. Henry, Jr., San Francisco, Cal., assignor of one-half to The Pelton Water Wheel Company, a Corporation of California. 968,944.

FEED WATER REGULATOR. Rudolph M. Hunter, Philadelphia, Penn., assignor to Charles Motley Clark. Philadelphia. 968,948.

FEED ARRANGEMENT FOR TUBE BOILERS. Louis Marie Gabriel Delaunay-Belleville, Paris, France, assignor to Societe Anonyme des Establissements Delaunay-Belleville, St. Denis, France, a Corporation of France. 968,042.

FEED WATER REGULATOR. Timothy J. Quirk, South Boston, Mass. 969,078.

MUFFLER. Hugo C. Gibson, New York, N. Y. 969,101.

STEAM SUPERHEATER. George J. Churchward, George H. Burrows, and Clifford C. Champeney, Swindon, England. 969,088.

ELECTRICAL INVENTIONS AND APPLIANCES

ELECTRIC MOTOR METER. Arthur Zipplies, Mosbach, Germany, assignor to Aktiengesellschaft Korting's Elektricitatswerke, Berlin, Germany. 968,631.

ELECTRIC SAFETY SWITCH. Clarence D. Platt, Bridgeport, Conn. 968,796.

ELECTRIC SWITCH OPERATING MEANS. Albert Whiton Bailey, Spokane, Wash. 968,520.

AUTOMATIC SYNCHRONIZER. Paul MacGahan, Wilkinsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 968,579.

ELECTROPLATING TANK. Edwin R. Williams, Streator, Ill., assignor to The Streator Metal Stamping Company Streator, Ill., a Corporation of South Dakota. 968,622.

ELECTRIC OVEN. Leon F. Parkhurst, and Harry G. Weeks, Binghamton, N. Y., assignors to Diamond Electric Company, Binghamton, N. Y., a Corporation of New York. 968,683.

ELECTROMAGNETIC DRILL. Frank E. Baney, Smelser, Wis., assignor of one-half to James E. Kennedy, Platteville, Wis. 968,729.

MEANS FOR ELECTRICALLY HEATING WATER-COOLED COMBUSTION ENGINES. Abbot Augustus Low, Horseshoe, and August Wassmann, Astoria, N. Y., said Wassmann assignor to said Low. 968,780.

SYSTEM AND METHOD OF ELECTRICAL DISTRIBUTION. Percy H. Thomas, Montclair, N. J., assignor to Cooper Hewitt Electric Company, New York, N. Y., a Corporation of New York. 968,896.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Do's for Engineers

Do all that you can for your employer.

Do the best of work at all times.

Keep your eyes open in the plant and try to make improvements.

Keep the engine in good running order.

Keep the oil holes clean.

Be sure that the machinery is in good condition before starting up.

Be sure not to leave the engine room very long while the engine is running, unless there is someone around.

See that all boiler connections are in the best of order.

Keep the water-gage cocks clean and blow them frequently.

Get the best men if you are a chief and develop them into better employees.

See that all nuts are tight.

Be sure that there are oil grooves where needed in the boxes.

Keep the details well looked after.

Make a job right for all possible needs —not just good enough.

Make all changes and repairs as good as the rest of the job.

Keep everything clean around the plant and in the best of condition.

Keep from getting into the habit of saying, "I have done enough for today, and will do the rest tomorrow," as tomorrow never comes.

Keep in touch with the latest developments in the engineering line by reading a good engineering magazine.

BUSINESS ITEMS

In order to better serve its many customers throughout the South, the International Acheson Graphite Company has opened a Southern branch at Atlanta, Ga.

Wm. B. Merrill & Co., Boston, Mass., advise us that their business has grown to such an extent that they have had to enlarge their plant to take care of the increased business. They have installed a 20-horsepower Fairbanks gas engine, new lathes, new planers, so that the orders can be kept up with.

The United States Indestructible Gasket Company, Hudson Terminal building, New York, is building a new plant in Brooklyn, N. Y., which will have a space for machines equal to 10 feet wide by 500 feet long. This new plant affards greatly increased facilities for the manufacture of its products, consisting of corrugated, flat and grooved copper, lead, steel, bronze, indestructibleite, Ames alloy, Norway and Swedish iron, Monel metal, sparkite, copper-asbestos, wire, paper and other gaskets, and other specialties.

The Lagonda Manufacturing Company, Springfield, Ohio, announce that they have bought out the Enterprise Machine Manufacturing Company, acquiring the rights and patents for the full line of water strainers and tube cleaners manufactured by the latter. Special attention is being paid to the development of the water strainer. Although this strainer has been on the market for several years and is well beyond the experimental stage, the Lagonda Manufacturing Company has made several important improvements and the apparatus is now known as the Lagonda-Enterprise strainer. This strainer is suitable for removing impurities from boiler feed water, circulating water, etc.

NEW EQUIPMENT

A new electrical substation will be equipped at Ingersoll, Ont.

Robert McGinnis, of O'Neil, Neb., will erect a modern creamery.

The city of Hull, Que., will extend its waterworks system.

Edmunston, N. B., will install a new waterworks system.

Ottawa, Canada, will spend $2,000,000 extending its water system.

The Greenville (Tex.) Railway Company will construct a power house.

The Oneonta (N. Y.) Milk Company is erecting a two-story creamery.

The State Tuberculosis Sanatorium, Alto, Ga., will erect a power house.

The city of Vernon, Tex., voted bonds for the construction of waterworks.

Chippewa, Ont., will make large extensions to its waterworks pumping plant.

Prince Rupert, B. C., is in the market for equipment for its electric-light plant.

The Brookline (Mass.) Water Works contemplates the addition of another pump.

The Exchange Packing House Company, Colton, Cal., will erect a packing plant.

The town council, Fort Mill, S. C., is considering the construction of waterworks.

The city of Britton, Okla., will vote on issuing $35,000 bonds for its waterworks.

The city of Tulsa, Okla., will install another pump of 4,000,000 gallons capacity.

North Buxton, Ont., wants figures for a new pump house, wheel, boiler, engine, etc.

The Goliad (Tex.) Water and Light Company contemplate establishing a light plant.

Allen P. Wilson, New Boston, N. H., is in the market for a 15- to 20-horsepower oil engine.

The plant of the Union Ice Company, Mentone, Cal., was destroyed by fire. It will be rebuilt.

The Tulsa Corporation, Tulsa, Okla., will install a 600-kilowatt turbine and additional boilers.

Fire did about $12,000 damage at the plant of the Cadillac (Mich.) Water and Light Company.

The city council, Marion, Ind., is said to have purchased site for a municipal street-lighting plant.

The New Method Laundry Company, Los Angeles, Cal., is having plans prepared for engine-room addition.

The Arlington Hotel Company, Santa Barbara, Cal., contemplate installing an electric-light and power plant.

The Davenport (Iowa) Locomotive Works has awarded contract for the construction of a $10,000 power house.

The city of Tacoma, Wash., has passed an ordinance providing for the construction of a substation to cost $104,000.

The Evanston (Wyo.) Electric Light Company proposes to construct a power plant as soon as water rights are secured.

The city of Haskell, Okla., will open bids September 22 for waterworks. Two pumps, boiler, fire hydrants, etc., will be needed.

The Wheeling (W. Va.) Electrical Company has been granted a franchise and will at once commence work on a $700,000 plant.

Plans are being prepared for a new building for Youngs Market Company, Los Angeles, Cal. Complete refrigerating system will be installed.

The Bartlesville (Okla.) Interurban Railway Company will install another 500-horsepower battery of boilers and a direct-connected generating set.

The city of Mansfield, Ohio, contemplates making additions and changes in its waterworks. Information can be obtained of John Cahall, director of public service.

The Hillsboro (Ill.) Electric Light and Power Company has purchased site for a new plant which will be four times the size of the present central station.

J. J. Lambach, Wm. Harper, W. J. Benjamin and W. B. Bunker, all of Las Vegas, N. M., have been granted a franchise to erect a power plant. This is to cost $100,000.

Bids will be received by the trustees of public affairs, Jackson, Ohio, until September 29, for furnishing material and enlarging and improving its municipal electric-light plant.

The San Diego (Cal.) Electric Railway Company will erect a new power plant covering a whole city block in Artic, Cal. This will cost about $225,000 and will have a capacity of 35,000 kilowatts.

The Milwaukee (Wis.) Electric Railway and Light Company, recently obtained permit to build an $80,000 addition to the Commerce street power house; also a substation at Clinton and Maple streets to cost $25,000.

The San Joaquin Valley Power Company has begun the erection of a station on Union avenue, Bakersfield, Cal. The plant will contain eight boilers of 300 horsepower each, two engines and two 2500-kilowatt generators.

NEW CATALOGS

Trenton Engine Company, Trenton, N. J. Catalog. Reeves steam engines. Illustrated, 38 pages, 6x9 inches.

American Steam Packing Company, Boston, Mass. Catalog. Packings. Illustrated, 56 pages, 4½x6½ inches.

De Laval Steam Turbine Company, Trenton, N. J. Catalog. Centrifugal pumps. Illustrated, 96 pages, 6x9 inches.

B. F. Sturtevant Company, Hyde Park, Mass. Catalog No. 175. High-pressure blowers. Illustrated, 44 pages, 6x9 inches.

Ridgway Dynamo and Engine Company, Ridgway, Penn. Catalog. Four-valve engines. Illustrated, 22 pages, 8x10½ inches.

The Direct Separator Company, Syracuse, N. Y. Catalog. Sweet's combination oil and water separators. Illustrated, 12 pages, 6x9 inches.

Schutte & Koerting Company, Philadelphia, Penn. Catalog 1, Section A. Koerting universal double-tube injectors. Illustrated, 16 pages, 8x11 inches, paper.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

SALESMEN—We have an A-1 proposition for high-grade salesmen. Address Box 247, Power.

WANTED—Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," Power.

AN ENGINEER in each town to sell the best rocking grate for steam boilers. Write Martin Grate Co., 281 Dearborn street, Chicago.

WANTED—Draftsman familiar with boiler-room practice. October 15. Location Northeastern Ohio. State age, qualifications and experience fully. Address Vulcan Furnace Company, Ellicott Square, Buffalo.

WANTED—Competent chief engineer for a plant in a village near Rochester, N. Y.; must be capable of making his own repairs and doing steam fitting; salary $1200 to start. Address Box 278, Power, stating experience in detail.

Power and The Engineer

Vol. 32 NEW YORK, SEPTEMBER 27, 1910 No. 39

AT a banquet given last June by the graduating mechanical engineers of the University of Illinois, Professors Thorpe and Goodenough of that institution spoke a few parting words on the successful engineer, using as their text the time-honored formula for indicated horsepower, which is known to almost every engineer.

$$\frac{P\ L\ A\ N}{33{,}000} = I.\ H.\ P.$$

One of our contributors was so impressed with their treatment of the subject that he forwarded the substance of what was said to POWER, and in slightly modified form we are repeating the gist of their remarks.

Although the talk was delivered to technical men, it applies equally well to the practical worker, and in fact to all men of any profession or trade.

To begin with, the "33,000" in the usual horsepower formula is a constant and therefore may be represented by a symbol.

For our purpose $1 \div 33{,}000$ is best represented by K, making the formula read:

$$P\ L\ A\ N\ K = I.\ H.\ P.$$

As P comes first it is likely to be of great importance, which is true of *Patience.* No characteristic is more helpful under trying circumstances, and nowhere is the exercise of patience more necessary than in the engine and boiler rooms. The long hours of service, the Sunday jobs and the repair work common to even the best of plants, demand that this one quality be developed to an unusual degree.

PLANK = I. H. P.

P = Patience A = Accuracy
L = Loyalty N = Nobility
K = Knowledge

I. H. P. = Important High Position

L fittingly comes next; it ought to be prominent, for *Loyalty* to one's employer as well as to one's higher self is the essence of success. The good of the firm demands that all pull together. No minor jealousies or personal affairs should ever be given precedence, for, even from a selfish motive, loyalty to the firm means success to the individual.

Yet no engineer is a real success unless he has the next quality, *Accuracy,* denoted by A. There are few occupations in which accuracy is necessary to such an extent as that of the engineer.

As a fitting addition to the list of good qualities we have N for *Nobility.* Be noble! Treat your fellowmen as a man among men. Live as you would like to see men live—as you would have your son live.

Last, but by no means least, there is K for the indispensable *Knowledge* that enables the engineer to make the best use of the four qualities mentioned.

Turning to the other side of the equation, there are the symbols, *I. H.P.*, the equivalent of the first member. The result of such a combination of qualities can only be *Important High Position.*

You have then Patience, amplified by Loyalty, strengthened by Accuracy and magnified by Nobility; all these multiplying your Knowledge, and the whole resulting in Important High Position, or Success.

Adopt this "plank" and see how near the top it will land you.

Think it over.

Plant of the Holton Power Company

By Howard H. Bliss

A small hydroelectric plant at Holtville, Cal., supplying power and light to the various towns of the Imperial valley. The supply water is taken from the Colorado river at a point 60 miles distant from the power house and discharges below sea level.

The hydroelectric plant of the Holton Power Company, situated in the Imperial valley, that part of California latest to be developed, embodies some interesting features. This station, which is located at the town of Holtville, draws its water supply from a canal probably longer than that of any other power system in America; the water travels more than 60 miles from the diversion works to the point at which it is used, passing through parts of California and Mexico and twice crossing the international boundary. After giving up its energy at the station, the water is discharged below sea level and flows 40 miles to the Salton sea, a large lake whose surface is 250 feet below the level of the ocean.

The Holton company's hydroelectric plant and its steam auxiliary are the only generating stations in Imperial county, which is 4000 square miles in area. Although of moderate size, this plant normally supplies all the electric energy used by the 20,000 inhabitants of this district, and even sends some power across the border into Mexico.

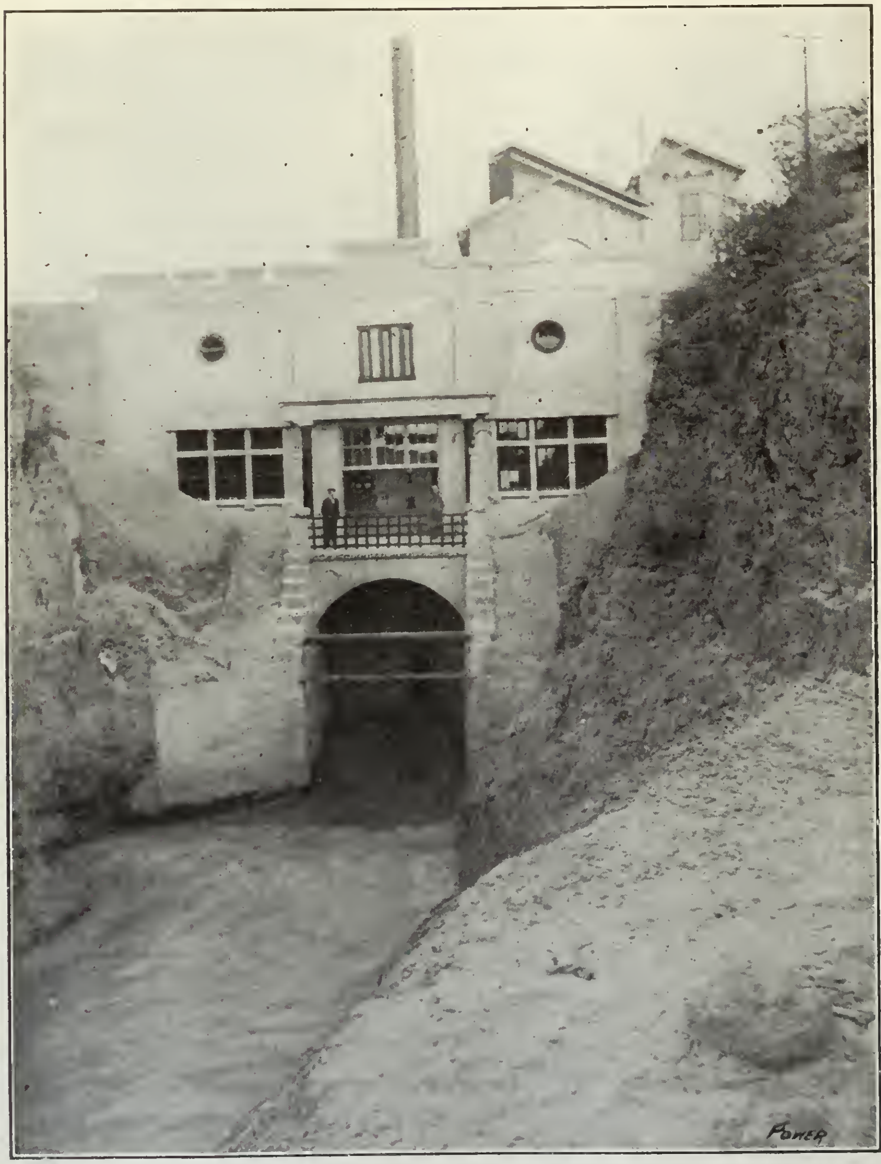


Fig. 1. New Power House with Old Plant in Background

Water Supply

Water is taken from the Colorado river 10 miles below Yuma. It is carried south about a mile to the Mexican border and then 50 miles south and west through northern Mexico in the main canal of the California Development Company. From this great waterway, which is about 100 feet wide and from 6 to 25 feet deep, a branch turns north about 10 miles east of Calexico, and after crossing the international boundary, carries the water to Holtville. That portion of the water which is not diverted into the Holtville main is used for irrigating the land of the Imperial valley. A map of this district is shown in Fig. 2.

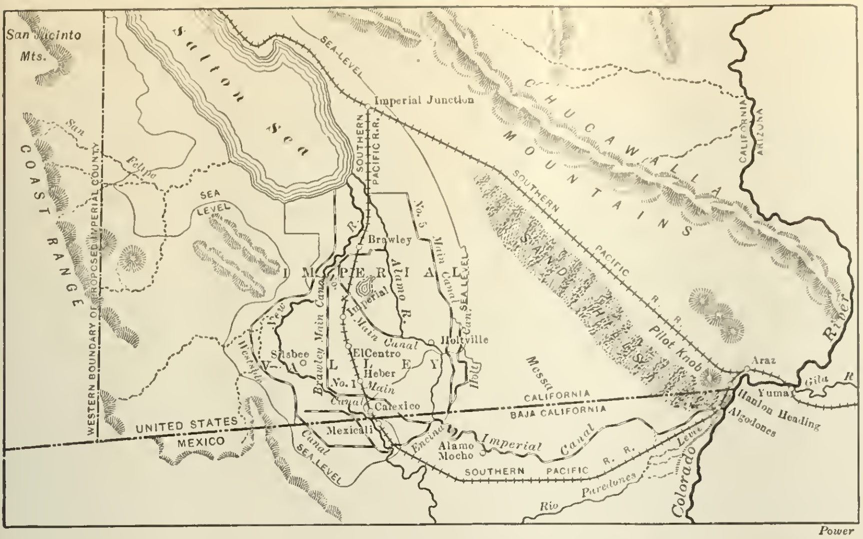
From a point on the Holtville main, 7 miles north of the boundary, the power company's 25x4-foot ditch extends 2½ miles westward and empties into a small reservoir. From this point the water is carried through a 54-inch pipe line to the power house. Practically the entire 1300 feet of this penstock is redwood stave pipe, buried underground. At a point close to the station where the static-pressure head is 33 feet, a riveted steel pipe replaces the wooden pipe. By means of a sharp bend this passes through the roof of the power house and terminates at the turbine, where the static head is 55 feet.

The draft tube is built up of steel plates, varying from a circular cross-section of 4 feet in diameter at the turbine to a 4x8-foot ellipse at a point 26 feet below. This pipe leaves the turbine vertically, but by means of two 45-degree bends, the course of the water is altered so as to issue horizontally. Although there is a drop of 26 feet between the shaft of the turbine and the center of the discharge outlet, the latter is submerged to such an extent that the suction head averages only about 20 feet.

From the waste weir beneath the power house the water is led through a short race to the bed of the Alamo river, thence to the Salton sea. Here it is evaporated, as the sea has no outlet. The hot sun and dry air of the valley are causing the gradual disappearance of this great body of water, having an area of over 200 square miles, despite the fact that it is continually supplied with large volumes of water from the waste weirs of the California Development Company.

Building and Equipment

The small building shown in Fig. 1 spanning the walls of a water-worn cañyon, houses the equipment. The structure is of concrete and has one main room, 52x29 feet, and several smaller rooms. The machinery was manufactured and installed by the Allis-Chalmers Company, and consists of a turbine, a three-phase



Power

Fig. 2. Map of Imperial Valley

generator and a step-up transformer set, with the usual switchboard and exciter equipment. The turbine is of the axial outward-flow type, double ended, with a horizontal spindle and runs at 450 revolutions per minute. Direct-connected to the turbine is a 2400-volt, revolving-field type of generator, rated at 700 kilowatts. The three transformers are connected in delta and raise the voltage to 15,000 volts at which the current is transmitted over the valley, the greatest distance being 24 miles, and is stepped down at the various points of utilization. A 15-kilowatt compound-wound exciter is belted to the main shaft. The generating set and switchboard are shown in Fig. 3, the latter consisting of four panels mounted on a concrete frame.

Surge Pipe

To replace an old surge pipe, the action of which had been unsatisfactory, the company has just constructed a sheet-steel reservoir supported by a timber structure at the rear of the power house. The reservoir holds about 40,000 gallons when the plant is not running; when water is flowing through the pipe line and the head at the power house is reduced by friction, the water level in the reservoir is much lower.

Distribution of Power

Power is transmitted by a pole line to the various towns in the district most of which use the power for both lighting and manufacturing purposes. The wooden poles were treated with hot carbolineum from the butt to 2 feet above the ground, but owing to the dry air of this locality they have shown no signs of decay.

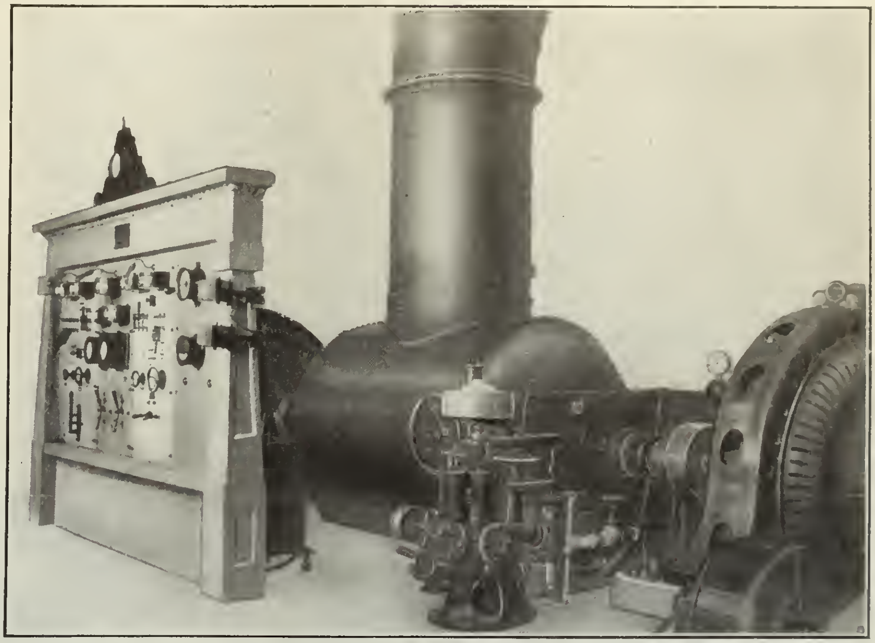


Fig. 3. Generating Set and Switchboard

Owing to the large number of manufacturing enterprises using power, the day load on the station is nearly equal to the night load. This results in an unusually high load factor

The company has offered large inducements for consumers to use all they can for day motor loads, giving a rebate of 75 per cent. from the regular rate of 20 cents per kilowatt-hour.

In order to cope with emergencies, the old plant, shown in the background of Fig. 1, is kept in readiness to supplement the new plant. It contains a 250-kilowatt turbo-generator set working under an available head of 52 feet.

Auxiliary Steam Plant

At El Centro is located the steam auxiliary plant containing a St. Louis tandem-compound Corliss engine, belt-connected to a 250-kilowatt generator of the revolving-armature type. To supply steam to the engine there are two Stirling boilers, rated at 300 boiler horsepower each, using oil as fuel. The plant is run non-condensing, with no superheater between the high- and low-pressure cylinders, hence it is very extravagant in the use of fuel.

Not long ago, S. Kirlin met Eccentric Si, who is a queer fellow at best. He jotted down a few of his most brilliant remarks, and here they are.

I writ t' th' Power fellers an' ast 'em ef they cud tell us whar we cud sell th' old worn-out rope frum our transmishun system. They writ back an' told us t' try a cigar factery.

Old Hal Mossback bot a farm. Says he's goin' t' raise hydraulic rams on it. Sed th' big rain tother day washed everthin' off'n it 'cept th' morgige.

They'r bildin' a nursin' bottle factery over t' Jayville. When th' ingin kum fer it th' feller thet's goin' t' play ingineer sent th' eccentric back t' th' factery an' roasted 'em fer sendin' out a wheel with th' hole bored so fer frum th' center.

Sum wun told Bill Grimes thet ef he'd put a bushel uv green cucumbers inter his biler thet it'd take th' scale out'n 'er. Bill opened 'er up t' put 'em in an' ther wuz so dumd much scale in 'er thet he didn't hev room fer only a peck. When he fired up an' started th' ingin she turned over a few times, went t' groanin' an' doubled up her connectin' rod.

Bill Blowoff's biler blowed up with him on top uv it. He went up in th' air 'bout a quarter uv a mile an' kum down in th' crick; didn't hurt him much, but he got sorter damp 'fore they got 'im pried loose frum th' mud. Sid Diggles told Bill thet he wuz a dumd fool fer kumin down after gettin' so clost t' heaven, fer he wasn't likely t' ever get thet nigh t' it agin.

Combination Engine and Boiler

By Warren H. Miller

In this product of German design and manufacture the engine, boiler and auxiliaries are combined, permitting a high degree of superheat to be utilized by the engine and doing away with the numerous heat losses found in the average plant. The water consumption of this type of engine is about 10 pounds per horsepower-hour.

It is a big step in economy from the 35 pounds steam consumption of the average 60-horsepower high-speed engine to the 10 pounds steam consumption of the German combination steam engine and boiler of 60 horsepower, and a still greater advance from the five pounds of coal per horsepower-hour of small 60-horsepower plants to the one pound per horsepower-hour of the combination engine and boiler. This disparity may be accounted for by the reduced cylinder condensation, and by careful elimination of the apparently insignificant losses which ordinarily occur about a small plant. The Germans have developed possibilities in a careful use of coal, fire and water, so as to obtain in the ordinary small semi-portable plant a better economy than is realized in our large central stations.

The illustrations show two types of these engines: one, a noncondensing simple engine with the boiler and feed-water heater attached; the other, a compound engine with the boiler, feed-water heater and a jet condener attached. The sectional view gives a clear idea of how the engine, boiler and auxiliaries are combined, making the unit semi-portable; for all that it requires is two brick piers, to install it ready for use. The boiler is of a type quite different from that ordinarily seen. The cylinder and the steam dome are cast in one piece and are riveted direct to the shell of the boiler, and at least half the boiler is superheating surface.

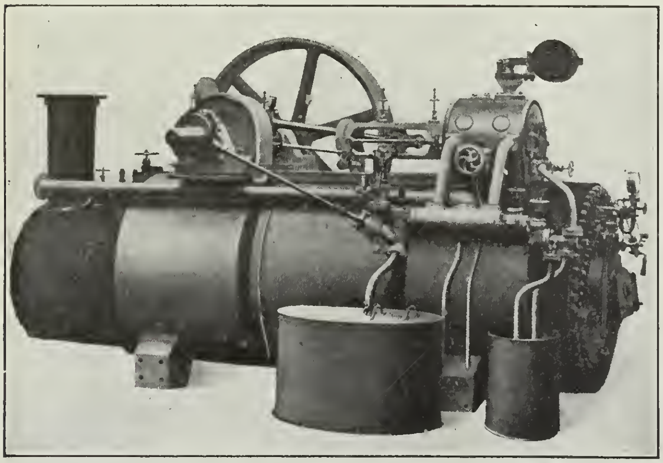


Fig. 1. Simple Noncondensing Engine

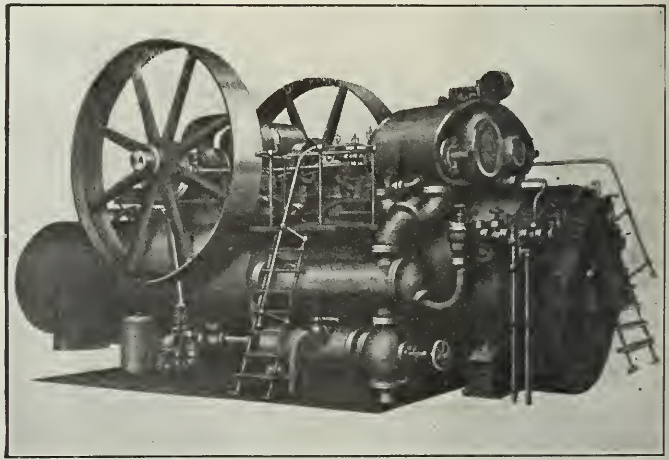


Fig. 2. Compound Condensing Engine

It can be readily seen that whatever mechanical difficulties these arrangements may introduce, they at once eliminate condensation and radiation from the steam pipes and cylinder walls; return all unused heat to the boiler instead of to the air; and utilize the heat from the waste gases for superheating the steam. Yet there is nothing in this disposition of the parts that constitutes poor mechanical design.

For the purpose of contrast, compare this type of engine with an ordinary isolated boiler and engine plant of from 60 to 100 horsepower. Usually a plant of this size has an internally fired locomotive-type boiler, with a 25-foot iron stack. The portion below the grate is exposed to direct radiation to the ground, and the stack temperature is anywhere from 500 to 600 degrees. Sometimes the boiler is lagged, sometimes it is not; the feed water is drawn from a barrel or possibly from a primitive hotwell; and the steam pressure is usually about 80 pounds. On the other hand, the combination engine and boiler herein described has a corrugated-iron, internally fired furnace which can easily be kept clean and efficient; and the gases, after passing through the tubes, pass over the high-pressure superheating coils located in the smoke box, returning over a second set of low-pressure superheating coils, and thence to the stack at about 370 degrees. The whole unit is lagged and a feed-water heater which raises the feed-water temperature to 170 degrees is located in the path of the waste gases. The usual steam pressure is about 180 pounds.

In the typical small steam plant there is usually about 25 feet of piping between the boiler and the engine; the cylinder of the engine is lagged with cast-iron plates fitted over an asbestos filling; a complete and elaborate system of drains is provided; the engine is simple noncondensing and runs at about 250 revolutions per minute. With the combination plant there is no steam piping exposed to the temperature of the air; the steam dome and the low-pressure cylinder are cast in one piece with the steam in the dome surrounding the cylinder; the high-pressure cylinder is direct-connected to the high-pressure superheater; the steam enters the high-pressure cylinder at about 650 degrees Fahrenheit and expands down to about 235 degrees; it is then reheated to 360 degrees, and is again expanded to a vacuum of 27 inches at which it enters a jet condenser located at the side of the boiler. The revolutions of the engine are 170. Practically all the refinements found on the largest engines are present in this little plant, such as superheating the steam between the high- and low-pressure cylinders and jacketing both of these cylinders with an envelop of steam or gases hotter than the interior of the cylinder.

Such radical improvements in steam economy when introduced into the engineering world are usually either adopted at once by steam users, or are rejected because of some inherent defect in operation. During the first five years that these engines were placed on the market, approximately 1500 simple engines, aggregating 140,000 horsepower, and 800 larger compound engines of from 400 to 500 horsepower, aggregating about 1,500,000 horsepower, were introduced into England and the Continent by one firm of builders alone. Their field of application is broad, and includes isolated electric-light plants, agricultural machines, shops, mill work and many portable installations. They are provided with furnaces for burning either anthracite or soft coal, or special furnaces for burning tan bark, logs, straw, corn husks, lignites, coffee husks and other combustible refuse. They are also arranged for crude oil and other liquid fuels.

RESULTS OF TESTS ON STEAM ENGINE.

	Test No. 1.	Test No. 2.
Mean speed of engine in revolutions per minute	219.3	219.2
Mean speed of countershaft in revolutions per minute	230.7	230.4
Total indicated horsepower	47	59.4
Brake horsepower of engine (3 per cent. added for loss by belt and countershaft bearings)	43.2	55.1
Mechanical efficiency, per cent	91.9	92.7
Consumption of steam per indicated horsepower per hour, pounds	10.02	9.5
Consumption of steam per brake horsepower per hour, pounds	10.9	10.29
Heat units required per indicated horsepower per hour, feed water at 100 degrees Fahrenheit, B.t.u.	6063	5903
Coal consumption per indicated horsepower per hour, pounds	1.23	1.16
Coal consumption per brake horsepower per hour, pounds	1.32	1.23
Thermal efficiency of the engine (measured from feed water at 100 degrees Fahrenheit), per cent	19.3	19.8
Total efficiency of plant per brake horsepower, per cent	13.4	14.3
Barometer, inches	29.96	30.05
Mean vacuum (by barometric reading), inches	26.1	26.3
Oil consumption per brake horsepower per hour for cylinder, ounce		0.06
Oil consumption per brake horsepower per hour for gearing, ounce		0.08

RESULTS OF TESTS ON THE BOILER.

	Test No. 1.	Test No. 2.
Duration of test, hours	5	4
Mean steam pressure pounds per square inch	180	180
Temperature of saturated steam degrees Fahrenheit	373	373
Temperature of steam at admission to high-pressure cylinder degrees Fahrenheit	644	680
Temperature of steam at admission to low-pressure cylinder degrees Fahrenheit	340	376
Mean superheating in first superheater....degrees Fahrenheit	304	342
Mean superheating in second superheater degrees Fahrenheit	136	160
Pounds of feed water evaporated per square foot of heating surface per hour	3.05	3.68
Mean temperature of feed water. degrees Fahrenheit	100	100
Total evaporation......pounds	18.02	18.16
Mean draft in smoke box, water column in inches	0.27	0.27
Mean temperature of the engine room.....degrees Fahrenheit	70	73
Mean temperature of hot gases in the furnace degrees Fahrenheit	1227	1347
Mean temperature of hot gases in front of the first superheater....degrees Fahrenheit	698	787
Mean temperature of hot gases in front of the second superheater....degrees Fahrenheit	494	526
Mean temperature of hot gases in smoke box degrees Fahrenheit	376	412
Efficiency of boilers, including superheaters.......per cent.	75.4	77.7

There is no particular difficulty introduced by having the cylinders located in the stack or in the steam dome. In both, the end plates are entirely exposed, and as the exterior of the cylinder walls seldom requires attention, their inaccessibility counts but little. The clearances are arranged to allow for the additional expansion of the steam due to superheat. At the temperature used the steam has about 24 per cent. greater volume per unit of weight than when saturated; this gives a correspondingly greater horsepower per pound of steam. The piping of the superheaters is mild steel, as are all the leads to the cylinders. The cylinders are cast of tough gray iron and are tested by hydraulic pressure after being riveted and calked. The stuffing boxes of the piston rods and valve rods have a high grade of asbestos packing and give no particular trouble in operation. The pistons are especially constructed for use with superheated steam, the moving parts being light and having well balanced areas. The steam valves are of the balanced piston type, with steel rings and cast-iron bodies. The rest of the engine is very similar in design to ordinary steam engines of similar size and running at 200 revolutions per minute, although no attempt has been made to introduce a crank-oiling or gravity-lubrication system; they are lubricated by oil cups located at various points.

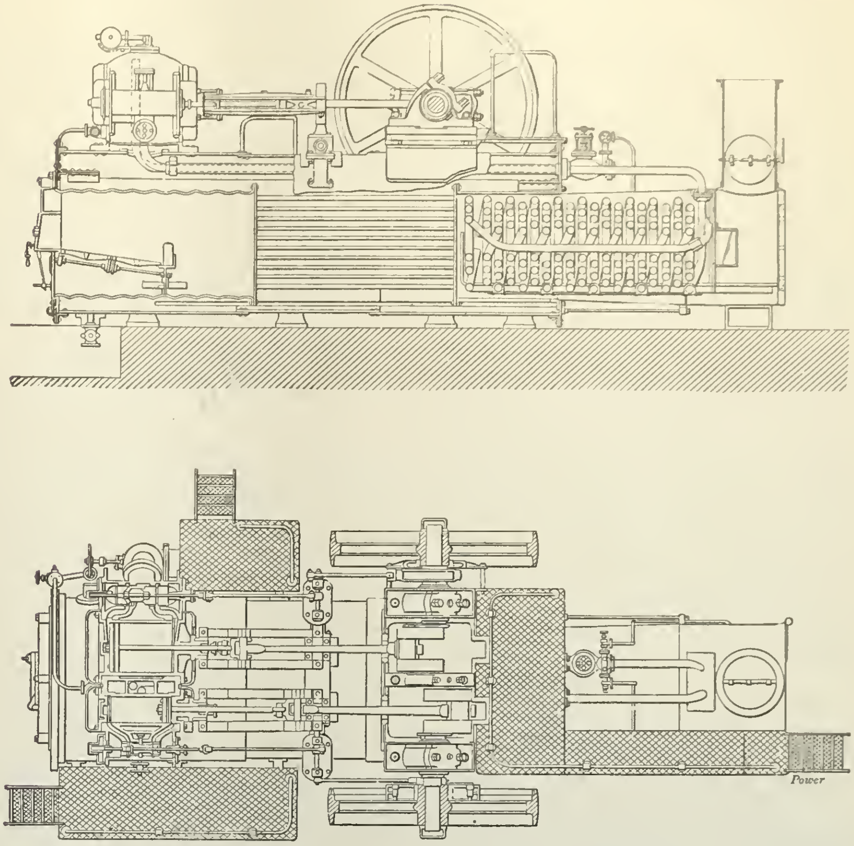


FIG. 2. PLAN AND ELEVATION OF COMPOUND ENGINE

The boiler is designed so that by unbolting and removing the rear head, the entire interior, including the furnace, tubes and superheater, may be removed in one piece for the purpose of cleaning.

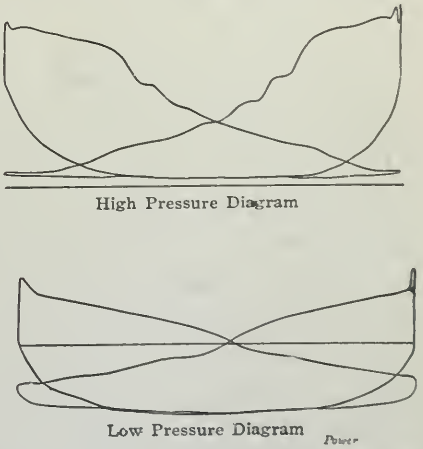


FIG. 4. DIAGRAMS TAKEN FROM COMPOUND ENGINE

The front head is smaller than the rear head and is bolted to the inside of the flange; this is a simple expedient, and one that should give no trouble if proper gaskets are used; at the same time it vastly simplifies the work of cleaning.

For removing the soot from the tubes of the superheater, handholes are provided in the smoke box; this permits them to be blown down daily without going to the trouble and expense of a thorough cleaning, which can be given when the whole interior of the boiler is removed.

These engines and boilers are made in about fifty different sizes which are divided into three general groups: first, the small, simple engine of from 12 to 115 horsepower, having the cylinder located in the steam dome and provided with one superheater; second, the cross-compound engine of from 45 to 430 horsepower, having both high- and low-pressure cylinders placed side by side in the steam dome, provided with one superheater and running either condensing or noncondensing; third, the tandem-compound engine of from 20 to 85 horsepower with the high-pressure cylinder located in the steam dome and the low-pressure cylinder in the smokestack, having superheaters for both the high- and low-pressure steam.

Fig. 4 shows diagrams taken from one of these compound engines. There is a sudden drop in the pressure immediately after admission, due to the drop in the temperature of the steam, the cylinder not yet having been heated sufficiently after the previous expansion; also there is considerable reëvaporation during expansion, due to the cylinder being jacketed with live steam and the return of the heat which was taken up by the cylinder walls at admission.

The results of a test on a 40-horsepower engine of this type by Professor Jesse, of the Royal Technical Institute of Berlin are shown on the preceding page.

Serious Troubles from Small Things

By T. A. Emerson

Troubles with prime movers in central stations, sometimes involving a total or partial shutdown, are frequently caused by negligence on the part of attendants or by some apparently insignificant detail which has been overlooked in the original layout. A few such incidents are cited.

One of the most common causes of trouble in a power house is the failure of the lubrication system in some form or another. This is somewhat surprising considering the attention that has been given to the subject for so many years, but apparently no system, whether forced or gravity feed, has been adequate to overcome the human element of carelessness. One of the most essential duties of the watch engineer is to go around at frequent intervals and see that the oilers are doing their duty. An instance of this may be cited where serious trouble was occasioned by an inexperienced oiler. The bearings of the engine, driving the air pump of one of the main units, were lubricated by means of siphons, and the oiler, being inexperienced in this class of work, instead of pushing the wicks down into the tube in the oil cup, simply dropped them into the oil well, with the result that the engine was started and run for a short time with its bearings quite dry. Of course, it could not withstand this long, and the engineer in charge soon found that this main unit was running against atmospheric pressure and could not carry the load. Consequently, until the peak of the load was over, the station was in a demoralized state and low-voltage complaints were received from all parts of the lines.

Another instance of failure in lubrication occurred with the sight-feed lubricator on a high-speed engine. The piston of the lubricator had become clogged and no oil was passing into the engine. It was impossible to open the lubricator while the engine was running, and it was also necessary to keep the engine running until the heavy load had passed. Fortunately, emergency cups had been fitted at the tops of the cylinders on each line, and the cylinders were kept lubricated by flushing them at frequent intervals; and the rods were kept from heating by applying oil swabs.

An instance somewhat similar to the first one related was due to the breaking of the governor belt and the consequent racing of an engine which drove a compound circulating and air-pump set. This governor was of the ordinary ball type and when the balls were raised the steam was cut off. Hence, when the belt snapped, full steam was admitted to the engine, causing it to race. It was necessary to shut the stop valve of the air-pump engine as quickly as possible in order to avoid any damage, and the main engine, whose vacuum was thus affected, began to exhaust against the atmosphere by means of the automatic atmospheric valve. This lowered the voltage, and in order to bring the lamps up to their normal brilliancy, it was necessary to start the air-pump engine again and govern it direct from the stop valve, that is, adjust it according to the duty on the pumps; in this way the run was completed, although it meant taking a man from other duties and keeping him at watching the pump. This incident emphasizes the necessity of a periodic overhauling of all details; if the belt had been regularly overhauled it is probable that such trouble would have been avoided.

Intimately connected with lubrication is the subject of the packing used in the stuffing boxes of the engine. No inexperienced hand should be allowed to attend to this part of the maintenance. A breakdown, which fortunately did not affect the running of the station, owing to a spare unit being available, was due directly to a badly packed stuffing box. This box, which was on the high-pressure cylinder, was packed with metallic packing but the work had been done very hurriedly. The first indication of trouble was the excessive dryness of the rod when the engine was running. Immediately after this was noticed, and before anything could be done, the friction of the rod caused the packing to run. The load was at once taken off, and the engine was shut down. Upon examination the rod was found to be only slightly scored, so that without any further trouble the gland was repacked. Care was taken to screw down the gland little by little on each stud and to keep the gland true with the rod.

In the early days of electric lighting, the governors of the engines were often made too light to take up the fluctuations in the load, but this trouble is now seldom met with. The specifications for electric-light-plant equipment are now so severe as regards governing, that with careful maintenance there should be very little trouble due to hunting or surging. However, a rather curious instance of governor trouble occurred with a certain marine type of engine which operates its governor from the crank shaft through a flexible coupling. A set of screws on this coupling slacked back, and the sleeve of the governor could not make the full travel. As the engine was governed on the throttle this could not be closed sufficiently to control the speed of the engine at light loads; therefore, it became a very difficult matter to parallel the alternator, coupled to the engine, with the other machines. It was some time before the cause was discovered and at first it puzzled the engineer in charge.

The subject of piping being so intimately connected with prime movers, reference may be made to the very common practice of leading all the exhaust branches from the engines into one common exhaust pipe. The reason for this system being adopted is that the arrangement permits of cheapness, but the method has its dangers. In one particular instance the exhaust steam from several engines was led into one large exhaust pipe; the pipe cracked and occasioned such an escape of steam as to compel the shutting down of the whole plant until the crack was temporarily repaired.

Care and Operation of Steam Boilers

By William Westerfield

A discussion of the proper and improper methods of firing and their influence upon combustion, together with some useful hints upon cleaning fires.

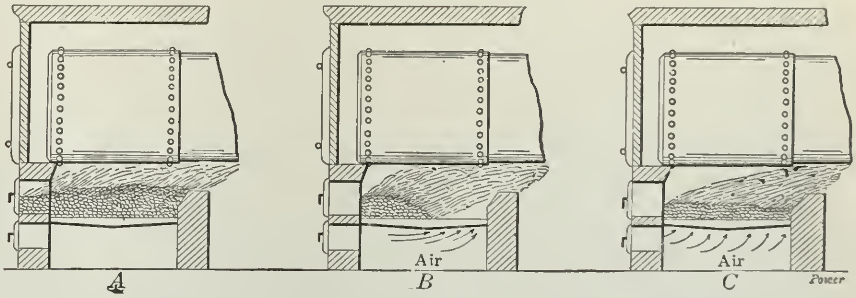
Efforts to secure economy and efficiency in operation should begin in the boiler room, not that these factors are more important there than elsewhere, but it is not wise to seek them in one part of the plant and neglect them in another. As generation of power begins in the boiler room, it is the logical place to begin to locate losses and errors that count against efficiency and economy. Then, in order, the entire equipment should be gone over, studying every part in its relation to the whole and learning what each unit is expected to do, and if it is not giving the proper service the cause should be found and removed.

One of the first things to do upon taking charge of a boiler room is to see that the boilers are properly fired and that they are supplied with water at as nearly a uniform rate as possible. Every engineer ought to know how to properly fire a boiler, because if he does not he is liable to be victimized in the boiler room by those who are either too lazy or too ignorant to do the work properly.

It is important that the fireman understand the part which air plays in the process of combustion. The carbon and hydrogen which forms the greater part of coal, will not burn unless combined with the oxygen of the air, and the amount of heat produced from the combustion will depend upon the proportions of air and fuel gases, also upon the nature of the compound formed by the mixture, which is dependent upon the temperature of the furnace. To make this clear, carbon is the principal element in the fuel, and oxygen the element in the air, upon which combustion is dependent. One pound of carbon uniting with two pounds of oxygen results in perfect combustion, the product being carbonic-acid gas, or CO_2, developing 14,500 heat units. The term CO_2 expresses the quantity of each element in the compound; that is, one part of carbon is expressed by C, and two parts of oxygen by O_2. If insufficient oxygen or air is supplied to the furnace the carbon or combustible matter will not burn to the carbon dioxide usually formed, and the resulting product will be CO, or carbon monoxide, the term CO indicating equal quantities of the two elements. In this case only 4450 heat units will be developed, which is a loss of 69 per cent. of the heat value of the fuel as a result of insufficient air.

A large volume of air is needed with a quick-burning fuel, like bituminous coal, at the moment of ignition when a fresh charge has been placed on the fire. The free gases in this class of fuel are distilled very rapidly, in fact so rapidly that it is impossible to supply air in sufficient quantity to secure perfect combustion, as is evidenced by the volume of smoke emitted from the stack at this time. After the free gases have been distilled and while the solid carbon of the fuel is being burned, but a comparatively small quantity of air is required.

A good index to furnace conditions is the appearance of the incandescent bed of fuel. If the fire burns down brightly, so that after the distillation of the volatile gases it appears as a white incandescent mass, one may be assured that conditions are nearly right. But, if the fire burns to a dull red, it is certain that the fuel is not being burned to advantage. For this last condition there may be many causes: The draft may be insufficient or the boiler setting may have cracks in it through which cold air enters, thus chilling the gases before they have come in contact with the heating surfaces. The boiler settings should be as nearly air tight as possible, so that the air will have to pass through the grates and the fire. It may also happen that the draft through the grates is insufficient on account of heavy firing, and good combustion is impossible where fires are carried so thick as to seriously obstruct the draft.



Correct and Incorrect Ways to Maintain a Fire

One method of firing often met with is shown at *A* in the accompanying illustration. The fuel is piled on the grates so long as there is space for it between them and the boiler shell. As very little air can get through such a fire, it is clear that poor combustion will result. At *B* is shown another method too often met with. In this case the fireman wants coal where it can be seen, and yet not be consumed fast enough to require frequent replenishing. In this case what little heat that is produced by the combustion of the fuel is taken up by the great volume of cold air passing up through the bare grates between the fuel bed and the bridgewall. A good fireman will maintain the fire in a thin even bed, as shown at *C*, keeping the grates free from ashes and clinker, so that air may pass freely through the grates and fuel bed. Since there is a difference of about 69 per cent. in favor of the last mentioned method over the others, it would seem that the fireman does not always receive the consideration due him.

When fresh fuel is added to a fire it is best to place the first shovelfuls well up toward the bridgewall, and fire from this point toward the furnace doors. Thus, the fire is not obscured by the dense smoke from the fresh fuel, and the fireman can see just where the fuel is placed.

When cleaning the fire of a boiler be sure to have the other boilers on the same load doing their best. Never allow all fires to become dirty at the same time. When ready to clean the fire under one boiler cut down on the feed water so that the boilers under which the fires are not being cleaned will be able to carry, without unnecessary forcing, the additional load thus temporarily thrown upon them. It is much better to arrange the intervals of cleaning for the different boilers so that there will be plenty of time for building up the new fire, to avoid great changes in pressure and temperature. It often happens that poor judgment in this matter is responsible for a heavy drop of pressure, and that a number of boilers will have their fires cleaned one after the other. In this case as soon as the fires are cleaned it is necessary to begin forcing them, and in a short time they have been ruined and are ready for another cleaning.

There are so many fuels requiring different handling that it is almost useless to give instructions of a general nature as to how to clean fires. The essential point to be kept in mind is to do it in a manner that will cause the least variation of temperature in the boiler. A very good plan with bituminous coals is to build a bank on one side of the grate and add to this, a little at a time, until there is a bank of coked fuel In the meantime the other side is allowed to burn down as low as conditions warrant. Then with the slice bar all clinker should be loosened from the grates and pulled out with the hoe. The bank of coked fuel may now be shoved over to the clean side of the grates and the work with the slice bar and hoe repeated on the other side and fuel scattered from this side over the clean grates with such additional fuel as judgment warrants.

Time is a very important factor in cleaning fires, and preparation should be made for the task. Every tool that will be required should be placed within handy reach as every moment wasted during the cleaning process means keeping the boiler out of commission just so much longer.

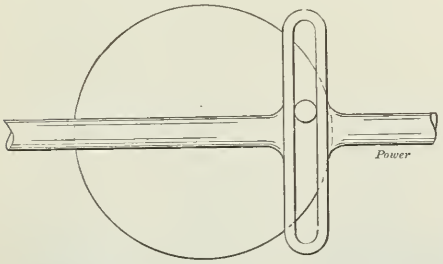
Inequality of Piston Displacement

By Guy Wise

> *In the present analysis of the problem of unequal piston displacement due to the angularity of the connecting rod, the treatment is as simple as possible consistent with a clear understanding of the facts; the elementary case of the crank and slotted yoke serving as an introduction to the subject.*

While the crank and connecting rod as used for changing reciprocating to rotative motion is a simple mechanism, its use introduces some peculiar problems worthy of study, one of which is the variation of piston displacement. This problem has probably been made familiar to most engineers in the statement that the piston travel is greater for the first quarter revolution of the crank travel than it is for the second quarter.

If the crosshead is driven directly by the crank then the conditions are represented by the crank and slotted yoke shown in Fig. 1. With this device the movement of the crosshead corresponding to any quarter turn of the crank is of the same amount. This may be proven and the displacement of the crosshead determined for a movement of the crank through any angle, as follows:



F 1. YOKE AND SLIDING PIN

In Fig. 2 the crank B of length X, has started from A and has mov through the angle $A B P$ (for convenien call this angle B), while at the same time the slotted yoke has moved from A to N. It is evident that

$$A N = A B - B N. \qquad (1)$$

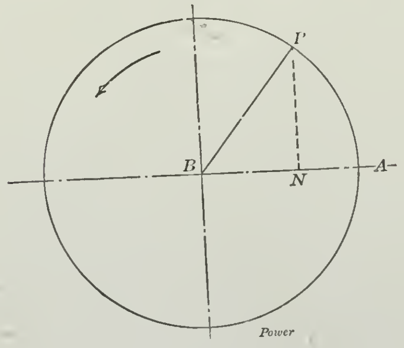


FIG. 2. RELATIVE TRAVEL OF CRANK PIN

Also in the right triangle $B N P$,

$$\text{Cosine}^* \ B = \frac{B N}{B P}$$

From which, $B N = B P$ *cosine* B.

Now, $A B$ and $B P$ are equal. Substituting these known values in equation 1, it becomes

$$A N = B P - B P \text{ cosine } B = B P \ (1 - \text{cosine } B). \qquad (2)$$

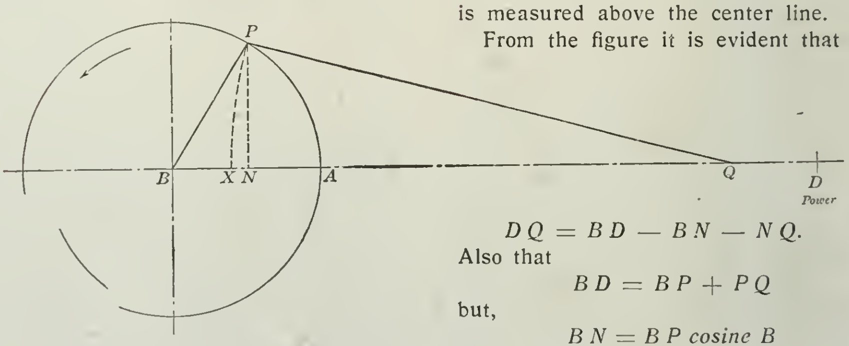


FIG. 3. DIAGRAM REPRESENTING CRANK AND CONNECTING ROD

By means equation 2, it is possible to determine displacement of the crosshead for a particular position of the crank. For example, the displacement when the crank has moved through an angle of 60 degrees is

$$A N = B P \ (1 - \text{cosine } 60 \text{ degrees}).$$

The value of the cosine of 60 degrees is found, in the table of natural cosines printed in all engineering handbooks, to be 0.5. Therefore

$$A N = B P \ (1 - 0.5) = 0.5 \ B P.$$

When the crank is vertical or has moved through an angle of 90 degrees, then

$$A N = B P \ (1 - \text{cosine } 90 \text{ degrees}).$$

The cosine of 90 degrees is found to be zero, hence

$$A N = B P \ (1 - 0) = B P.$$

The results just determined for the crank and slotted yoke show that between 60 and 90 degrees the space traveled by the crosshead is the same as from 0 to 60 degrees, and that when the crank is at mid-position the crosshead is also at mid-position.

The above discussion gives some idea of the action in the case of the crank and slotted yoke and is preliminary to taking up the case of the crank and connecting rod shown diagrammatically in Fig. 3. In this sketch the crank $B P$ has started from A and has moved through the angle $A B P$ (denoted by B), the connecting rod $P Q$ has moved through the angle $N Q P$ (denoted by Q), and has drawn the crosshead from D to Q. The displacement $D Q$ of the crosshead is measured from D, the crank angle B is measured from A and the connecting-rod angle Q is measured above the center line.

From the figure it is evident that

$$D Q = B D - B N - N Q. \qquad (3)$$

Also that

$$B D = B P + P Q$$

but,

$$B N = B P \text{ cosine } B$$

and

$$N Q = P Q \text{ cosine } Q.$$

Substituting these values in equation 3 it becomes

$$D Q = B P + P Q - B P \text{ cosine } B - P Q \text{ cosine } Q.$$

Rearranging the terms gives,

$$D Q = B P \ (1 - \text{cosine } B) + P Q \ (1 - \text{cosine } Q). \qquad (4)$$

Equation 4 is the means of calculating the piston displacement for any crank angle. A comparison of equations 2 and 4 shows that the first term, $B P$ (1 —

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*In a right triangle the ratio of the leg adjacent to any angle to the hypotenuse (the side opposite the angle) is the cosine of the angle in question.

cosine B), of equation 4, is the same as the displacement for the crank and slotted yoke given by equation 2; so that the last term of equation 4, PQ (1 — *cosine Q*), must represent the amount of the inequality of travel due to the use of the crank and connecting rod, the inequality varying in value as the angle Q varies. Graphically this inequality equals the distance XN. This is therefore what is meant by saying that the inequality between the head-end and crank-end displacements of the piston is due to the angularity of the connecting rod of a reciprocating engine.

Thermometer Correction Curves

By Frederick Ray

> *With an ordinary mercurial thermometer and thermometer well it is difficult to obtain accurate readings owing to the fact that a large portion of the thermometer stem projects out of the well and is exposed to the temperature of the air. The accompanying set of curves shows the corrections to be added for various immersions of the thermometer stem and various temperatures.*

The ordinary mercurial thermometer is graduated when entirely submerged; hence, to obtain the correct temperature reading with such a thermometer it is necessary to submerge it until the whole of the mercury in the bulb and in the stem is surrounded by the fluid whose temperature is desired. In practice this is often impossible, especially when using thermometer wells in taking temperatures of fluids in pipes. These wells are usually only a few inches deep, whereas the thermometers are from 12 to 18 inches long; consequently a large proportion of the graduations are outside the well and exposed to the temperature of the surrounding air. This results in the exposed part of the stem, as well as the part of the mercury column contained within it, being at the temperature of the air instead of at the temperature of the well.

As the expansion of the glass due to the temeprature change is only about one-seventh that of the mercury, it follows that the volume of the glass capillary tube varies about one-seventh of that of the mercury column contained within the tube for any given change in temperature. Therefore, the length of the mercury column must increase or decrease as the temperature of the stem is increased or decreased. It is evident that if the indicated temperature of the stem is different from that which would result from complete immersion, a correction must be made to the readings to obtain the correct temperature at the bulb. Following is the method of calculating such corrections and the results for various immersions and temperatures are given in the accompanying set of curves.

The mean coefficient of expansion of mercury varies about ½ per cent. in the range covered by the set of curves which have been computed, using a coefficient of 0.000102 as an average value; this

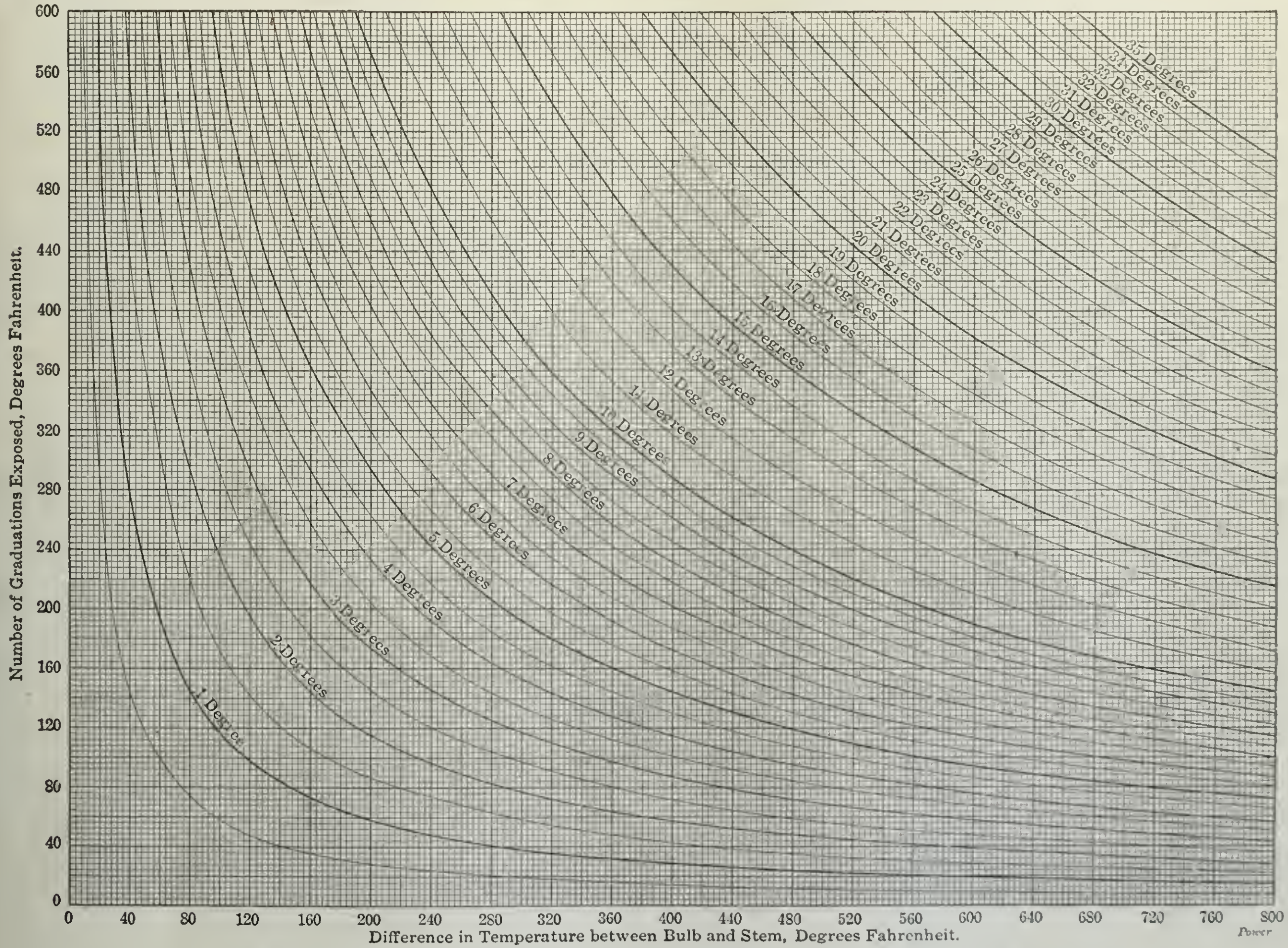


FIG. 1. CORRECTION CURVES FOR VARIOUS IMMERSIONS OF THE THERMOMETER

makes the maximum error not over ¼ of 1 per cent. Considering the mean cubical expansion of glass as 0.0000147, the reading on the Fahrenheit scale may be expressed by the formula:

$$\theta = 0.0000873\ X\ Y,$$

where

θ = Number of degrees correction to be made to the scale reading.

X = Number of graduations exposed to the air.

Y = Number of degrees difference between the indicated temperature of the bulb of the thermometer and that of the air or the stem of the thermometer.

If the stem is cooler than the bulb, as is most always the case, the correction is added; if hotter, it is subtracted. By assuming certain values for θ and solving the equation, the various points in the corresponding curves were obtained.

As an example in the use of the curves, refer to Fig. 2, which shows the cross-section of a steam pipe into which is screwed the ordinary mercury well containing a mercurial thermometer. The column of mercury in the stem of this thermometer stands at 370 degrees and the top of the well extends to the graduation corresponding to 180 degrees, leaving 190 degrees of the stem exposed to the air. A second thermometer suspended beside the first one shows that the temperature of the air surrounding the stem is 70 degrees. Hence, the difference between the temperature of the well and that of the air is 300 degrees. Referring to the curves, the intersection of the horizontal line corresponding to 190 degrees exposed, with the vertical line corresponding to 300 degrees difference in temperature, is found to be on the curved line marked 5 degrees. This means that a correction of 5 degrees should be added to the thermometer reading, making the correct temperature 375 degrees Fahrenheit instead of 370 degrees.

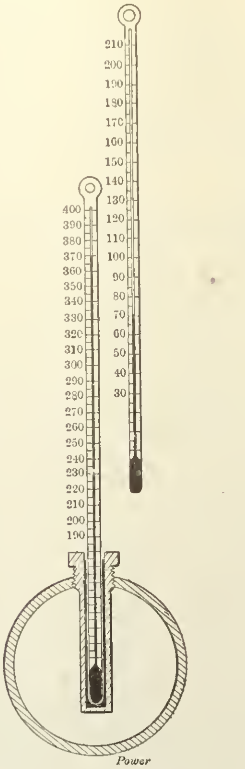


Fig. 2. Thermometer Inserted in Steam Pipe

The curves have been drawn for the Fahrenheit scale and will not apply directly to Centigrade readings. Such readings can be reduced to the Fahrenheit scale, the corrections obtained and then changed back again. The same result, however, can be obtained in a simpler manner by taking the Centigrade readings and finding a correction from the curve, then by multiplying this correction by 1.8 the result obtained will be the desired correction expressed in degrees Centigrade. In the example given, if the thermometer readings are assumed to be degrees Centigrade, and these readings are applied to the curve just as before, the 5 degrees correction must be multiplied by 9/5, which gives 9 degrees Centigrade as the actual correction.

Measuring and Expressing Vacuum

By R. M. Neilson

> *Instead of expressing the vacuum in inches of mercury and then referring to the barometer to compute the absolute pressure, it is common in Great Britain to correct the vacuum reading to apply to a 30-inch barometer and thus give the corrected reading only. The custom of expressing vacua as percentages is not to be recommended.*

It is usual in steam-engine practice, as well as in other industrial connections, to express pressures below atmosphere as "inches of vacuum." By this is meant that the pressure in question is less than atmosphere by an amount equal to the pressure given by a column of mercury of a certain hight measured in inches. For example, by, say, 25 inches of vacuum is meant a pressure which is less than atmospheric pressure by an amount equal to the pressure given by a column of mercury 25 inches high.

In the accompanying illustration, *V* is a vessel, the pressure within which is less than atmospheric; *A* is a tube, the upper end of which is in communication with the interior of *V*, while the lower end is immersed in mercury contained in a dish *C*, the surface of the mercury being exposed to atmospheric pressure. The mercury stands at a hight in the tube of 25 inches above the level of the mercury in the dish, so that this 25-inch column of mercury, plus the pressure in the vessel *C*, balances the atmospheric pressure. There is in this case said to be a 25-inch vacuum in the vessel *V*.

This method of expressing vacuum is sufficiently good for rough measurements, when the vacuum is not very high, that is, the pressure not very low; but it will be seen that it does not definitely express the pressure, but only the difference between the pressure and atmospheric pressure, and the latter is not constant.

With steam turbines, which benefit greatly by high vacuum, it is desirable to know the absolute pressure in the condenser and at the exhaust end and other places in the turbine, as the temperature of the steam, and likewise the work obtainable from the steam, depend on the

INCHES AND PERCENTAGE OF VACUUM.

Absolute Pressure in Lb. per Sq.In.	Vacuum in Inches of Mercury; Barometer at 30 In.	Percentage Vacuum; Barometer at 30 In.	Absolute Pressure in Lb. per Sq.In.	Vacuum in Inches of Mercury; Barometer at 30 In.	Percentage Vacuum; Barometer at 30 In.	Absolute Pressure in Lb. per Sq.In.	Vacuum in Inches of Mercury; Barometer at 30 In.	Percentage Vacuum; Barometer at 30 In.
0.0000	30.0	100	1.670	26.6	88⅔	6.631	16.5	55
0.0491	29.9	99⅔	1.719	26.5	88⅓	6.876	16.0	53⅓
0.0982	29.8	99⅓	1.768	26.4	88	7.122	15.5	51⅔
0.1474	29.7	99	1.817	26.3	87⅔	7.368	15.0	50
0.1965	29.6	98⅔	1.866	26.2	87⅓	7.613	14.5	48⅓
0.2456	29.5	98⅓	1.916	26.1	87	7.859	14.0	46⅔
0.2947	29.4	98	1.965	26.0	86⅔	8.104	13.5	45
0.3438	29.3	97⅔	2.063	25.8	86	8.35	13.0	43⅓
0.3929	29.2	97⅓	2.161	25.6	85⅓	8.595	12.5	41⅔
0.4421	29.1	97	2.259	25.4	84⅔	8.841	12.0	40
0.4912	29.0	96⅔	2.358	25.2	84	9.087	11.5	38⅓
0.5403	28.9	96⅓	2.456	25.0	83½	9.332	11.0	36⅔
0.5894	28.8	96	2.554	24.8	82⅔	9.578	10.5	35
0.6385	28.7	95⅔	2.652	24.6	82	9.823	10.0	33⅓
0.6876	28.6	95⅓	2.751	24.4	81⅓	10.07	9.5	31⅔
0.7368	28.5	95	2.849	24.2	80⅔	10.31	9.0	30
0.7859	28.4	94⅔	2.947	24.0	80	10.56	8.5	28⅓
0.8350	28.3	94⅓	3.045	23.8	79⅓	10.81	8.0	26⅔
0.8841	28.2	94	3.143	23.6	78⅔	11.05	7.5	25
0.9332	28.1	93⅔	3.242	23.4	78	11.30	7.0	23⅓
0.9823	28.0	93⅓	3.340	23.2	77⅓	11.54	6.5	21⅔
1.031	27.9	93	3.438	23.0	76⅔	11.79	6.0	20
1.081	27.8	92⅔	3.684	22.5	75	12.03	5.5	18⅓
1.130	27.7	92⅓	3.929	22.0	73⅓	12.28	5.0	16⅔
1.179	27.6	92	4.175	21.5	71⅔	12.52	4.5	15
1.228	27.5	91⅔	4.421	21.0	70	12.77	4.0	13⅓
1.277	27.4	91⅓	4.666	20.5	68⅓	13.02	3.5	11⅔
1.326	27.3	91	4.912	20.0	66⅔	13.26	3.0	10
1.375	27.2	90⅔	5.157	19.5	65	13.51	2.5	8⅓
1.421	27.1	90⅓	5.403	19.0	63⅓	13.75	2.0	6⅔
1.474	27.0	90	5.648	18.5	61⅔	14.00	1.5	5
1.523	26.9	89⅔	5.894	18.0	60	14.24	1.0	3⅓
1.570	26.8	89⅓	6.140	17.5	58⅓	14.49	0.5	1⅔
1.621	26.7	89	6.385	17.0	56⅔	14.74	0.0	0

absolute pressure, quite irrespective of the pressure of the atmosphere.

The power of custom is, however, so great that engineers still stick to the mode of expressing low pressures as "inches of vacuum," but to enable the absolute pressure to be calculated, it is usual, with low-pressure readings, to mention the hight of the barometer when recording the "inches of vacuum." If the vacuum in a condenser is said to be 28 inches and the barometer to be at 30.4, then we know that the absolute pressure within the condenser is

$$30.4 - 28 = 2.4$$

inches of mercury. Instead of giving the two figures, or two sets of figures, one referring to the vacuum and the other to the barometer, it is common in Great Britain to correct the vacuum reading to apply to a 30-inch barometer and to give the corrected reading only. For example, if the vacuum were 28 inches and the barometer 30.4 inches, the corrected vacuum reading would be

$$28 - (30.4 - 30) = 27.6.$$

The correction can usually be performed mentally. A high vacuum should not be expressed in inches of vacuum without mention being made of the barometric hight, unless a correction for a 30-inch barometer has been made.

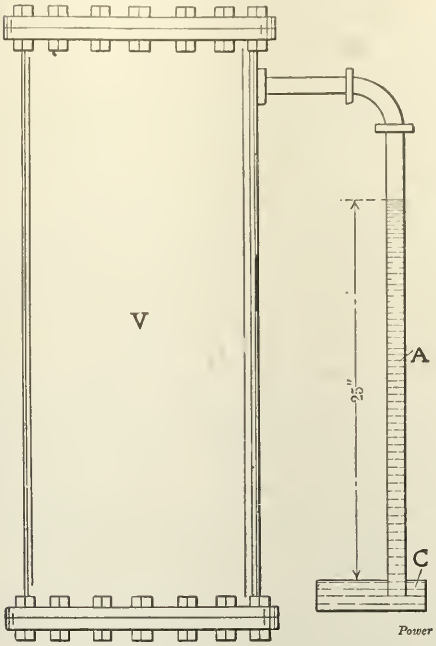
Sometimes vacua are expressed as percentages. If the vacuum is, say, 27 inches with a 29-inch barometer, it is spoken of as a 93.1 per cent. vacuum, 27 inches being 93.1 per cent. of 29 inches. It is often specified in Europe that the steam consumption of a steam turbine must not exceed a certain amount with a certain percentage vacuum. This practice is not, however, to be recommended. A vacuum of 96 per cent., for example, with the barometer at 28 inches represents an absolute pressure of

$$\frac{4}{100} \times 28 = 1.12$$

inches of mercury, or 0.55 pound per square inch; whereas, with a 31-inch barometer, the same percentage of vacuum represents a pressure of

$$\frac{4}{100} \times 31 = 1.24$$

inches of mercury, or 0.594 pound per square inch, a difference of nearly a tenth of an inch of mercury, which will affect the steam consumption of a turbine from a half per cent. to one per cent. When the vacuum in a guarantee of steam consumption is specified as a percentage it will naturally be to the turbine builder's advantage to arrange, if possible, for the official tests to take place under a low barometer.



MEASURING VACUUM IN TANK

In the accompanying table are given the inches and percentage of vacuum, both for a 30-inch barometer, corresponding to pressures from zero to atmospheric pressure. For example, if in a vessel there is an absolute pressure of, say, 1.474 pounds per square inch, a mercury column in communication with the vessel should indicate a vacuum of 27 inches, if the barometer is at 30 inches. If the barometer is not at 30 inches, the mercury column would not indicate 27 inches, but its reading after correction for the barometer would be 27 inches.

It has already been said that the absolute pressure is important in connection with the performance of a steam turbine. The atmospheric pressure, on the other hand, has no effect on the performance except that it may slightly influence the chances of air leakage. With air pumps the case is, however, different. An air pump is affected by the absolute pressure in the condenser, which largely determines the necessary capacity of the pump; it is also, however, affected by the atmospheric pressure which determines, first, the work required to be done in compressing and delivering the air and, second, the effect of reëxpansion of any undischarged air within the pump cylinder. In a steam-consumption guarantee test of a turbine, when the vacuum is specified as a percentage, it was suggested that it was to the advantage of the turbine manufacturer to have the test run under a low barometer. In the case of a guarantee test of an air pump, when the vacuum is specified as a percentage, it will be to the advantage of the air-pump manufacturer—or at least the individual or firm delivering the air pump—to have the test run under a low barometer, because the work of compression and the reëxpansion effect, if any, are both nearly proportional to the ratio of compression, i.e., to the percentage of vacuum; but the amount of air which can be delivered per stroke depends on the admission density of the air which, with a given percentage of vacuum, will be greater with a high than with a low barometer.

Rotary and Other Air Compressors

By F. R. Low

Pokorny & Wittekind, of Frankfort-on-the-Main, Germany, make machines for compressing air and pneumatic tools for using the air after it has been compressed. They use an ingenious arrangement of trunk piston wherein the annular space surrounding the trunk is the high pressure and the other end of the cylinder the low.

The air passes from the low- to the high-pressure cylinder through a vertical U-tube intercooler mounted on top of the cylinder and often the largest feature of the machine which, on account of the absence of piston rod and guides, is very compact. The pistons are packed with cast-iron rings, the joint being made by lapping the ends over a block which has been left in the ring groove, as in Fig. 2.

They were just assembling at the time of our visit a four-stage machine to compress air to a pressure of 1455 pounds gage with exhaust steam of atmospheric pressure. The pressures for the several stages are

37: 162:: 470: 1455.

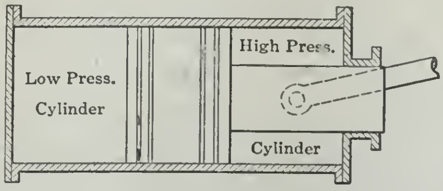


FIG. 1

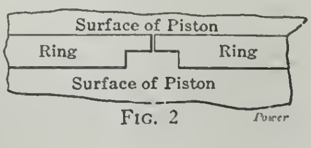


FIG. 2

The power required is 190 horsepower, and the single steam cylinder has a diameter of 44 inches and a stroke of 20 inches. The capacity is 388 cubic feet per minute of free air.

Another and still more remarkable machine was a rotary compressor designed to compress 1,236,000 cubic feet of free air per hour to a pressure of 117.6 pounds in 12 stages. It runs at 4000 revolutions per minute. This is an exceptionally high pressure for a rotary.

High Speed Compound Engine Design

By J. D. Speedium

> *The complete design, including all computations and working drawings, of a 500-horsepower, high-speed, vertical, compound engine is taken up and represents the latest British practice in this line. The present article carries the design up through the cylinders, and subsequent instalments will take up the other parts of the engine in order.*

In this series of articles, the design and construction of a modern high-speed, inclosed, forced-lubrication steam engine is dealt with in every detail, and in such a manner as to clearly demonstrate present-day British practice. Commencing with the specifications for a 500-horsepower engine, the design of the engine is next dealt with and a complete set of working drawings included.

SPECIFICATIONS

The engine is to be capable of developing 500 indicated horsepower, and to carry 25 per cent. overload when supplied with steam at a pressure of 150 pounds per square inch, exhausting into a condenser having a vacuum of 26 inches of mercury, and to run at a speed of 300 revolutions per minute.

Cylinders: The cylinder castings are to be made of hard close-grained cast iron. Each cylinder with its valve box is to be a separate casting, designed to eliminate, as much as possible, distortion due to the temperature of highly superheated steam. Top and bottom covers are to be accurately machined and polished all over the inside surfaces, and clearances are to be reduced to a minimum. The valve boxes are to have liners fitted to them. Relief valves and drain cocks must be of ample size to be fitted. The cylinders and covers are to be covered with a thick layer of nonconducting composition, afterward lagged with planished steel.

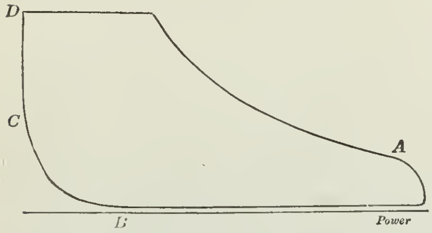


Fig. 1. Ideal Diagram

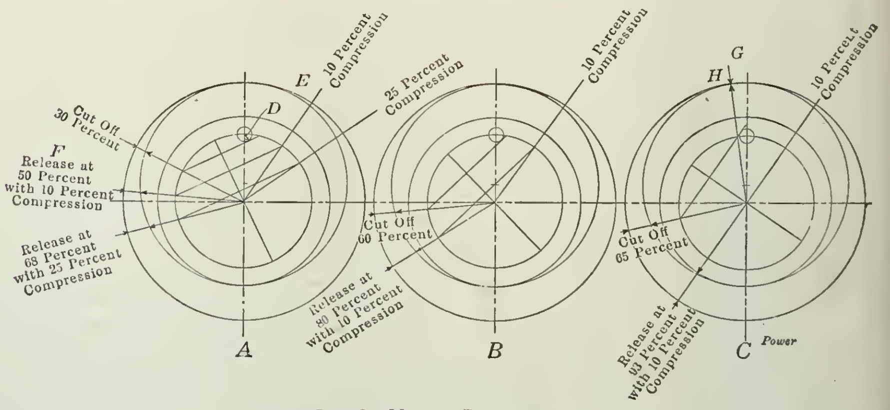

E
D
10 Percent Compression
25 Percent Compression
Cut Off 30 Percent
F
Release at 50 Percent with 10 Percent Compression
Release at 68 Percent with 25 Percent Compression
A
10 Percent Compression
Cut Off 60 Percent
Release at 80 Percent with 10 Percent Compression
B
G
H
10 Percent Compression
Cut Off 65 Percent
Release at 93 Percent with 10 Percent Compression
C
Power

Fig. 2. Valve Diagrams

The material used must have a tensile strength of not less than 28,000 pounds per square inch, and a bar, 2x1 inch by 3 feet 6 inches, cast from the same ladle, must be capable of withstanding a load of 3800 pounds at the center, when placed upon supports 3 feet apart, and have a deflection at the center of not less than 3/8 inch.

Valves: The valves are to be of the piston type and of cast iron, fitted with solid floating rings having water grooves cut on the face not more than 2 inches apart. The rings must be secured by means of junk plates to allow of the end play being taken up.

Frame or Trunk: The working parts of the engine are to be inclosed in a trunk strongly ribbed and made of cast iron. Doors are to be provided opposite each crank at the front and back of the engine to give easy access to all working parts. The trunk is to be securely bolted to the bed by closely pitched studs so as to make an oil-tight joint.

Bed: The bed is to be of massive design and provided with doors allowing easy access to the oil pump and strainer. The material used for the trunk and the bed must have a tensile strength of not less than 18,000 pounds per square inch, and the maximum load of the engine must not stress any part more than 500 pounds per square inch.

Distance Piece and Crosshead Guide: Each cylinder is to be carried on a cast-iron distance piece with which must be formed the crosshead guide. This casting must also form the bottom cylinder cover. A separate gland must be fitted at the top of the trunk to prevent leakage of water from the glands getting into the crank chamber. The crosshead guides are to be of the bored form.

Main Bearings: The main bearings are to be of ample size, fitted with cast-iron bushes lined with antifriction metal. The pressure per square inch as calculated from the indicator diagram, when the engine is developing full load must not exceed 200 pounds per square inch on the bearing.

Piston Rods and Crossheads: The piston rods and crossheads must be forged solid and made of Siemens-Martin acid steel having a tensile strength of not less than from 76,000 to 84,000 pounds per square inch, and an elongation of not less than 20 per cent. in 2 inches. The bushes are to be of phosphor bronze and the slippers of cast iron.

Connecting Rods: The connecting rod is to be forged from Siemens-Martin acid mild steel having a tensile strength of not less than from 56,000 to 64,000 pounds per square inch and an elongation of not less than 25 per cent. in 2 inches. It is to be formed with a jaw at the top end, into which a case-hardened crosshead pin will be shrunk. The bottom end is to be fitted with brass bushes lined with antifriction metal.

Valve Gear: The valves are to be operated direct from the crank shaft by means of eccentrics, the eccentric straps being lined with antifriction metal. The eccentric-rod pins must be case-hardened, and the valve-spindle crossheads fitted with adjustable brasses.

Pistons: The high-pressure piston is to be of cast iron, and the low-pressure piston of wrought steel having a tensile strength of not less than from 76,000 to 84,000 pounds per square inch.

Crank Shaft: The crank shaft is to be forged solid with the cranks placed at 180 degrees, of Siemens-Martin acid steel having a tensile strength of not less than 56,000 to 64,000 pounds per square inch and an elongation of not less than 27 per cent. in 2 inches.

Flywheel: The flywheel is to be of plate form, solid and of cast iron, having a tensile strength of not less than 18,000 pounds per square inch. It is to be left a tight fit and forced onto the crank shaft by hydraulic pressure. The weight of the rim is to be such that the cyclic variation does not exceed 1/500. A rack is to be cast in the rim so that the engine may be barred round in either direction.

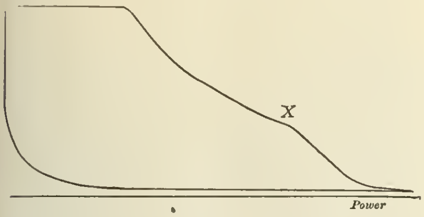


FIG. 3. TOO EARLY RELEASE

Governor: The governor is to be of the centrifugal type fixed on the crank shaft, and acting, as directly as possible, upon a throttle valve; the throttle valve to be of the double-seated balanced type.

Lubrication: This is to be of the forced type, all parts, including the governor, to be automatically lubricated from a valveless pump driven from one of the eccentrics actuating the valves.

The first points to decide are the size of cylinders. Actual experience has shown that the most economical size of cylinders for any given power is less than what would be expected from purely theoretical considerations. The most economical mean effective pressures for various steam pressures for simple-, compound- and triple-expansion engines both condensing and noncondensing are given in Table 1.

TABLE 1. ECONOMICAL MEAN EFFECTIVE PRESSURES.

Pressure in Pounds per Square Inch.	Simple Engines.		Compound Engines.		Triple-expansion Engines.	
	Condensing.	Non-condensing.	Condensing.	Non-condensing.	Condensing.	Non-condensing.
50	26	30				
75	31	35	27	29		
100	36	40	30	34		
125			34	39	29	36
150			38	44	32	40
175			40	49	34	44
200			42	54	37	48

In the present instance the pressure being 150 pounds per square inch and the engine running condensing, the most economical mean effective pressure will be 38 pounds, referred to the low-pressure piston. The area of the cylinder may now be found by the following formula:

$$A = \frac{I.H.P. \times 33{,}000}{M.E.P. \times P.S.}$$

where,

A = Area of low-pressure piston,

$I.H.P.$ = Indicated horsepower of the engine,

$M.E.P.$ = Mean effective pressure,

$P.S.$ = Piston speed.

The piston speeds usually allowed in practice are given in Table 2.

TABLE 2. PISTON SPEEDS.

Horsepower of Engine.	Speed in Revolutions per Minute.	Piston Speeds in Feet per Minute.
		Not exceeding
50 to 200	600 to 500	650
200 to 500	500 to 325	750
500 to 1000	375 to 250	800
1000 to 3000	300 to 175	800 to 1000

In the present case the piston speed may be taken at 700 feet per minute and the area of the low-pressure cylinder will be

$$\frac{500 \times 33{,}000}{38 \times 700} = 620 \text{ square inches}$$

which represents approximately 28 inches diameter; and as the piston speed is 700 feet per minute, the stroke will equal

$$\frac{700}{(2 \times 300)} = 1.16 \text{ feet}$$

say, 14 inches stroke. The size of the high-pressure cylinder must now be found. The ratio of high- to low-pressure cylinders must be such that the cutoff in the high-pressure cylinder is about 0.6 of the stroke, when the engine requires nearly boiler pressure, about 145 pounds, to develop the maximum load. To fix the ratio of cylinders it is necessary to determine the number of times the steam must be expanded to give a mean effective pressure of 38 pounds under normal conditions and 47.5 pounds at 25 per cent. overload.

By referring to a steam-expansion table and allowing for a back pressure in the cylinders of three pounds and a diagram factor of 72 per cent., the number of expansions will be found to be 6½. This may be easily checked by the following formula:

$$M.E.P. = \frac{P \times (1 + hyp.\ log\ R) \times D}{R} - B$$

where,

$M.E.P.$ = Mean effective pressure,

P = Absolute pressure in pounds per square inch,

R = Number of expansions,

D = Diagram factor for type of engine as follows:

Simple engines	78 per cent.
Compound engines	72 per cent.
Triple-expansion engines	65 per cent.

B = Back pressure in pounds per square inch.

In the present example

$M.E.P.$ = 47.5 pounds,

P = 145 + 15 = 160 pounds absolute,

D = 72 per cent.,

B = 3 pounds,

and,

$$\frac{1 + hyp.\ log.\ 6\frac{1}{2}}{6\frac{1}{2}} = 0.44;$$

therefore,

$$(160 \times 0.44 \times 0.72) - 3 = 47.6 \text{ pounds}$$

as the mean effective pressure required.

Taking the mean cutoff in the high-pressure cylinder at 61.5 per cent. of the stroke and allowing for a ratio of expansion of 6.5, as already determined, it will be found that a ratio of cylinder areas of

$$0.615 \times 6.5 = 4 \text{ to } 1$$

will be required. The diameter of the high-pressure cylinder will therefore be 14 inches.

It may appear, at first sight, that the cutoff arranged for is very late in the stroke to give an economical engine, but that is not the case, and as the valve setting is of vital importance if maximum economy is desired, this question will be dealt with at some length.

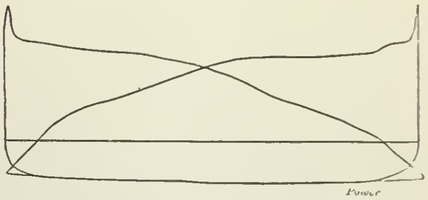


FIG. 4. BEFORE ADJUSTING VALVE SETTING

THE INDICATOR DIAGRAM CONSIDERED IN CONNECTION WITH VALVE SETTING

The valves of an engine if carefully set by an experienced engineer will, as a rule, produce a reasonable indicator diagram.

The indicator diagram which corresponds most closely to the theoretical figure is usually looked upon as the most perfect, and the setting which produces this result is regarded as the ideal one. This is correct to a certain extent, and is applicable to engines having separate steam and exhaust valves, but where only one valve is used for both steam admission and exhaust, the best looking diagram does not always give the best results.

Providing a valve setting gives a good running engine without knocks or pounding, the most important point to consider is economy in steam consumption. Consider for a moment the elementary points in connection with the steam distribution to an engine cylinder.

In Fig. 1 is shown what would be called an ideal diagram from a simple noncondensing engine. If the cylinder has separate valves for steam and exhaust (as in a Corliss engine for instance, with the valves driven through a wristplate) giving efficient opening and closing to valves, it is not difficult to produce a very similar diagram. Steam can be admitted just before the piston arrives at the end of the stroke, and as a quick opening is given to the valve by the wristplate, there is no difficulty in obtaining a good steam line closely approximating to the boiler pressure right up to the point of cutoff, when, owing to the almost instantaneous closing of the valve, a sharp corner is obtained. Again, the expansion may be carried to almost the very end of the stroke, and the exhaust valve then opened quickly to save loss due to back pressure or excessive rounding of the corner at *A*. When the clearance volume is small, compression need not commence before the point *B* shown on the diagram, and with a small lead, as already referred to, the vertical line *C D* giving full pressure at the commencement of the stroke is easily obtained.

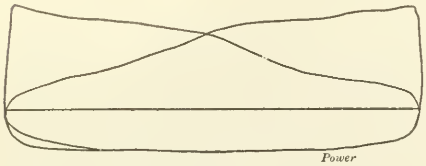


Fig. 5. After Adjusting Valve Setting

It should be noted here that the best results as regards economy are always obtained by expanding the steam to a fairly large degree, and even in compound and triple-expansion engines with cylinders of high ratio, these results can be btained only when cutting off steam early in the stroke. Early cutoff, combined with a late release, can very conveniently be arranged for in two-valve engines, but not so in the case of single-valve engines when driven by an eccentric.

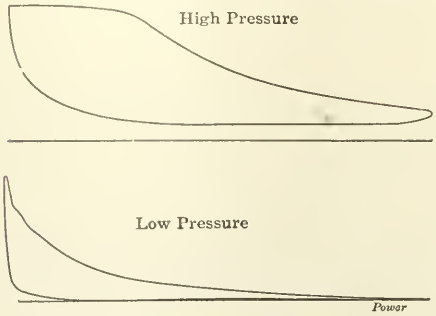


Fig. 7. Late Release in Drop-valve Type of Engine

This point may be best illustrated by means of a valve diagram, and for this purpose the Zeuner diagram, combined with a crank and piston diagram, will be used. Fig. 2, *A*, *B* and *C* represent such diagrams. In *A* steam is cut off at 30 per cent. of the stroke, giving the engine a small lead of ¼ inch as shown at *D*, and the angle of the eccentric is at once fixed. Owing to the small clearance volume, it is not advisable to commence compression before 90 per cent. of the stroke has been open to exhaust, as at *E*. This, therefore, fixes the point of release as shown at *F*. Hence, release occurs at 50 per cent. of the stroke, which is very uneconomical, and has the effect of producing an indicator diagram as shown in Fig. 3. Here the release takes place at *X*, and the pressure almost immediately falls, and for the remainder of the stroke little or no work is done by the steam in the cylinder. Comparing the diagram in Fig. 3 with that shown in Fig. 1 it will be readily seen that far more work is obtained from the same quantity of steam by the distribution shown in Fig. 1.

By making the cutoff later, better results as regards release can be obtained, as shown in diagram *B*, Fig. 2. In this diagram the cutoff is shown as taking place at 60 per cent. of the stroke, but even here the release takes place at 80 per cent. of the stroke, if compression is arranged for 10 per cent. This brings us to an important point—Is it advisable to produce a good admission line at the commencement of the stroke by giving a lead to the valve, or is it beneficial to sacrifice lead in favor of release? This

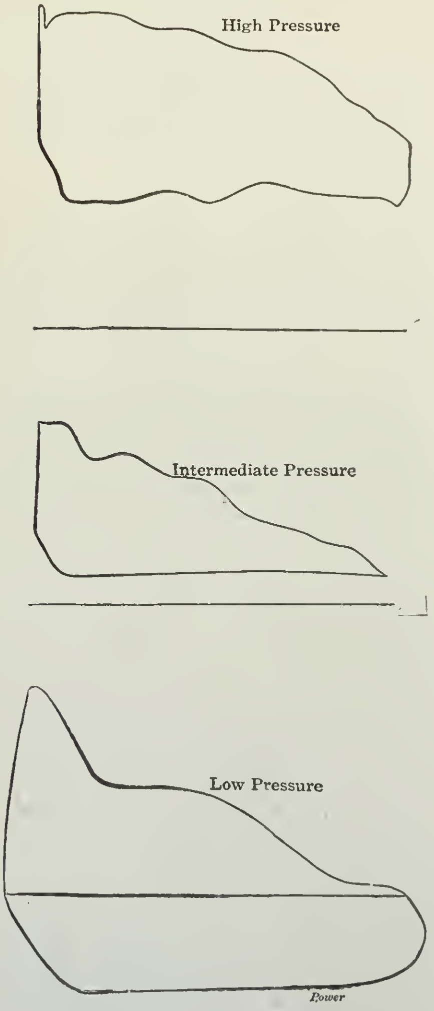


Fig. 6. Diagrams Showing Lack of Lead

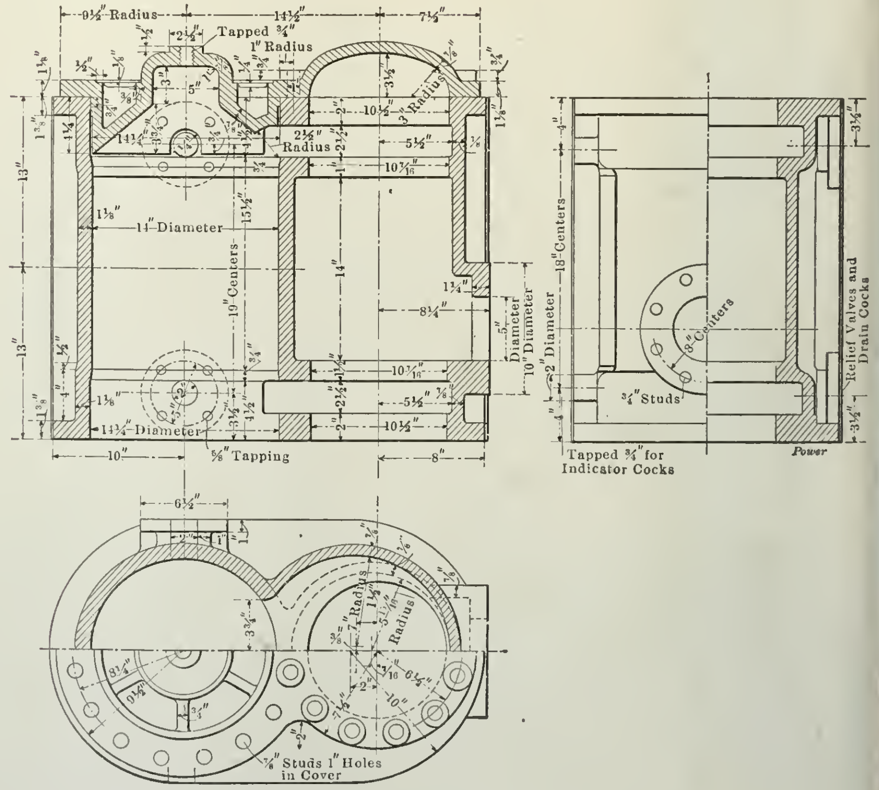


Fig. 8. High-pressure Cylinder

point is not as fully understood as it might be, but owing to the necessity of obtaining maximum economy with high-speed engines, which are all of the single-valve type, driven by eccentrics, it has been fully considered by the builders of those engines, particularly in England.

Diagram *C*, Fig. 2, shows what can be done in this direction. In this diagram the cutoff takes place at 65 per cent. of the stroke, and compression at 90 per cent., but owing to the alteration in the lead of the valve a release at 93 per cent. is obtained. By giving a negative lead to the valve it may be argued that a loss takes place at the commencement of the stroke, but against this must be placed the late release and consequent gain at the end of the diagram.

Actual results are always the best guide and it has been found by experience that considerable advantage is gained by modifying the valve setting similar to that shown in *C*, Fig. 2. It should be noticed that although admission does not commence until *G* is reached by the crank, the piston has moved only through a distance *H* or about 2 per cent. of the stroke.

In confirmation of the above arguments, indicator diagrams are shown in Fig. 4, taken from a high-speed engine of English make with lead given to the valves and consequently early release. In Fig. 5 is shown a set of diagrams from the same engine with valve settings altered to give results similar to those shown in valve diagram *C*, Fig. 2. By altering the setting from Fig. 4 to Fig. 5 a saving of 10 per cent. in steam consumption was obtained.

In Fig. 6 is shown a set of indicator diagrams taken from a triple-expansion high-speed engine. The lack of lead is again very noticeable, but a good release is shown, which no doubt compensates for any defect in the steam-admission line.

There is no doubt that the economical performances of the Corliss drop-valve engine and the drop-piston-valve engine are largely due to the efficient distribution of the steam, and a comparison between the distribution of steam in the various types is therefore instructive. In drop-valve engines the advantages of late release are at once discernible by referring to Fig. 7, which shows a set of diagrams taken from a Sulzer engine. Corliss engines having their valves driven in a similar manner give equally good results.

The details of the valve setting for the engine under consideration will be dealt with fully when dealing with the piston valves.

CYLINDERS

The design of the cylinders may now be proceeded with. In Figs. 8 and 9 are shown detail drawings of the high- and low-pressure cylinders respectively. It will be noticed that they are of very simple design, and as both top and bottom covers are separate, the cylinder casting itself is a very plain one.

The speeds through the inlet and exhaust passages, steam ports, etc., as used in this design, represent current practice.

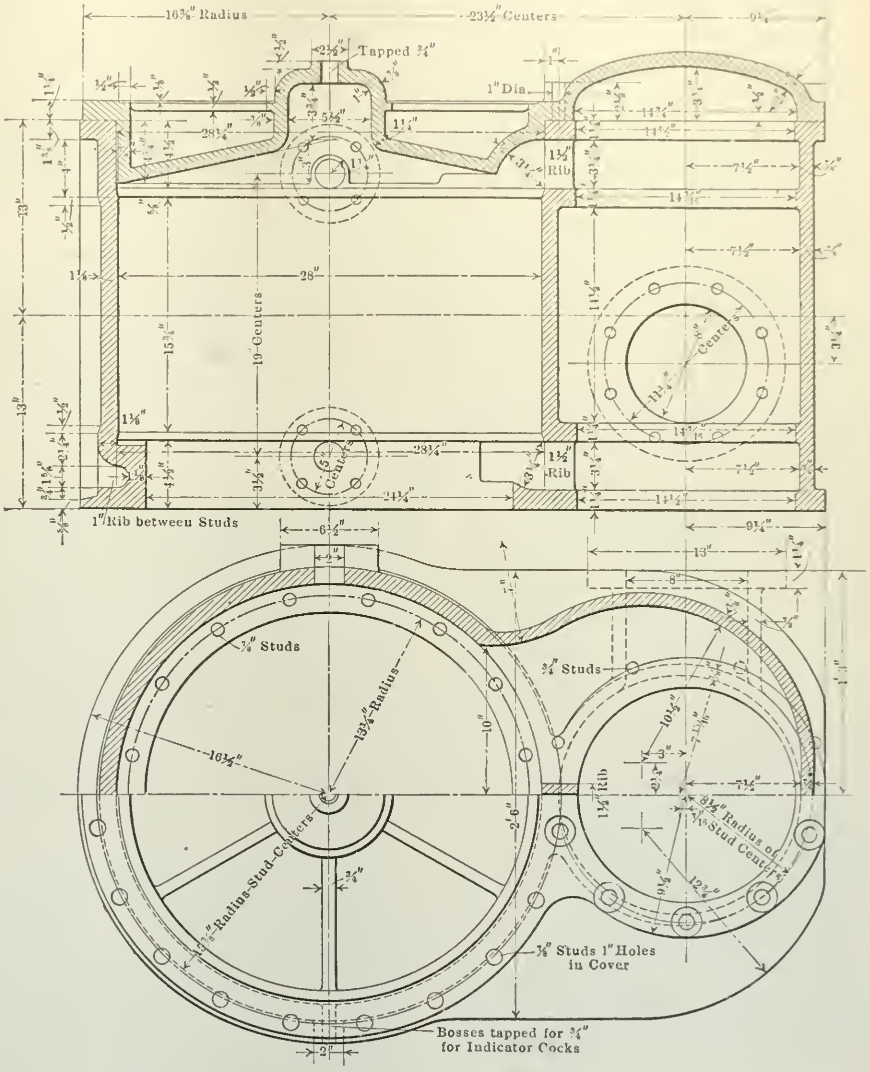


FIG. 9. LOW-PRESSURE CYLINDER

The size of the steam inlet to the high-pressure cylinder should be such that the velocity does not exceed 7000 feet per minute and the exhaust 5000 feet per minute. The inlet to the low-pressure cylinder must be arranged to suit a velocity not exceeding 9000 feet per minute, and the final exhaust from this cylinder should not exceed 5400 feet per minute. The sizes may be found by the formula,

$$\frac{P.S. \times A}{V} = \textit{area of pipe or port in square inches}$$

where,

$P.S.$ = Piston speed in feet per minute,
V = Velocity of steam in feet per minute,
A = Area of cylinder in square inches.

The area of the steam ports may be fixed by this rule, but the following speeds of steam should not be exceeded: High-pressure cylinder, 6000 feet per minute; intermediate-pressure cylinder, 7000 feet per minute; low-pressure cylinder, 8000 feet per minute.

The speed through the ports of low-pressure cylinders of compound engines is taken as for intermediate cylinders of triple-expansion engines.

The steam inlet of the high-pressure cylinder must therefore equal

$$\frac{700 \times 153.9}{7000} = 15.39$$

square inches, approximately 4½ inches diameter, the nearest even size being 5 inches diameter.

The exhaust equals

$$\frac{700 \times 153.9}{5000} = 21.5$$

square inches or approximately 5¼ inches diameter. As, however, the size of the exhaust from the high-pressure cylinder is usually made the same diameter as the inlet to the low-pressure cylinder, this must be determined as follows:

$$\frac{700 \times 615}{9000} = 47.8$$

square inches, representing a diameter of about 7¾ inches, say, 8 inches. This is therefore the diameter of the high-pressure exhaust and the low-pressure steam inlet.

The area of the steam ports of the high-pressure cylinder must be

$$\frac{153.9 \times 700}{6000} = 17.9 \text{ square inches.}$$

The area of the steam ports of the low-pressure cylinders will require to be

$$\frac{700 \times 615}{7000} = 61.5 \text{ square inches}$$

and the exhaust pipe

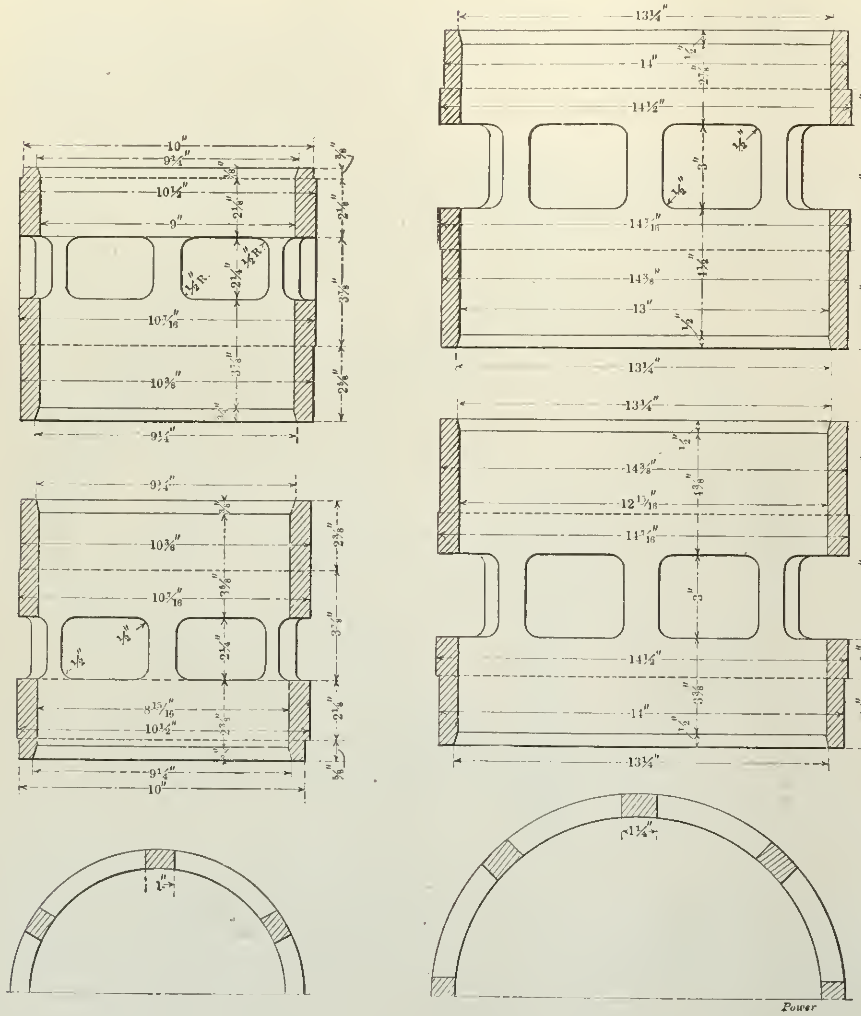


FIG. 10. HIGH- AND LOW-PRESSURE VALVE-CHEST LINERS

$$\frac{700 \times 615}{5400} = 79.7 \text{ square inches,}$$

representing a diameter of approximately 10 inches as a minimum.

In designing the cylinders, the barrel should be kept as free from ribs and as independent of the other parts of the cylinder as possible, particularly in the case of high-pressure cylinders where superheated steam is used. This is with a view to preventing distortion due to unequal expansion of the various parts of the casting at high temperature.

The clearances call for special attention in high-speed engines, owing to the shortness of the stroke. This point will be clearly realized when it is considered that in the case of a slow-speed engine, of the power under consideration, the speed would be about 80 to 90 revolutions per minute, or one-fourth the speed of the high-speed engine. As the piston speed would be about the same, or probably slightly less, it will be seen that the stroke would be about four times that of the high-speed engine. Assuming, therefore, that the steam ports are made the same area in each case, and the clearance between the cover and the piston at the end of the stroke is the same, the total clearance volume would remain the same. Taking the clearance volume of the slow-speed engine at 5 per cent. of the cylinder volume, under similar conditions it would be 20 per cent. of the high-speed engine cylinder volume, and in case of the high-speed engine any loss due to clearance would take place four times as often Under ordinary conditions it is rarely possible to make the clearance less than from 10 to 12 per cent. It should be noted that in the case of high-speed engines, only one port is used for both steam and exhaust, whereas in the case of slow-speed engines the steam is usually admitted and exhausted by separate ports.

The ports must therefore be made very short and straight, the joints of the cylinder cover and valve-chest cover generally fixing the length. As will be seen by referring to the drawings, Figs. 8 and 9, in order to reduce the length of the ports to a minimum, the cylinders and valve-chest covers are cast together. The bottom covers are formed with the casting which carries the cylinders, and the same casting is also extended into the frame to form the crosshead slides.

The general dimensions of the cylinders are fixed by the design of the piston and the valves adopted. These details will be considered separately. It will be noted that liners are fitted to the valve chests, but not to the cylinder barrel. Liners are rarely used for the barrel, and as experience has shown that steam jacketing is useless in this class of engine, there is no necessity for fitting them if the cylinder casting is made of hard close-grained metal. The addition of a liner would also slightly increase the length of the steam ports.

The valve-chest liners are shown in detail in Fig. 10. They are usually made of the same metal as the cylinders, and are forced into place, being finally finished when fitted into the cylinder. No other method of securing them has been found necessary.

The length of the cylinders is determined by the stroke and the type of piston, the clearance between the piston and the covers being kept as small as practicable. In all engines of this class it is usual to machine either the inside of the covers or the faces of the pistons. The clearance allowed varies from ⅛ to ¼ inch according to the size of engine. Relief valves of ample size are usually fitted to the cylinders, Table 3 representing the practice generally followed.

TABLE 3. SIZES OF RELIEF AND DRAIN VALVES.

Diameter of Cylinder, Inches.	Diameter of Relief Valve, Inches.	Diameter of Drain Valve. Inches.
12	1¼	⅝
16	1½	⅝
20	1¾	¾
24	2	¾
30	2½	¾
36	3	1
48	3½	1¼
60	4	1½

The cylinders are lagged with magnesia or asbestos nonconducting material, which is again covered by sheets of planished steel in the usual way, a thickness of from 1½ to 2 inches being allowed for the nonconducting material.

It will be noticed on referring to the high-pressure cylinder drawing that very little steam-chest capacity is allowed for, as the throttle-valve box is bolted direct to the cylinder. This, however, is beneficial for governing, as the steam between the throttle valve and the cylinder is beyond the control of the governor. The same remarks also apply to the steam chest of the low-pressure cylinder.

Owing to the great loss of heat due to radiation and conduction, the most efficiently designed cylinder is the one which occupies the smallest cubical capacity, other things being equal. This feature will be noticed by referring to Figs. 8 and 9, the flanges being only just large enough for sufficient lagging.

In the case of small compound engines up to about 300 brake horsepower, the high- and low-pressure cylinders are frequently cast together with the piston-valve chamber between them, one piston valve distributing the steam to both cylinders. The valve really consists of two valves placed one above the other. This system is applicable only to engines with cranks opposite one another; that is, placed at 180 degrees.

Peculiar Design of Corliss Engine

In one of the steam plants of Augusta, Ga., is a vertical steam engine of a design not frequently met with. It is of about 850 horsepower capacity, has a cylinder 42x42 inches, and runs at a speed of 72 revolutions per minute. Fig. 1 shows the general appearance of the engine, while Fig. 2 is a near view of the valve gear. It may be seen from Fig. 1 that the engine was originally built for reversing, but one of the eccentric rods has been removed, so that the engine now runs in but one direction. The exhaust valves are shown next to the side of the cylinder connecting with the pipe running to the condenser, which is placed on the roof.

Referring to Fig. 2, it may be seen that the stem of the governor, which is located below the platform of the engine, has a protruding stem, to which is attached an arm running to and pivoted on the frame of the engine; another section extends back over the top of the governor stem. On the inner side of this attachment is an extension, and, when in proper position, comes in contact with the trip arms, which are pivoted at a point about the center. The upper ends are fitted with trip blocks, while the lower ends curve out toward the governor. On the valve stems, which extend down below the dashpots, are fitted catch blocks which engage with those secured to the trip arms.

When the engine is in operation the governor stem rises; this lifts the connecting levers and forces the trip plate on the inner side to take a lower position. When it reaches the proper point it comes in contact with the curved ends of the trip arms, forcing them in at the bottom end and out at the top. This releases the catch blocks and the valve is then closed by the action of the dashpot.

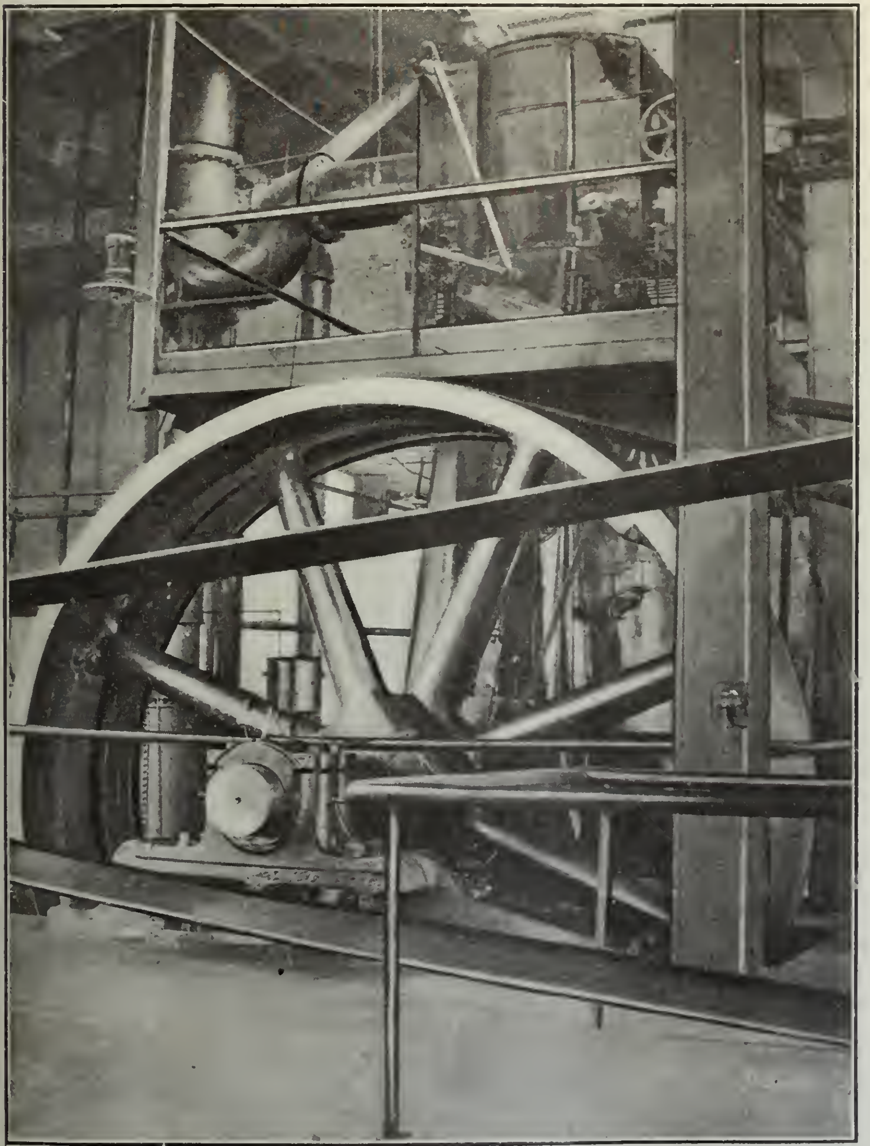


Fig. 1. Odd Design of Vertical Engine

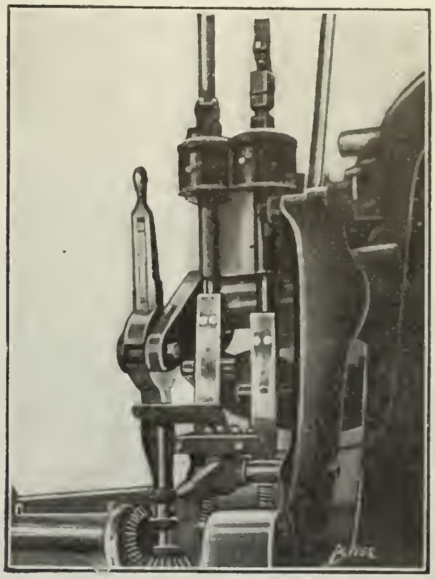


Fig. 2. Valve Gear

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Practical Points in Operating a Producer

A writer in the *Mechanical World* gives the following rather elementary but very sensible suggestions for the starting and care of a gas producer.

Before starting the fire, see that the generator is shut off from the scrubber and that the purge pipe is open. Lay the fire, using coke to start with, on a layer of ashes reaching above the air grate [?] and having lit it, close and seal the fire door when the fire is well established. Start the blower gently, and gradually build up the fire until the fuel bed is about 3 feet thick. When the fire has burnt up fairly red, turn a little steam into the air supply and apply a light from time to time to the testing cock on the generator outlet pipe. As soon as it will light, turn the gas through the scrubber, but not before, as it is no use putting dirt from bad gas into it. Burn the gas off from the dry purifier, making sure first that it is shut off from the holder, if one is used, or the engine main, if not. Gradually turn steam into the fire until the air at intake is about 60 degrees Centigrade, and take a gas sample from the sawdust scrubber. If it contains about 23 per cent. of CO and 16 per cent. of H, turn the gas into the holder or engine main and start the engine, but burn off the excess of gas at the scrubber of a pressure plant until the engine is well under load and the blower is regulated for a continuous supply.

You can then settle down for a week or so to determine the conditions throughout the plant which best suit the coal you are using. The composition of the gas should be approximately 4 per cent. of CO_2, 25 per cent. of CO, 16 per cent. of H, and as much CH_4 as you can get. To obtain this result you must ascertain the best thickness of fire, the best temperature for the blast (of a pressure producer) and the quantity of coal required per hour. The secret of success lies in finding out these conditions and keeping rigidly to them.

A measured quantity of coal should be fed in at regular intervals of not more than a quarter of an hour. The right amount of ashes should be withdrawn from the bottom also at regular intervals, and the fire well poked down solid after each withdrawal, care being taken that no holes are left through the fuel bed. Gas analyses must be taken frequently until the right steam supply is obtained for the required composition. If the percentage of CO is too low, reduce the amount of steam going into the fire. The best temperature for the blast of a pressure producer may be anywhere between 45 and 65 degrees Centigrade, according to the particular coal used.

Having once obtained the composition required, make a note of the temperatures throughout the plant, the pressures as shown by the water gages, the hight of the fire and the quantity of coal required per charge. If these conditions are maintained, the quality of the gas will be uniform hour by hour. The temperature of the gas as shown by the pyrometer or the first thermometer after the generator will be found to be an unfailing indication of the state of the fire. If it varies, it will be found that the air or steam or coal has varied, or that clinker has formed in the fire. Observe that every time a fresh coal is tried the necessary conditions must be determined afresh, but once found, the chemical analysis can be discontinued, though an occasional test for CO_2 is useful, especially if the temperature of the gas has risen appreciably.

Bollinckx Producer and Engine

The accompanying engravings illustrate very clearly the principal features of construction of the suction anthracite gas producer and single-acting engine built by A. Bollinckx, Brussels, Belgium. The generator is of the usual up-draft class, but differs in several important details from American practice. The gases are taken from the fuel bed by a central collecting cone located above the active

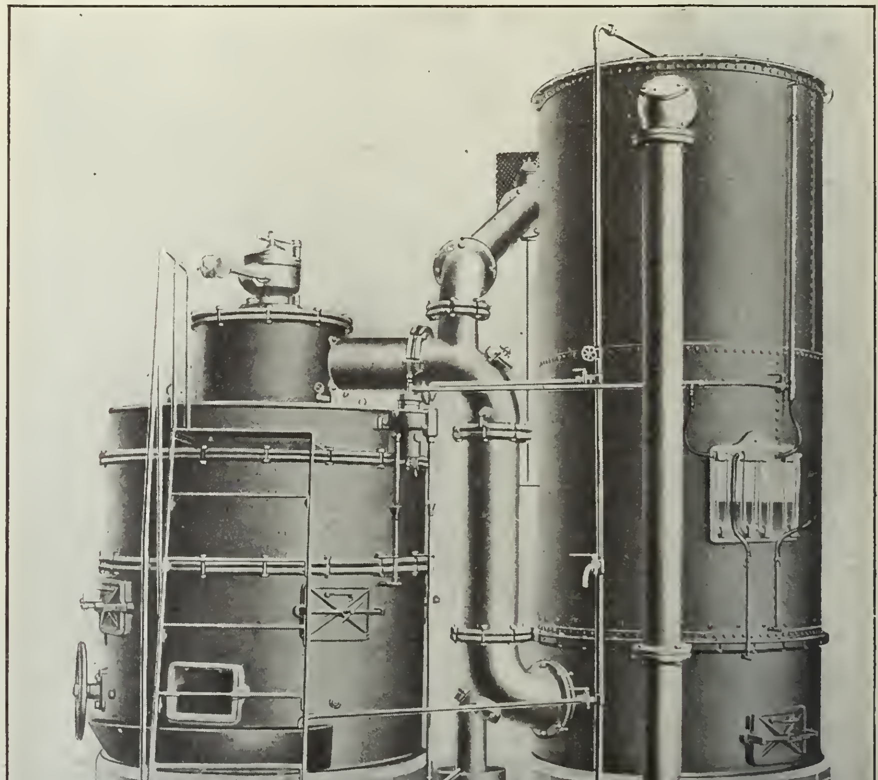


Fig. 1. Bollinckx Suction Gas Producer

zones of the bed, but surrounded by the green fuel, which therefore absorbs some of the sensible heat from the outgoing gases. This has two useful effects, that of reducing the temperature of the gases, and, consequently, the quantity of cooling water required in the scrubber; and that of reducing the amount of heat that must be supplied to the fuel later in order to distil the volatiles out of it, which means that the incandescent zone does not need to supply quite so much heat as it otherwise would.

The vaporizer proper consists of a shallow spiral trough *V*, Fig. 2, around the outside of the cast-iron wall surrounding the incandescent zone, but before the water is delivered to the trough it is warmed in a deep pan *H* built in the top of the generator. From this pan it overflows into the outside pipe *T* whence it drops into a short leader through which it passes to the upper trough of the spiral. This arrangement of feed pipes is employed in order to maintain a constant volume of water in the preheater and to enable the attendant to observe the flow of water, or stoppage of it, from the heater to the vaporizer.

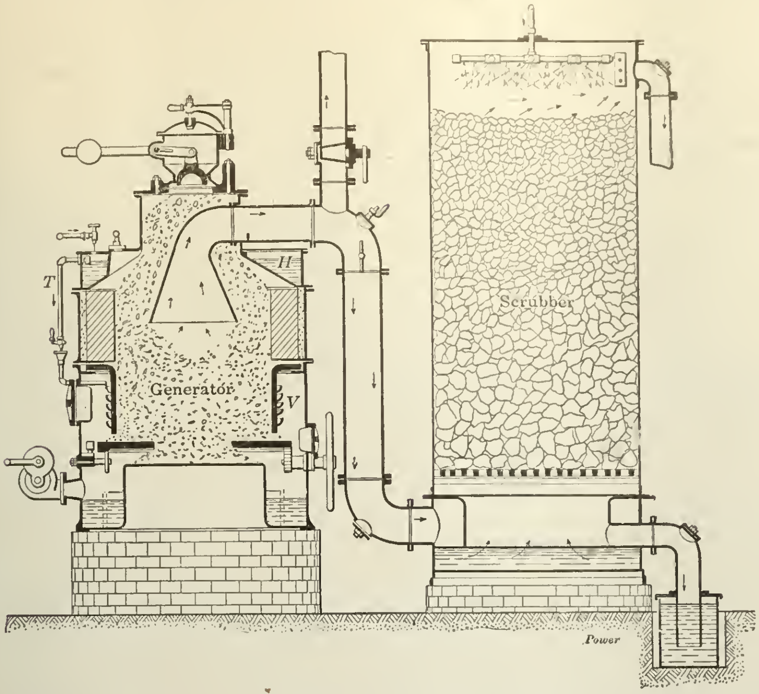


Fig. 2. Sectional Elevation of Bollinckx Generator and Scrubber

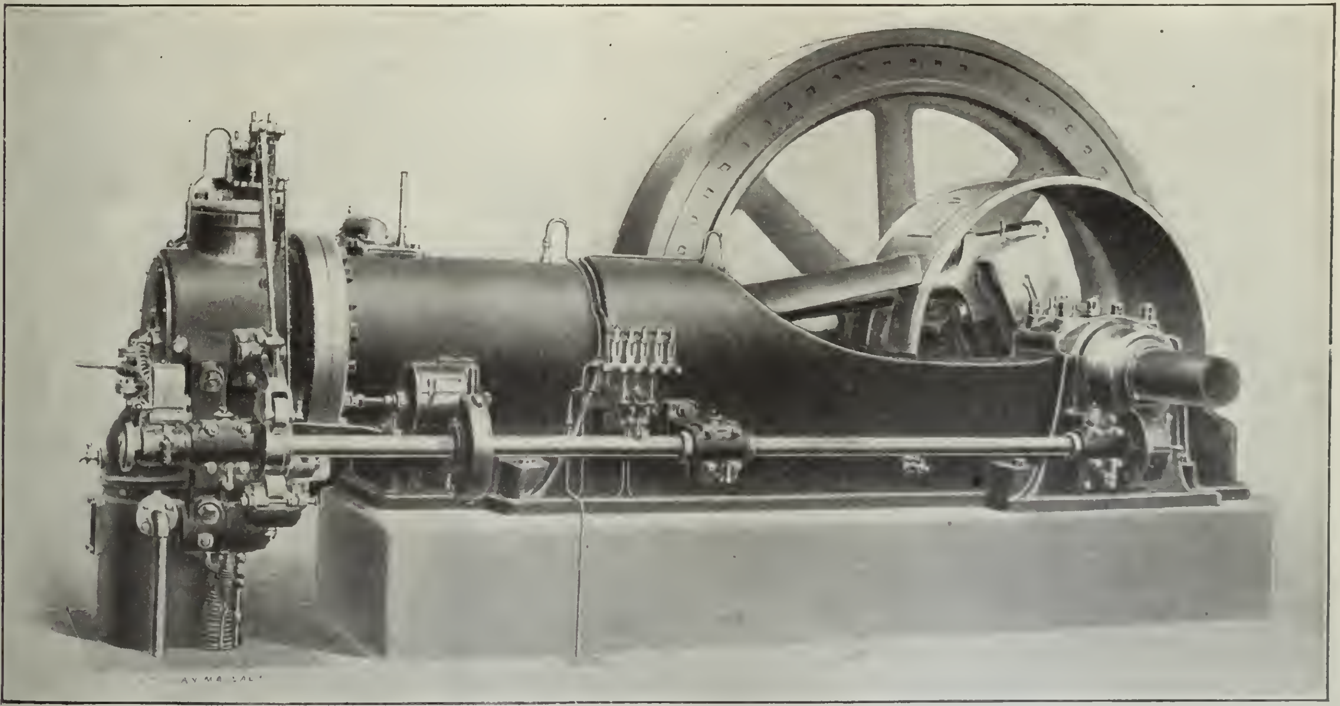


Fig. 3. Bollinckx Single-acting Four-stroke Engine, with Quantity Regulation

The grate is perfectly flat and rests on rollers mounted on studs set into the ashpit wall. A rack is formed around the edge of the under side of the grate and this meshes with a pinion on the shaft of the wheel by means of which the grate is revolved to shake down the fuel bed. The ashpit is "floored" with a pool of water which extends through an opening in one side of the shell into a trough on the outside; the outline of the opening is shown in the sectional view, and the trough outside of it is shown in Fig. 1. Presumably, this arrangement is to permit raking out ashes while the generator is in service. The ashpit door is left open during operation, the air for the fire entering there.

The generator is provided with the usual hand blower for starting the fire and purge pipe for getting rid of the "green" gases and smoke resulting therefrom. Instead of the more common three-way valve at the junction of the purge pipe and the gas-delivery pipe, Mr. Bollinckx uses a stop valve in the purge pipe and a damper form of valve in the delivery pipe; the method of using these is too obvious to require description.

The scrubber is the familiar coke-filled tower, with a water spray at the top and a sump alongside the foundation. As Fig. 1 shows, a gage board is mounted on the side of the scrubber carrying gages which indicate the pressures in the bottom of the scrubber, just below and above the "grate," at the top of the scrubber, and in the gas pipe leading from the scrubber to the engine. No dry purifier is used.

The engine, as Fig. 3 indicates, is of massive design. The external cylinder wall is cast in one piece with the main frame; the inner wall is a liner, with flanges fitting in the outer wall at both ends and the middle, and the combustion chamber is formed in the cylinder head, which is bolted to the outer barrel. The piston is unusually long with respect to its diameter, with the gudgeon pin set almost in the center of its length, and the length of the connecting rod is between five and six crank lengths.

The inlet valve is set in a removable cage and the exhaust-valve stem works in a removable sleeve but its seat is integral with the combustion-chamber wall. As Fig. 4 shows, the combustion chamber flares gradually out to the diameter of the cylinder bore, so that pockets, corners and abrupt angles are obviated.

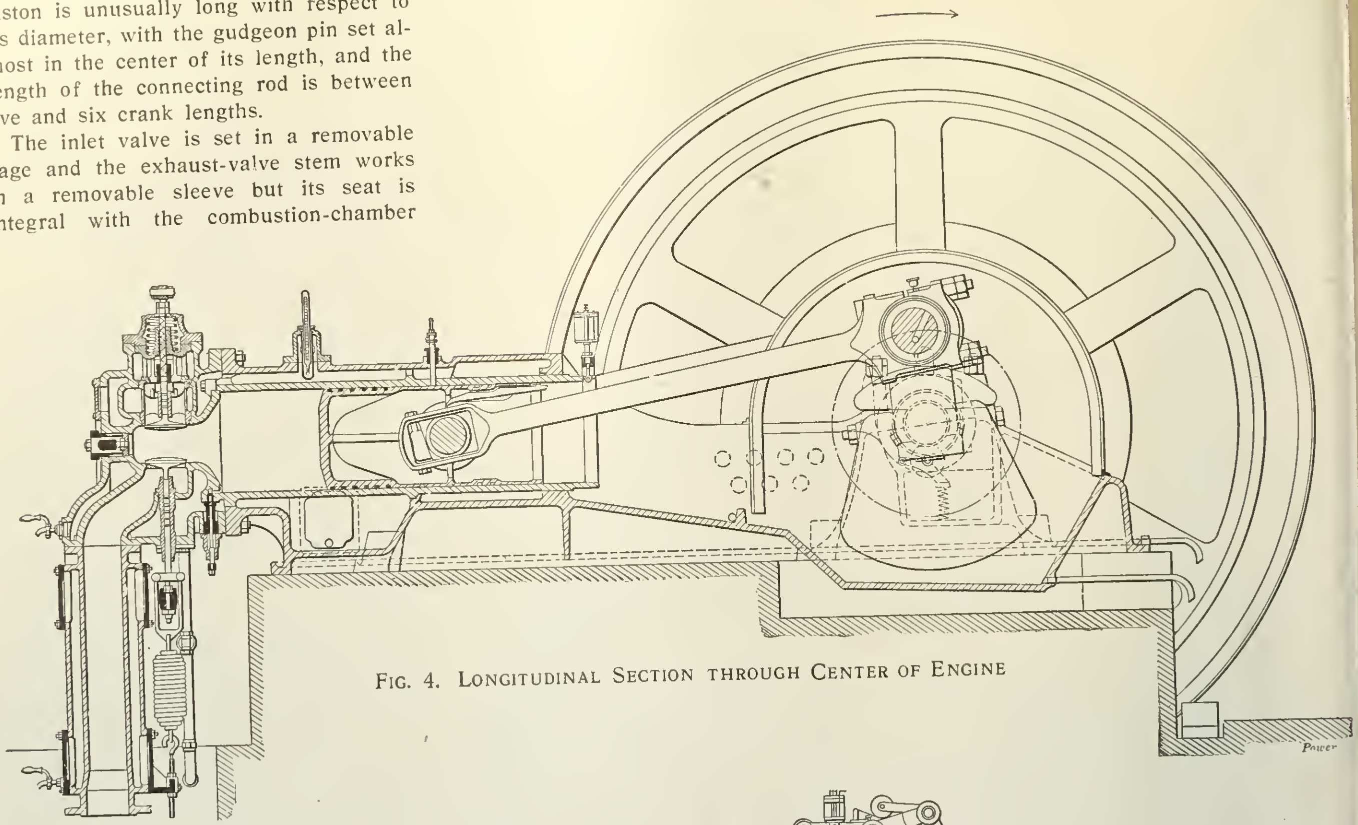


Fig. 4. Longitudinal Section through Center of Engine

Speed regulation is effected by varying the quantity of mixture admitted, and the governor accomplishes this by inserting a thin wedge more or less between a cam and a roller on coöperating members of the valve gear, thereby varying the opening of the inlet valve; the farther the wedge is inserted, the wider is the valve opened. From Fig. 3 it may be seen that both the inlet and exhaust rockers are worked by a single cam. The other features of the valve gear are so clearly shown in Figs. 3, 4 and 5 as to require no explanation.

The ignition system comprises a high-tension magneto which delivers an impulse of current to a jump-spark plug set in the center of the cylinder head. The magneto armature is oscillated momentarily by a trigger actuated from the cam shaft at the proper instant; between ignitions the armature is stationary.

The exhaust pipe is water jacketed from the combustion chamber down to a point beneath the floor level in order to protect the attendant from accidental contact with the pipe. The jacket is provided with two handholes (see Fig. 5) for inspection and cleansing purposes.

Lubrication of the piston, valve stems and gudgeon pin is accomplished by means of a force-feed oiler, the pump of which is actuated by a cam on the secondary shaft (which the retoucher has carefully painted out in Fig. 3). The oil cup shown at the front end of the cylinder barrel in Fig. 4 is no longer used; this sectional drawing was made before the oil cup was displaced by a feed from the pump.

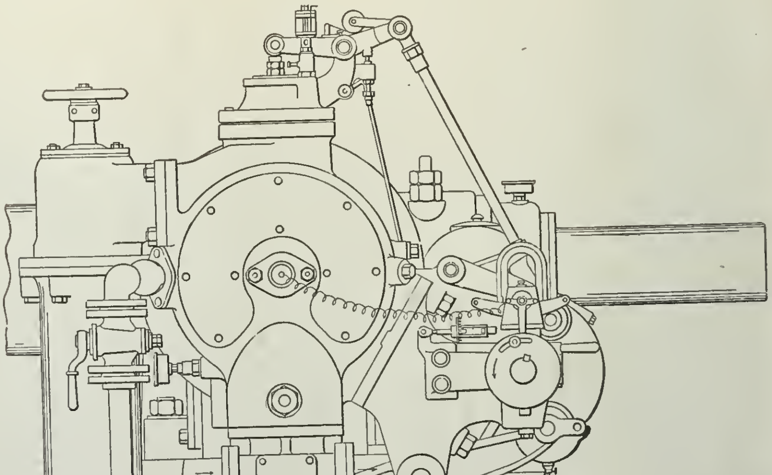


Fig. 5. End View of Valve Gear

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Parallel Operation of Direct Current Generators

By C. L. Greer

In many power stations, especially those of the smaller sizes, whenever a new generator is needed it is frequently installed by the operating force. If the machines are direct-current and are to operate in parallel, certain conditions must be fulfilled to insure satisfactory operation and for cases like that mentioned the following may be of interest to those in charge of the work.

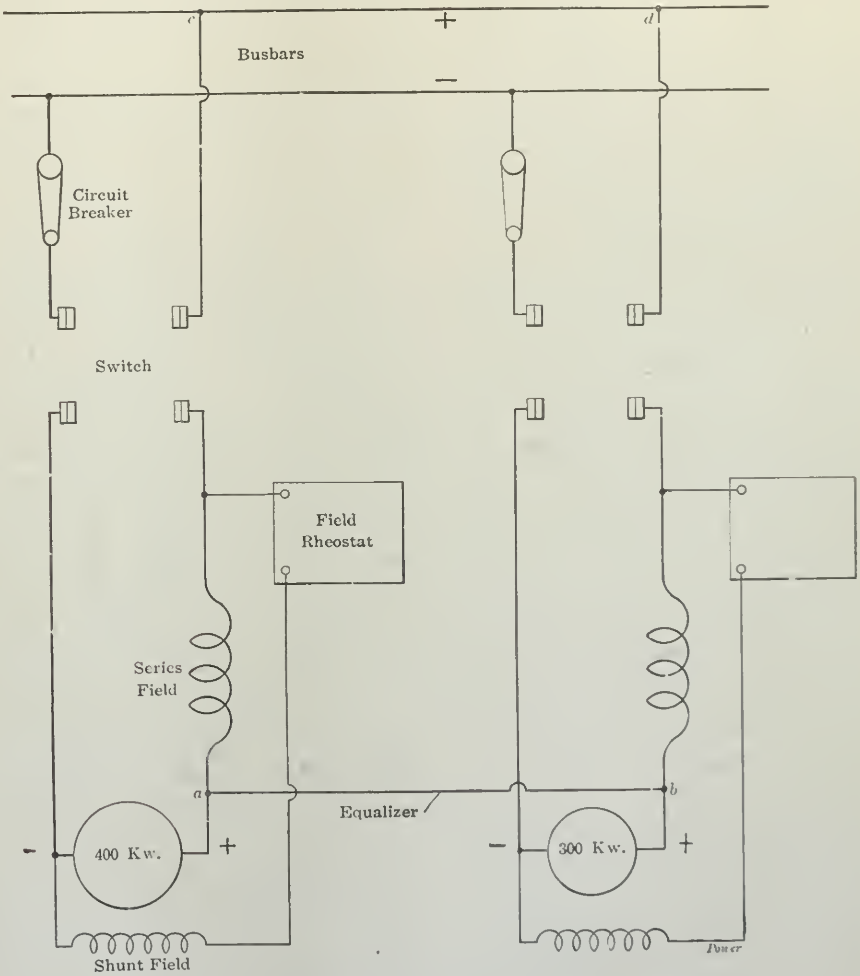
When direct-current machines of the same capacity and characteristics operate in parallel there is usually no difficulty experienced in getting a proper division of the load between them, but when machines of different capacities are to operate together the question of proper load division assumes a different aspect.

With a steady load and constant attendance the matter is not of so much importance as the proper division of the load may be accomplished by the adjustment of the field rheostat, but with a varying load such as that of a street-railway system, to give satisfactory operation each machine must automatically take its share of the load.

The first requisite for satisfactory parallel operation of direct-current machines is that they have the same characteristics. They must both have the same degree of compounding for any percentage of their rated load. If they do not compound alike, it is better not to attempt to operate them together unless the compounding can be made the same in both, which may sometimes be done by putting a dead resistance in the series field circuit of one machine; the material used for this resistance, or resistor, must have carrying capacity enough to pass the full-load current of the machine. If it is contemplated to operate machines of different types in parallel, however, it is well to consult with the makers regarding the changing of their characteristics.

Aside from the first requirement mentioned, there is another which must be met if satisfactory operation is to be obtained. This can be more easily explained with the aid of the accompanying diagram, which shows the connections of two generators intended for parallel operation. It is assumed that the two machines have the same degree of compounding and are alike in every way except in capacity. When the machines are carrying current there will be a certain drop in electromotive force between the positive brush of each machine and the positive busbar, this drop being due to the resistance of the field winding and the cable. Now to get a proper division of the load between the machines it is necessary that the drop from *a* to *c* in the one machine and its lead must equal the drop from *b* to *d* in the other when both are carrying their rated loads. To fulfil this requirement it is necessary for the series winding and lead of the smaller machine to be of higher resistance than that of the larger. If the distance between the two points is the same for both machines, then it is evident that the lead or the series field conductor (or both, perhaps) of the smaller machine must be smaller than that of the other. The operator cannot control or alter the size of the field conductor, so that need not be considered.

When conditions are normal there will be no difference in potential between the points *a* and *b* and therefore no tendency for current to flow between them. The conditions are practically the same as if one armature were delivering current to the two paths *a c* and *b d*; the current will divide between the two paths in proportion to their conductivity, which, if properly proportioned, will be such that only the current of one machine will flow from *a* to *c* and *b* to *d* respectively. But suppose that the conductivity of the leads is so proportioned that that of the smaller machine is greater than it should be with respect to the other. Then the current will not flow through the two leads in proportion to the capacity of the individual



Generators of Different Outputs Connected in Parallel machines but a portion of the current of the larger machine will flow through the equalizer connection to *b* and out through the series field winding to *d*, strengthening the field of the smaller machine and thereby causing it to take an increased portion of the load. The way to correct this is to increase the resistance of the lead of the smaller machine enough to reduce the current to normal.

When machines do not properly divide the load the effort is sometimes made to make them do so by changing the field shunts or adding shunts if none are in use. This will not have the desired effect because when the two machines are in parallel the shunts are also in parallel and while changing the shunts will change the compounding of the machines as a whole it will have no influence on the division of the load.

To indicate how the principle under discussion may be applied in any case, it is perhaps best to assume a case and figure it through. Suppose that the 400-kilowatt machine represented in the diagram be already in operation and that it is desired to install the one of 300 kilowatts and make sure that each will take its share of the load when operating in parallel. It is assumed that the machines are alike except in capacity and that the resistances of the series field windings are such that when carrying full load the drops across the windings themselves will be the same. Suppose that the large machine requires 30 feet of cable to connect it to the positive bus-bar and that 50 feet of cable are required for the one of smaller capacity. Assuming a pressure of 650 volts the current of the large machine will be

$$I = \frac{W}{E} = \frac{400{,}000}{650} = 615$$

amperes and that of the smaller will be

$$\frac{300{,}000}{650} = 462$$

amperes. For illustration, suppose a cable of 600,000 circular mils to be used for the large machine (in practice it would likely be larger); then the problem is to determine the size of the cable for the smaller machine to give satisfactory load division. The drop from the 400-kilowatt machine to the switchboard, in the positive lead, will be 0.35 volt, and it is necessary, therefore, to make the positive lead of the 300-kilowatt machine of such resistance that the full-load drop in that will be 0.35 volt. This may be done by making the fraction

$$\frac{\text{amperes} \times \text{distance}}{\text{circular mils}}$$

equal in both cases. For the larger machine the equivalent of this fraction is

$$\frac{615 \times 30}{600{,}000} = 0.03075$$

For the smaller machine, then,

$$\frac{\text{amperes} \times \text{distance}}{\text{circular mils}}$$

must equal 0.03075; consequently,

$$\frac{\text{amperes} \times \text{distance}}{0.03075}$$

must equal the circular mils of the cable to be used, which figures out thus:

$$\frac{462 \times 50}{0.03075} = 751{,}220$$

circular mils; the nearest commercial size is 750,000 circular mils. This is larger than the cable of the 400-kilowatt machine, and the reason for it is that the length of the cable on the smaller machine is 60 per cent. greater than that of the larger machine. If it is undesirable to put in so large a cable for the smaller machine, a resistance bank of some kind could be inserted in the lead of the large machine to properly proportion the resistances of the two circuits. The effect would be the same but the inserted resistance should have ample current-carrying capacity to prevent excessive heating.

No matter what size cables are used the resistances of the two connections must be so proportioned that the drop will be the same for both machines between the equalizer junction and the main bus-bar when each is delivering its full-load current.

This rule holds good for machines of the same capacity, and where there is an appreciable difference in distances between the machines and switchboard the cables should be cut the same length and that of the nearest machine coiled or made up in some way to keep it out of the way.

In the example given it was assumed that the series field windings were on the positive side of the machines; if they were on the negative side the rule would then apply to the negative cable.

LETTERS

The Balancing Coil of a Three Wire Generator

Mr. Kerr's letter* on the subject of the three-wire generator contains one important error. He says, "as the coil is an autotransformer with a one-to-one ratio, it would have to be but one-half the size of a transformer of the same volt-ampere capacity" (true), "or 9 per cent. of the capacity of the generator" (wrong). He has already figured the capacity of the coil correctly at 18 per cent. of the generator capacity. The fact that the coil is half the size of a transformer of equal capacity does not reduce the size of the coil; it means that a transformer to do the same work would be twice as large, or equal in size to an autotransformer having 36 per cent. of the generator capacity.

This error does not occur in the next paragraph, which is inconsistent with the statement quoted. With an allowance of 25 per cent. unbalancing, as it is commonly termed, the neutral current being one-fourth that in the heavier loaded main conductor, each balancing coil, when two are used, must have a volt-ampere capacity of practically 4½ per cent. of that of the generator, and it would be the same size as a transformer of 2¼ per cent. of the generator capacity.

Geo. W. Malcolm.

Brooklyn, N. Y.

Polarity Indicators

R. T. White, in the August 16 issue of Power, mentions a simple method of determining the polarity of two direct-current conductors. His method, though very good, necessitates the use of three articles: first, conducting wires; second, water, and third, a container for the water.

Were Mr. White to take a potato, one of the common or garden variety, and insert his two wires into a freshly cut surface, he would notice the region around one of the wires become green, due to the copper dissolved off the wire; this wire, surrounded by the green stain, is the positive wire. If both wires become surrounded by dark colored stains, the current is alternating.

Another method of differentiating between alternating and direct current, recently advanced in an Eastern paper, is to hold a bar magnet near a lighted incandescent lamp; if the "juice" is alternating, the filament will vibrate, if direct, the filament will be attracted or repelled, according to the sign of the magnet pole presented.

Mr. White is mistaken when he states that air bubbles form around the negative wire; the bubbles are of hydrogen gas.

Wesley McArdell.

Brooklyn, N. Y.

When a boiler is to be cleaned, allow the fire to die out, and leave the dampers open with the fire doors closed for at least twelve hours, when the steam gage should indicate no pressure. Then admit air on top, open the blowoff cock, and allow the water to run out. Remove the upper manhole plate, and use vigorously a large hose with ample water pressure to dislodge loose scale, and wash out the boiler thoroughly. Now remove the lower manhole plate and all the handhole plates, and entering the boiler, remove the lodged scale at the turn of the flanges and below and between the tubes. See that all parts of the boiler are entirely free of scale and clean internally and externally from head to head, and see that the furnace is cleaned out from end to end. Examine carefully all parts of the boiler for defects, particularly the parts that are subjected to the direct heat of the furnace,

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*Page 1688, September 20 issue.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Condenser Trouble

The condensing outfit of a certain plant, when first installed, maintained an average of about 26 inches of vacuum. The plant was run from Monday morning to Saturday night. After it had been in service a few months it was noticed that when starting on a Monday morning the vacuum was about 25 or 25.5 inches, but toward the latter part of the week the vacuum dropped to about 23 inches. Later on, the vacuum was found to be about 23 inches when starting on Monday morning and as low as 18.6 inches by Saturday.

All of the connections were examined and found to be tight and free from leaks. The condenser was opened and it was found that the tubes were clogged. They were bored out, this operation being repeated each week, with the only result that the vacuum would be up to 23 inches at the beginning of the week. It would drop to 19 inches again by the end of the week. It was evident from this that something other than clogged tubes was causing the trouble.

The condenser manufacturers were appealed to and a representative was sent to examine the outfit. The exhaust from the turbine contained a quantity of oil. It was found that this had accumulated on the outside of the tubes. The circulating water, taken from a nearby pond, entered at 60 degrees Fahrenheit and was not very warm when discharged. The following Sunday, when the plant was shut down, the valves of the air pump and the piston packing were removed and the condenser was flooded with a solution of boiling water, saturated with soda; this removed all of the oily sediment. After being thoroughly dried, the condenser was shut off from the turbine, the circulating-water inlet and outlet were disconnected, a blank flanged placed on both openings and a 50 per cent. solution of hydrochloric acid was pumped through the condenser. This solution thoroughly cleaned the surfaces of the tubes. After being washed free from the acid, the machine was carefully drained and again connected to the turbine and the circulating-water piping.

When the outfit was started the following morning the condenser produced 26 inches of vacuum. After two months' running the vacuum began to drop gradually. When it reached 23 inches the same operations had to be repeated.

The cause of the trouble was easily traced to the circulating water, which caused scale to form in the tubes, and the oil in the exhaust steam, which caused the sediment to form on the outside of the tubes.

Nathan Owitz.

New York City, N. Y.

Continuous-feed Oil Reservoir

In a plant of which I had charge, there were 12 sight-feed lubricators, which were used for supplying cylinder oil to various pumps and engines. As it required considerable time to fill these lubricators, we decided to install an oil reservoir and to supply cylinder oil to each lubricator continuously through suitable pipe connections between the main oil reservoir and the several lubricators. Fig. 1 shows how the main reservoir was connected. As three of the lubricators were about 200 feet away from the reservoir, great difficulty was experienced in getting these lubricators to feed, especially in cold weather. Changing the steam connection to the oil reservoir as shown in Fig. 2, overcame the trouble entirely, and these far-away lubricators feed nicely. The steam connection was run to the highest point in the engine room which was about 34 feet above the reservoir. The drop pipe filled with condensed steam and thus furnished a hydrostatic head equivalent to about 15 pounds. As the steam pressure was 100 pounds, there was 115 pounds total pressure on all of the oil lines and lubricators. The 15 pounds excess pressure was sufficient to overcome the friction in the lines and to insure a steady positive feed.

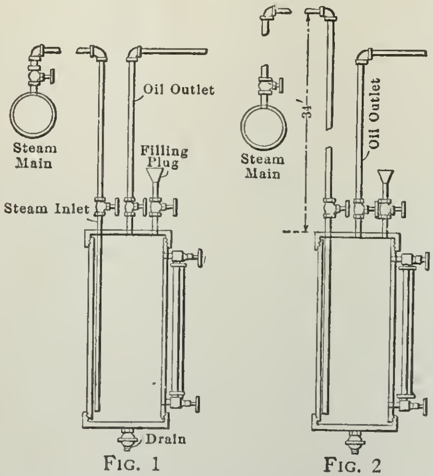


Fig. 1 Fig. 2

Original and Improved Arrangement of Steam Piping

Paul McIntire.

El Reno, Okla.

Why the Pump Made Only Two Strokes

The pump had been repaired in the early spring, then brought to the mine and tried out at the surface by connecting with the steam hose, and had run all right. It was then moved a few feet to one side and allowed to stand the rest of the season. The pump did not set level, however, but stood at quite an angle, and the cast-iron air chamber had been removed; a tarpaulin had been thrown over the rocker arms and piston rods, but the rest of the pump was exposed to the elements.

In the early part of November the pump was lowered into the mine, the old pump being disconnected, and the pipes connected to the new one. When steam was admitted the pump made only one stroke on each side and then refused to budge. The operating force worked with this pump from about 6 a.m. until 4 p.m. The water was then almost knee deep, or halfway up on the cylinders. At this point I was sent for in the hope that I might be able to start the pump; most of the tools, however, were under water. Something had to be done to prevent the mine from being flooded. Of course, the water did not raise as much at this time as when it first came up, as the area for the water to spread over was larger.

One rocker arm was disconnected and turned upside down, a new keyway was made, the key fitted, and a man operated the rocker arm, thereby working one side of the pump, the valve being disconnected and placed central over the ports. In about twelve hours the water was well down in the sump. The cylinder head was then taken off and the steam end and piston were found to be intact; the valve was found free and the ports were unobstructed; the packing from the rod was removed and everything found all right. A bar and sledge were applied to the spool on the rod, and the rod could not be budged. The cylinder cover of the water end was removed, and a block of hardwood applied there with a sledge, but still the plunger would not move. A jackscrew was then brought into play on the water piston and the pump was braced across the gangway; by using the bar and the sledge on the spool and bearing on the jackscrew at the same time, the piston was finally moved. The side plates were then taken off and the solid plunger was sandpapered and oiled, and it was then found to be perfectly free.

The trouble was caused by water getting into the valve chamber, over which the air chamber should have been, and carrying with it more or less dirty sand, fine ashes and other material that is around a coal mine; this had dropped onto the solid plunger and was forced back into the casing, wedging the plunger fast. When these things were removed the pump ran as good as it did when it first came back from the shop.

H. R. ROGERS.

Seattle, Wash.

Zinc in Boilers

Our plant is equipped with two 5x16-foot return-tubular boilers, one of which is about ten years old, and the other about four. The operation of the plant is such that it is necessary to use only one boiler at a time; this is usually the newer one, the older one being used while the other is being washed; or, sometimes in very cold weather, it is necessary to use both boilers.

Soon after taking charge of the plant, I found upon examination of the new boiler that it was badly scaled for one in use so short a time—at that time only about a year. Upon making inquiry as to what was used to prevent scale, I was informed that a peck of potatoes had been used every time the boiler was washed out. As the boiler was washed every two weeks, it was evident that the potatoes were not doing their work. Having read that zinc is used in marine boilers with good results, and having used commercial mixtures, both dry and fluid, in other boilers without good results, I decided to give the zinc a trial. I bought seven pounds of zinc, drilled two holes through it, ran a wire through one of the holes and suspended the zinc from one of the lower tubes. I soon discovered that this was not the proper way to use the zinc as it soon crumbled and fell to the shell of the boiler. I accordingly made a basket of heavy wire screen, suspended it from one of the lower tubes and placed the zinc in it.

It was six weeks before I noted any results, then I removed nearly one-half a bushel of scale, and each two weeks thereafter for two or three months, I removed more or less loose scale. At present the boiler is free from scale. The purer and closer grained the zinc is, the better. I then tried the zinc on the old boiler, which was badly scaled, and again had good results. Shortly after this the boilers were inspected and pronounced to be in first-class condition.

Our heater is of the open type, and formerly it was necessary to remove the pans and clean them every four weeks. Since placing zinc inside them it has been necessary to clean them only once—a period of nearly eighteen months.

D. A. DICKINSON.

Lima, O.

Firing with Gas

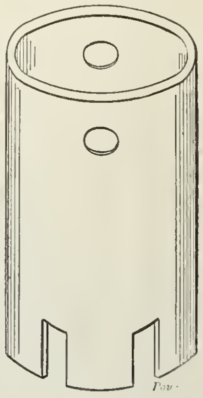
I would like to hear from some of the readers of POWER as to their experiences in firing with gas—how they arranged the checker work, draft and what kinds of burners gave the best results. I have tried several types of burners, but they all seem to allow most of the gas to pass up the stack only partially burned, as is indicated by clouds of black smoke.

F. P. WILSON.

Grafton, W. Va.

Homemade Tools

Having occasion to take valve seats out of the water end of our air pump, I made a wrench out of a piece of pipe by cutting notches in the end of the pipe to fit over the spokes of the valve seat. The seats are 5 inches in diameter and a piece of 4-inch pipe fitted nicely inside the outer ring of the seats. The figure shows how the wrench looks. By using a rod in the hole at the upper end of the pipe or a chain tongs, the seats can be taken out easily with a wrench of this description. In making the wrench, care must be taken to cut the slots so that each will bear on a spoke when in use. I made the slots with a hacksaw and chipped out the pieces with a chisel. It took 15 minutes to do the job.



WRENCH FOR REMOVING VALVE SEATS

Another handy little homemade tool can be made as follows: If a wrench for hexagonal nuts is wanted, select from the scrap pile a piece of pipe which will just fit over a nut of the required size. Heat the end of the pipe, place it over a nut and pound the pipe down about it. A rod through a hole in the other end of the pipe or a pipe wrench completes a good homemade box wrench. One of these has saved the day for me many a time. They can be used in a hundred and one different ways about a plant.

I. VIGGERS.

Everett, Wash.

Bushing Valve Stems

Four valve stems and valve-port covers on a 75-horsepower engine were badly worn, and we undertook to bush them. The stems were 1¼ inches in diameter. We obtained some 1¼x1-inch cold-rolled seamless tubing, and turned down the long ends of the valve stems, that project through the bonnets, so as to make a tight fit into the 1-inch tubing. We cut the tubing the right length and forced it onto the stems, then recut ¼-inch key seats in the stems for the bell cranks; this gave a ⅛-inch key seat in the bronze part of the stems.

The pivot ends of the valve stems were so short that we decided there would not be bearing enough to force the tubing on them as we did on the long ends. Therefore, we turned them just enough to make them round; this made them 1 1/16 inches in diameter.

The covers were then bushed to fit the stems. In bushing the covers we turned off the rings which hold them in position, and bored a 1½-inch hole in each of them. Next, we turned some cast-iron bushings to fit the covers, each having a 4x⅛-inch flange on one end and a 1/16-inch hole. This flange was to take the place of the rings which were turned off the cover. We forced these bushings into the covers, and put two pins through the flanges into the covers to make sure they would not work loose.

These bushings and sheaves have been in service for two years and have not shown the slightest wear, and we have packed the glands in the bonnets only once. It is certain that these steel bushings will outwear a set of bronze stems. and they cost only a quarter as much.

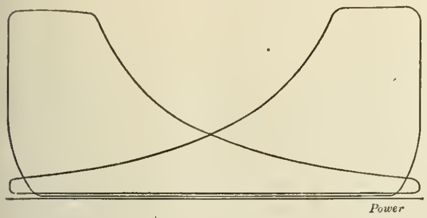
E. W. NEAL.

Pittsfield, W. Va.

Economy in Small Matters

While reading over the various items in several of the mechanical papers recently, my mind became centered on the question of high pressure in steam boilers and engines and the consequent results. High-pressure steam is theoretically an economy, but to secure this economy high-pressure conditions are necessary. When a plant is operating at top-notch load, especially an electric station, it may be economy to run at top-notch pressure. But it is questionable if it is wisdom to run very light loads at this same top-notch pressure, owing to the fact that, although the demand for steam is small, the loss by radiation from the various steam lines is high. Consequently, the percentage of condensation is greater than when the boiler is putting forth its full capacity of steam. Wear and tear is much less with low pressures and the water in the boiler being at a lower temperature, is better able to absorb the heat of the furnace gases than when everything is crowded to the full limit. Therefore, it seems to me that everything points to the economy of low pressures for light loads.

It is only by rigid care and attention to leakage and small details that high-pressure steam can be made to yield the economy, or any part of it, that theoretically it should yield. Engineers are sometimes apt to cry over the old-fashion machinery that they have to operate, as though they felt it to be impossible to secure good results from anything that is not right up to date. But they are entirely wrong in this matter. I have seen better results obtained with old-fashion engines and boilers and under old-fashion conditions than have been obtained in a good many cases with new uptodate equipment. These good results were obtained simply by the use of care and attention. There was no want of safety-valve tightness, and the firing, while done by hand, was done intelligently so as to keep the steam at the necessary pressure and not cause it to vary—up to the blowing-off point one minute and down to half the normal pressure the next. A varying steam pressure is far more wasteful than is generally realized. Steam pipes were well insulated and the valve bodies as well. If a condenser was part of the equipment, the vacuum was carefully maintained.



Diagrams Taken from Old Engine

Injecting pipes are, to a certain extent, porous and, consequently, if they are very long, they let in a lot of air. Painting the pipes with a good boiler paint will prevent this to a large extent. All possible air inlets, such as at flanges or valve stems, should be carefully watched. An engineer should not be satisfied so long as the vacuum is not up to the requisite amount. It is all very well to look for things new and modern, but before advocating them be perfectly satisfied that they are better under the same kind of conditions than the old. An old engine is not always necessarily wasteful. Of course, age presupposes worn-out valves, scored cylinders and rods, but these are not always the conditions that exist. The accompanying set of diagrams was obtained from an engine that was built in 1861 and has been continuously in use since that date. It is 16x36 inches in size and runs at 100 revolutions per minute.

No manufacturer having common sense, will use an old plant if he can afford a new and more economical one, and the best way for an engineer to enable him to do so is to keep a sharp lookout for all the small losses and details and get the best possible results out of the old machinery. This will not only prove the ability of the engineer to care for the old plant, but will also prove his ability to care for the new one when it can be afforded. In steam-engine economy it is often easy to cut down the amount of fuel used 25 per cent. or more by simply looking after the small items which appear to many engineers to be of small consequence but which amply repay the attention bestowed upon them.

Charles H. Taylor.

Bridgeport, Conn.

Binding Copies of Power

Having received the index to Power a short time ago it occurred to me that it might be useful to some other readers to describe my method of binding the copies of Power for future reference. The scheme is easily accomplished and is inexpensive.

First, assort the copies so that they will be in successive order, the pages running from 1 to 1190, and comprising one volume for the first six months of the year. Then remove the covers; also the wire staples which hold the leaves together should be removed before the reading matter is separated from the advertising, otherwise the pages are liable to tear. Now, remove whatever glue that remains on the reading section and place the index at the front. Put some old sheets of paper on the outside of the outer pages so that the reading matter will not be soiled while the book is being finished; these sheets can be torn out later. Straighten the backs of the copies and lay them down on a bench or table, put a board on top of them and a weight on the board so that the paper will be pressed firmly together; if this is not done, large spaces are apt to remain between the individual copies. Now take a brush and some shellac or glue and put two or three coats on the back edge. When it is sufficiently dry, separate the book from the table and board by means of a knife, otherwise the book is liable to separate at the center on account of the shellac sticking to the board and table.

When this has been done procure a common nail four or five inches long and file it to a taper so that it can be pulled out easily after it has been driven into the paper. Drive five holes in the back margin of the book; then with a needle and a strong thread run it through all the holes, and tightly bind the leaves together. The book is now ready for the cover. For this I used a piece of insulating fiber paper, such as is used by electricians; this is about the same thickness as cardboard. After cutting a piece large enough for the cover measure the thickness of the book and, with the aid of a square, make two creases at the center of the cover. The fiber paper is rather stiff and hard to shape, consequently if the paper curls it may be made straight by moistening it with a piece of damp waste. Now, put a weight on it and leave it in this position for two or three hours; after which, again paint the back of the book with shellac, also between the creases on the cover, and put the latter on carefully, rubbing your fingers over the back of the cover where it is pasted so that the entire surface will stick to the cover. Repeat this for five or ten minutes or until the shellac is well set. The edges of the pages will be rough, but these can be trimmed by taking them to a printing office and putting them in the paper cutter. If fiber paper is not available, a sheet of tagboard can be used for the cover. This can be had at any printing office for 5 or 10 cents.

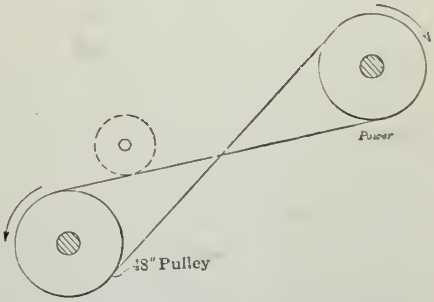
The binding of Power did not cost me a cent outside of the time spent, as the material was picked up around the plant. I consider these bound volumes of great value as a reference library and would advise other readers to begin binding their copies at once as they are ten times more valuable when bound than when scattered all over the engine room. When indexed they are worth as much as a book costing $5, and yet they cost only $1 per volume.

Louis J. Gorilla.

Ironwood, Mich.

Trouble with the Belt

The accompanying sketch shows the arrangement of belt drive which has given us considerable trouble. The pulleys run at 400 revolutions per minute and are connected by an 8-inch, crossed belt, which continually runs off. Would an idler, placed as shown by the dotted lines, prevent the belt from running off, or would it cause too much friction at the cross? Also, would it be necessary to use a cone pulley for the idler, as the belt slants somewhat?



Belt Drive Showing Proposed Location of Idler

I should appreciate opinions of Power readers upon the foregoing.

Ray Hamill.

Shelby, Mich.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Verdict on Canton Explosion

The verdict of the Canton coroner on the cause of the boiler explosion in that city is paralleled only by the one given by Mark Twain. Buck Fanshaw, suffering with delirium tremens, sent a bullet through his brain, cut his throat and leaped from a window three stories above the sidewalk. The coroner's jury, with their intelligence unblinded by the sad fate of their lamented friend, rendered a verdict of death by the visitation of God.

Seriously, however, is it not time for the public to have boiler explosions investigated by experts capable of fixing the cause instead of persons holding political office and absolutely without either the technical or practical training necessary to solve the question? In this case, essential facts, revealed by the testimony, were boldly ignored in order to assign a cause not warranted by the facts. The coroner knew nothing of boiler construction or operation. The mute testimony of the parts remaining could not pass through the veil of ignorance that surrounded the official who expects engineers to applaud his verdict. The fatuity of the man is appalling.

Perhaps, however, we may hope for a statement from the builders of the ill-fated piece of apparatus giving us their reasons for the construction that has resulted in such a disastrous explosion. Curiously, the boiler that exploded at Midvale, O., last winter was of the same construction. If these explosions were not due to structural design causing excessive flexure resulting finally in the explosion, what else was the cause? The water-hammer theory will not account for the explosion, for water hammer would rupture the plate directly on top.

In my opinion the time has arrived when boiler explosions should be investigated properly. Neither the maker, the owner, nor the attendants should be screened. This business of perfunctorily turning over such matters to minor officials having no knowledge of boilers is stale and rank. Nobody is punished and none censured unless he be killed in the affair. The matter is forgotten until recalled by the next explosion. Massachusetts, alone, of all the States has rules worthy of consideration, but it does not appear that the Buckeye State will even consider the adoption of similar rules until more explosions have occurred and more lives have been sacrificed.

The discussion of this and similar explosions in POWER should result in the various associations connected with steam engineering taking some positive stand in regard to demanding the adoption of the Massachusetts rules in the several States and having the governors appoint men of ability to investigate explosions in the future.

A. J. FORAN.

Paterson, N. J.

Flue Gas Analysis and Calculations

An error appeared in the article on "Flue Gas Analysis and Calculation" which was published in the August 9 issue. In the small table at the foot of the first column on page 1416, the molecular weight of nitrogen was given thus:

$$N = 14.$$

Correctly, this should have been,

$$N_2 = 28,$$

for a molecule of any gas is considered to be composed of two atoms. As the atomic weight of nitrogen is 14, the molecular weight is, of course, 28.

This necessitates a revision of the calculations at the foot of the second column. The weight of N, instead of being

$$80 \times 14 = 1120,$$

is

$$80 \times 28 = 2240.$$

Then the total weight, instead of being $28 + 440 + 228 + 1120 = 1816$ weights, is $28 + 440 + 288 + 2240 = 2996$ weights. The part by weight for CO is, therefore,

$$28 \div 2996;$$

that of the CO_2 is

$$440 \div 2996,$$

etc., and *not*

$$28 \div 1876$$

and

$$440 \div 1876,$$

as was given. The last sentence in the third column on page 1416 must be changed to read: In the case cited the oxygen used to make the mixture is

$$624 \div 2996$$

of the whole.

The table on page 1417 headed "Convenient Form for Flue Gas Calculations" must be corrected for the nitrogen (N) weight in the last column to 2240 and the total weight to 2996.

The errors in no wise affect the accuracy of the actual calculations of results, for the erroneous figures were not used in obtaining the results but simply included to lead up to the principle of the method employed.

F. R. GRADY.

New York City.

Engineers' Wages

My experience covers several stages of machine-shop work, construction work, general engineering and finally operating work in which I have now been engaged for the past seven years.

Being wholly a practical man, educated by night study, employed principally by large concerns, I have met with all classes of engineers—technical graduates and practical men—and have made a careful study of them all.

Rarely is the technical graduate found as an operating engineer for he almost invariably turns to office work, construction work or to the sales department of some concern. This, then, leaves the operating work almost wholly to practical men.

Why have not the operators improved their positions?

My answer is that the practical man, as a rule, has been busy training himself not so much to carry the larger responsibilities which come with an advanced position but training himself and his hands to do skilled work with and on the operating units where his work calls him. Admiration is often expressed of the skilful work of one of these men, but how often the one that gives him the highest praise is compelled to say the fatal words that bind him forever to this work with no chance to advance.

These are the words, "A better man cannot be found as long as he is not given more than he himself can do. For if he is given charge of men he fails absolutely in the management of them."

Many a time have I heard this verdict passed; it is not uncommon to hear the same of technical men even after they have had several years of practical experience. Successful supervision of others is a gift, but training can go a long way toward assisting one to whom it does not come naturally. In this the technical man has an advantage hard to overcome for the training he has received has caused him to see the advantage of it. The operating engineer must start at the bottom and slowly work up. This takes several years and usually is more or less of a grind, for he must depend on those over him to advance him and instruct him.

This, to my mind, brings us, in the end, an engineer slow to take the initiative, slow to push himself forward and deficient in confidence in himself to handle things alone except on work with which he is perfectly familiar.

As to salaries, they are not what they should be but a gradual change is going on and with improved working hours, under better conditions, a different class of men is entering the field and gradually replacing those who are not so competent. With the license laws now in and going into effect, we must expect to get a better educated, better trained and better class of men all around in this field; men who by attaining better results will command better salaries.

William Welsh.

St. Joseph, Mo.

I have read Power for some years merely as a matter of self-education. My duties are not such that I may be termed in any sense an "engineer," nor my qualifications such that I could assume that title; yet I enjoy absorbing the information and endeavoring to solve the problems found in the columns of Power fully as well as though stationary engineering were my chosen profession or means of livelihood.

On any of the technical discussions I do not feel competent to speak, as I realize that what knowledge I have acquired is strictly elementary by comparison with that possessed by a vast majority of Power readers. There is one topic, however, dealing with the business side of the profession on which I feel that I have the privilege of speaking for the reason that I am a disinterested party, yet one who has had the opportunity to make some little observation of the relation between an engineer's efficiency and his wage-rate.

I have read that there are "good, practical, operating engineers bearing responsibility equal to or greater than that of foremen, superintendents, etc., yet receiving as their remuneration little more than the wages of unskilled labor," or words to that effect. Now, if this is true in any case, I have no suggestions to offer other than that it possibly is a matter which should be referred to the physicians and judge of that section of the woods who determine who of the population may be the guests of a certain well governed hotel maintained by each of the States, for there are ways too numerous to mention by which a man who really possesses qualifications can readily find a market for his services. If he endures the outrage of being underpaid, he thereby casts reflection on the truth of his claim to more than the "unskilled-labor" degree of intelligence.

I believe that a little story of my own personal experience will illustrate best what I wish to say on this subject and call attention to the causes operating to keep some engineers plodding along in the poor-pay rut other than the injustice of society in general, "Providence," or whatever cannot be blamed on their ancestors or the Republican party.

I once had a friend who was an "engineer"; he had proved it by drawing pay as one for some twenty years, and by never blowing up a boiler, nor completely forgetting the use of his oil can. He was steady, sober and reliable, always put in full time and never caused a shutdown. For these reasons he considered himself underpaid at $3.50 per day. He spent considerable time telling about this to anyone who would listen, and gradually he got into the habit of being dissatisfied with everything and everybody, until he earned the title from the other employees of the establishment of "Sore head"—this was paid promptly.

As time rolled on, the old engine and boilers wore out and were replaced by new ones of higher rating. To the surprise of all concerned the new equipment failed to develop the power which was expected of it. An investigation followed.

It was evident to the nontechnical parties interested that the space under the boiler was sufficient to burn the required amount of fuel; by comparison with similar installations it was evident that there was sufficient heating surface; rapid combustion led to the belief that there was draft enough; the lines of the engine were evidently properly proportioned, and the engine itself bore no indications of inefficiency. Yet, when the boiler was crowded to its capacity as soon as the load went on, the gage hand would drop to one-half of the pressure at which it should stand even when the load was known to be only a fraction of the boiler's rated normal capacity.

The "engineer" was called upon to explain. He said that he guessed they would have to run a boiler trial and analyze the flue gas and indicate the engine; but all of these things were the work of an "expert" and he could not be expected to know anything about them.

The folks about the place noticed that the exhaust was more like the noise of a skyrocket than that of a roman candle, but they were not engineers and that circumstance relates to what the firm paid an "expert"—something like $25 per day to investigate for them. This "expert," who really was one, told us that his learning was entirely self-acquired and obtained at night. It made him worth, some months, about one or two hundred dollars in addition to his regular salary without interfering with his regular duties.

Referring again to our disgruntled friend:

I have very vivid unpleasant recollections of once trying to interest him in a correspondence course similar to one I was pursuing as a pleasure. I do not believe it necessary to add any words of explanation; those to whom they are necessary would not heed them any more than our friend did. But it might be well for anyone disposed to believe some of the "hard-luck" tales that are told to investigate and determine for himself if interested. A thorough diagnosis of a case would disclose whether the party so downtrodden is fit to bear the title "engineer" or whether he is merely a "stopper and starter," an "oiler and wiper," or a "whistle blower." I believe that we all will agree that some power-plant equipment is intrusted to "unskilled labor" and that the "unskilled" ones get paid all that they are worth until they begin to think, when they will at once become of more value to their employers.

Now, as for the comparison with foremen, superintendents, etc., if these men all were interviewed there would, no doubt, be quite a few of them who would report that they were overworked and underpaid. But, like the breakfast food, in most cases an investigation would reveal that "There's a reason." If a foreman were called upon for an account of his department and replied "get an expert" the management, I believe, usually would—it would probably be necessary.

It would then probably get a *foreman.*

W. R. Smith.

Alton, Ill.

Pumping Layout

In his inquiry in the August 9 issue, Mr. Fillmen does not give the hight of the suction lift or the probable distance the water would be lowered when 75 gallons per minute were being taken from each well. If the water remains within good suction distance and does not carry too much air or gas, there will be no difficulty in handling the three wells with one pump, providing the suction lines are properly connected and suitable arrangements are made for priming the pump. However, if the water goes below easy sucking distance, an air lift will be more satisfactory. To ascertain the approximate quantity of air that will be required, it is also necessary to know the amount the water will lower or the total lift above the water level in the well. The discharge pipe from the wells carrying the air and water should be run vertically to the full hight and equipped with separating tanks at the top, as horizontal or inclined pipes are not satisfactory in an air-lift system, the air tending to slide along on top of the water and thus reduce the efficiency materially.

If the total lift does not exceed 80 feet and a submergence of 160 feet or a ratio of 2 to 1 is used, with the proper arrangement of piping, the amount of free air used will be approximately 0.55 cubic foot per gallon of water. The air pressure required will be 72 pounds. If the total lift is 100 feet and a submergence of 200 feet can be obtained, approximately 0.65 cubic foot of free air will be required for each gallon of water pumped. About 90 pounds air pressure will be needed in this case. The above is based on the actual amount of air delivered at the well, and makes no allowance for any deficiency in the volumetric efficiency of the compressor. The air lifts can be operated with considerable less submergence. In that case, more air and less pressure will be required.

G. A. Reichard.

Los Angeles, Cal.

Operating Mechanical Tube Cleaners

I read with interest the letter by Mr. Guèrin in the August 9 issue of Power in which he tells of some of the troubles he has had with boiler-tube cleaners and unskilled labor.

I agree with him that the operation of removing scale from boiler tubes should not be entrusted to ash shovelers or any other cheap help about the plant as most tube cleaners need careful handling if all of the scale is to be removed without injuring the tubes. However, the cleaner that I am now using could not make as much trouble as the one Mr. Guèrin used.

My tube cleaner employs the revolving-head principle, that is, there are a number of cutter wheels revolving on a spindle and bearing out against the tube walls under the action of centrifugal force. By repeated tests I have found that it is impossible to injure the tube if this cleaner is allowed to run in one spot for some time as evidently the centrifugal force developed is not powerful enough to cause the cutter wheels to remove any of the metal or to spread the tube. In fact, no serious spreading of the tube could occur as the swing of the arms is limited, that is, they can only bear out to a diameter very slightly larger than that of the tube.

Frank J. McMahon.

New York City.

Boiler Inspection

The editorial in a recent issue of Power on boiler inspection is right to the point and I will venture to say that there is not an engineer reader of this paper who, at some time, has not seen boilers inspected in the loose, don't-care manner described. It is of vast importance to the engineer as well as to the owner that a thorough inspection be made by a competent inspector. There are a great many engineers who are not thoroughly posted and do not thoroughly understand this class of work, because they have only had the opportunity to examine a few boilers. The inspector, however, is at it every day, and practice makes him perfect.

I believe that all boilers should be inspected by a Government inspector with the assistance of the engineer or fireman in charge of the boiler regardless of the insurance inspectors and at regular intervals, say, at least twice a year. Also, that all engineers or firemen in charge of boilers be compelled to pass a satisfactory Government examination covering the individual classes of boilers under their charge. Further, that a heavy fine or imprisonment or both be the penalty to the manufacturer, owner and operator of any boiler that is found to be defective or that explodes. A committee of competent men who thoroughly understand this class of work should be employed by the Government to pass on all boilers that are found defective by the inspector or engineer and on all exploded or otherwise damaged boilers.

Change the inspector's job from a political one to a meritorious one either under the civil service or Department of Labor.

The above, in my belief, is the only way in which to stamp out incomplete inspections and incompetent inspectors. Naturally this will reduce the number of defective boilers in operation and thereby decrease the number of explosions and the accompanying loss of life and property.

J. Case.

Hyattsville, Md.

Diagrams for Criticism

In the issue of August 16, J. D. Robertson asks for information concerning his indicator diagrams. I notice that the admission is rather late on the crank end. Further, I note that the cutoff is a trifle early at both ends. All other points appear to be all right; so, if these two defects are corrected, the running of the engine will be improved. No doubt the wavy lines noticeable on these diagrams are due to vibrations in the motion of the pencil when a sudden change in pressure occurs. These effects are generally shown by all high-speed engines. They show that the indicator piston is in good condition and is working easily.

C. Trube.

McKeesport, Penn.

In the August 16 Power Mr. Robertson presents a set of diagrams for criticism. While they are far from being bad, certain points may be improved, notably the steam line up to the point of cutoff which at present is hard to locate. It is taken for granted that the steam pipe is of sufficient size and that the ports are ample. I would first advance the eccentric so as to give an earlier opening and cause the steam line to be more nearly horizontal to the point of cutoff. This will make a more definite cutoff and cause higher compression, which, at the present is equal to about two-thirds of the boiler pressure. If less compression is wanted, simply change the exhaust rods to suit.

I suppose that there is a definite reason for carrying the high backpressure which is indicated by the distance between the atmospheric line and the back-pressure line. The back pressure seems to be about one-eighth of the boiler pressure. If the latter is 150 pounds, this would mean about 19 pounds back pressure. I advise Mr. Robertson not to bother with the technical defects of the diagram but to seek every possible way of decreasing the excessive back pressure which has a coal-wasting influence.

A. C. Waldron.

Revere, Mass.

Creeping Water Meter

The article by A. E. Holman in the August 16 issue of Power, is of more than passing interest to me because of similar experiences that I have had.

In a factory where I was engaged in making some tests, it became necessary to shut off a ¾-inch disk type of meter for some three weeks. During this time it ran up a bill of about $30 for water passed. The proprietors of the factory very justly refused to pay. This caused a lot of trouble with the water board of the city. The meter was connected by valves on both inlet and delivery side. When it was shut off only the valve on the delivery side of the meter was closed so that the inlet side communicated directly with the supply pipe.

As I did not know anything of this trouble until after the tests on which I was engaged had been finished, there was no opportunity to attempt a solution of the queer action of the meter and it still sets me to guessing at times. While Mr. Holman's explanation may be the proper solution in his case, it does not fit the case I have just described for the delivery pipe beyond the valve had been disconnected, which was evidence that the valve did not leak.

In another case, a neighbor had a ½-inch disk meter in his house that jumped his quarterly water bills from about $6 up to $15 or more. I took one of my plunger meters and connected it beyond the disk meter so that no water could be drawn until it had passed both meters. My friend's bills dropped back to about $5 per quarter. Both meters registered practically the same, and for over 15 months they continued to do so.

My solution of this case is that when a tap was opened wide the water pressure, which is about 125 pounds, set the disk to spinning at such a rate that the momentum kept it going after the tap was closed, but when the plunger meter was connected it acted as a brake and prevented the disk from attaining such a high velocity. In making the connections I had to use a number of tees in place of elbows; these also acted as a check to the flow of the water.

A third case fairly disgusted me with disk meters I wanted to use a ½-inch meter in a great hurry and went to the superintendent of the city water works to rent a meter that I could carry along with me as I had no plunger meter that I could get to the job in time. The superintendent kindly let me have a new meter, one that had never been in service. The first thing that I did was to calibrate the meter. To my surprise, I got less than 43 pounds of water for each foot registered.

This experience made me skeptical about my house meter which had been in use some eight years. As soon as possible I made a test of the house meter and found it to be shy about nine pounds to the foot. I entered complaint at our water-board office, and a city expert (?) was sent to make an investigation. This he did by opening the tap in the kitchen sink and then looking at the meter which was apparently registering the flow. He went back to the office and reported the meter as being all right.

When I tried to get redress from the "thick necks" in the water-works office I was told there was no law in the country to compel me to use the water and that I could "take it or leave it," just as I pleased.

The foregoing is submitted to show why I have lost faith in disk meters, which were evidently placed on the market to supply a demand for something cheap, as well as to show that a "cheap" machine is no more reliable than a cheap mechanic.

W. H. Odell.

Yonkers, N. Y.

In the issue of August 16, A. E. Holman describes the case of a creeping water meter which is of much interest to water-works people and of general interest to all of us.

We have a similar case where a water meter, attached to a main at the top of a rise, as shown in the cut on page 1477, registers about double the amount of water that seems actually to be consumed.

We have been totally at a loss to account for this condition as we had always believed that a water meter could not register *more* than the amount of water being consumed or leaking through. In our case there is no hot-water tank attached to the house fixtures as shown in Mr. Holman's cut; the hot water is gotten from an instantaneous water heater.

I would be very glad to hear from other water-works men who have had similar trouble or who can give information about water meters registering more than the correct amount consumed.

Robert L. Ruddell.

Glenville, W. Va.

License Laws

I have read with interest for many years the printed discussions on engineers' license laws. So far, I cannot agree with the opinion generally expressed that the engineers should work to have laws enacted requiring them to pass an examination and secure a license before they are permitted to operate a steam plant.

The foremost reasons generally advanced by the advocates of license laws are, public safety and the making of more competent engineers.

If we are to take the foregoing reasons seriously, it would seem that the engineer is posing as a public benefactor in the first case and in the second case he is asking the public to insist on his being a better workman than he may be of more value to his employer. The inconsistency is obvious. If, being an engineer, the danger of having an incompetent man operate a steam plant is more apparent to him than it is to the public and, as a public-spirited citizen he sees and attempts to do his duty, then he is consistent as a public-spirited citizen. He is indeed a public-spirited citizen who will deliberately force upon himself a condition which compels him to apply himself vigorously many hours each night through a period of years to ground himself in a profession, the branches of which are ever growing wider and more intricate, and, after belaboring himself with work, worry and inconvenience, submits meekly, nervously to a scorching examination by the very men in whose hands he has deliberately placed the power to give or refuse superciliously the right to him to earn an honest living. I say that he is a generous soul to force this upon himself in the interests of the dear public with never a thought of self.

As for the second reason, that of making more competent engineers, that too shows a broad and generous spirit. It borders on the humorous when we ask the State to *compel* us to be that which we so earnestly desire to be. No doubt, it will be argued that it is the few ambitious ones that are advocating this in order to compel the laggards to apply themselves that they may become competent and proficient. But, I ask, *why* you are interested in the great body of laggards becoming competent and proficient? Your answer, coming from such a source, cannot be other than thoroughly unselfish, ennobling and inspiring. You would probably say that you wish to share with your fellow craftsman the beauties and pleasures which are enjoyed only by those whose mental faculties are developed to the highest degree.

I believe that it is not condescending to grant that the engineer is governed by the same impulses and motives as the rest of humanity. When we say that selfishness is a very pronounced characteristic of the human family and that the engineer has inherited his just share, I believe there is no exaggeration. The foregoing granted, it becomes necessary to expect from the engineer *some* human conduct. To be frank, I have been and am now of the opinion that this license agitation is a veiled, insidious move with no other end in view than to make scarce the supply of engineers that the scarcity may fructify to those remaining. I believe that the craft to a man thinks that a strict license law would increase the salaries of all and make conditions better and that this is the underlying motive for all this license agitation.

If we look at the engineer in those States which are strictly licensed we will find that his hours and wages are not any better than those in the no-license States.

I am sure we would all be interested in such data as it is possible to gather from examining boards and other sources, to find out whether the license laws have been of any real benefit to the craft. I think that they have not. On the contrary, I think that they have been an annoyance and an added worry.

In this city we have had an ordinance for the last five years which requires us to be licensed. In that time we have paid for renewals approximately $5000 and in return we have not received directly or indirectly five cents' worth.

Approximately 1 per cent. of the population of this city is employed as operating engineers. Assuming that this proportion holds throughout the country, it will be seen from the census, that the operating engineer, at an average of $1 per head, would be paying about $1,000,000 per year if we had a license law in every State. Why we should desire to contribute this fund and be pestered with examinations, is more than I can understand. The total amount of property destroyed probably would not equal in value the above named sum, and it would be unreasonable to expect accidents entirely to disappear in the licensed area. In fact, they are not eliminated in those States where the examinations are making the boys scream.

The engineer need not work himself into a frenzy for the safety of the public or for the protection of his employer's property, because it is plain that the public depends far more on the intelligence of the employer than on that of the engineer and the employer would be vexed if he knew that the engineer thought that he could injure his property without his permission.

J. J. Nigh.

New Haven, Conn.

Clearance and Compression

By V. Dwelshauvers-Dery

> *The author replies to Professor Heck's criticism of his article on compression, pointing out that that criticism was founded on the empirical indicator diagram for which isothermal compression and expansion of a perfect gas are assumed. In practice there is no perfect gas, isothermal compression and expansion are not realized, nor do actual compression and expansion follow the same law. These facts, the author argues, invalidate Professor Heck's argument.*

In Professor Heck's article, on page 1657 of the issue of September 13, he seems to think that my experiments have merely opened the question of the economy of compressing steam in the clearance. This in itself would be a success not to be despised. But his theory has not only neither solved the question nor closed the discussion; it has not even opened up the question, since it was done already. Although well presented, this theory rests upon not a single practical fact nor a single experimental result, but upon nothing more than the "traditional theory" of the steam engine, which estimates the work performed at 20 and even nearly 40 per cent.; and the consumption at about 100 or 200 per cent.

Indubitably, his remarks tend to cast doubt upon the general applicability of the conclusions derived from my experiments, but they do not even approach a discussion of them. To the facts that Nature offers he opposes hypotheses without experimental basis, and on this needle point he erects a monument, attractive perhaps, but which even a light breath may throw to the ground in a heap.

My engine, he says, is too small; the phenomena of heat exchanges between the wall and the steam are too pronounced; their laws differ from those observed in large engines. But when one wishes to study experimentally the laws of a phenomenon, is it wise to surround oneself with conditions that reduce it rather than with those that accentuate it? To assume that my engine was too small is to undertake a precise determination of the size at which the conclusions will begin to change or to be overturned. I calmly await this information; only it must be supported by experimental facts, and not merely by statements which one frames in his office—the only sort presented in Professor Heck's article. I shall render this evident, proceeding step by step and presenting again the diagrams accompanying that article.

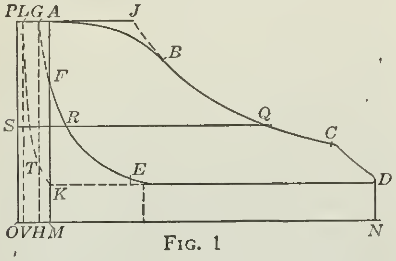


Fig. 1

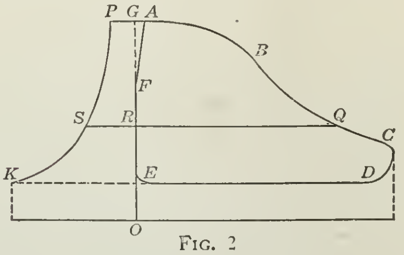


Fig. 2

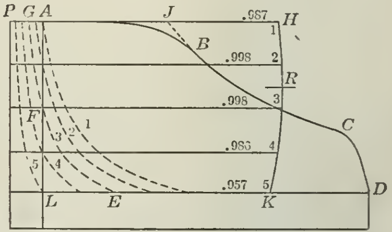


Fig. 3

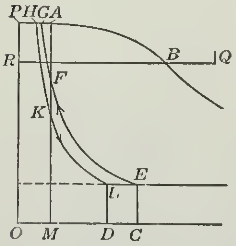


Fig. 4

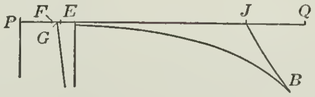


Fig. 5

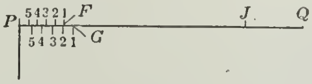


Fig. 6

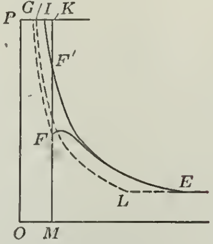
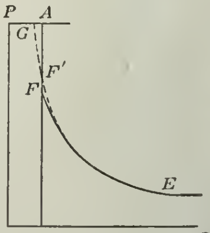


Fig. 7



Power

Fig. 8

Referring to Fig. 1, Professor Heck says: "In expansion, this steam is supposed to retrace the curve *G F E*, giving back the work that was expended in compressing it." This supposition is entirely gratuitous, since the *actual* law of expansion is not at all the same as that of compression. This fact is sufficiently demonstrated in POWER for June 28, 1910, on page 1164 and those following. If the law of expansion were the same as that of compression, there would be no room to discuss the economy of compression. Under that condition, *with complete expansion, complete compression would annul the detrimental effect of the clearance,** and incomplete compression or expansion could suppress only a part of it.

Professor Heck rests his entire argument upon the hypothesis that the expansion and compression of the steam act according to Boyle's law,

$$p v = \text{a constant},$$

a law which is applicable *only to a perfect gas and then only for an isothermal process.* But in steam engines, large or small, the temperature is constant neither during expansion nor during compression; and, moreover, the medium is a vapor which by no means resembles a perfect gas. In engines, large or small, there take place reëvaporations during expansion; and, during compression, the steam begins to become superheated until the instant when, having become hotter than the metal of the wall, it loses its superheat and begins to condense. These are *facts of experience* and not hypotheses; while the actual curves traced by an unconscious instrument, the indicator, are not equilateral hyperbolas.

Professor Heck says: "If the expansion curve is carried up to the initial pressure at *J*, the length *P J* shows the volume which the total steam in the cylinder would have at that pressure. Of this, *P G* is the clearance steam and *G J* is fresh working steam, so that *G J* is a measure of the steam received from the boiler." This statement is purely fanciful for several reasons, as follows:

First. During admission there has come from the boiler into the cylinder a certain weight of steam, a notable part of which is condensed, in order to compress the clearance steam, to reheat the metallic walls and to perform the work. The part condensed upon the walls, in large as well as small engines, is always considerable, so much so indeed that it has been given the special name of "missing quantity." It is reduced by various means which I shall not discuss here, such as jackets, superheating, etc. Without these it would amount to 20, 30 and sometimes even 50 per cent. of the weight of fluid coming from the boiler.

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*See page 1157, POWER, June 28.

Second. The extension $B\,J$ of the curve is arbitrary; if we are dealing with a curve given by the indicator, it will differ from one diagram to another; if it is drawn according to a hyperbolic curve, it is certainly false. Nothing is able to locate mathematically the position of the point J. Yet it is upon this position that Professor Heck rests his entire method. A single practical means is at hand to determine accurately the weight of steam present in the cylinder and the weight of condensed water on the inclosing walls at the moment of cutoff. From the indicator diagram and other experimental data are determined with precision the volume v occupied by the steam at cutoff and its pressure p; the steam tables give, for this pressure, the weight δ of unit volume, and the weight of the volume v is $v\delta$. The product is the weight of pure steam present. If it were necessary to determine the volume that this weight would occupy at the pressure $O\,P$, equal to p', we should look in the tables for the weight δ' of unit volume at this pressure, and should calculate the volume v' by the equation

$$v\delta = v'\delta'$$

which obviously transposes to

$$v' = v\frac{\delta}{\delta'}$$

From this is derived the dimension $P\,J$ in Fig. 1.

There is needed more than 25 and even 50 per cent. in order that the weight of steam represented by $P\,J$ or $G\,J$ shall be the same as that which has passed from the boiler into the cylinder, even in large engines running at high speeds, but without jackets, etc. For clearness I shall designate by W this last actual weight, measured in an experiment. We determine it exactly by means of a surface condenser where the condensed steam which has acted in the cylinder remains separate from the condensing water, and may therefore be weighed on discharge from the air pump. The engine is caused to run regularly during a time sufficient for the purpose; the number of revolutions made in this time is counted, and the water coming from the air pump weighed. Simple division gives the weight of steam that has come from the boiler and acted in the cylinder *per* revolution of the engine. *If the two piston strokes during one revolution are absolutely alike*, with regard to the clearance space as well as the point of cutoff, the weight of steam *per piston stroke* (W) will be equal to half of the weight *per revolution*. If the two piston strokes are not rigorously alike in every particular, nothing certain can be derived from the experiment.

It is necessary, on the other hand, to determine as accurately as possible the weight w, of the clearance steam, which cannot be ascertained directly from the experiment. The indicator diagram, taken with all necessary care, of course, shows the pressure and volume of the steam at the beginning of compression; with good steam tables one can deduce therefrom the weight,

$$v_o\,\delta_o = w$$

of pure steam in the cylinder at this moment, if the steam is saturated or wet. The degree of dryness or moisture cannot be determined unless the temperature of the steam at the same time is determined experimentally by means of an instantaneous thermometer, such as that of Armand Duchesne. Hirn supposed, and George Duchesne has demonstrated[*] that we may without danger of appreciable error assume that the steam is dry or slightly superheated because it is surrounded by walls hotter than itself. The fact has been proved experimentally upon the small engine of the Liège laboratory by Mr. Duchesne. Moreover, it is not disputed by Professor Heck, who says: "It is proper to take the clearance steam as dry at E."

Now, by the means just described, we determine the total weight $W + w$ of fluid present in the cylinder at cutoff, as well as the weight $v\,\delta$, of pure steam. What is called *quality* of the steam is then

$$\frac{v\delta}{W + w}$$

and the "missing quantity" is

$$W + w - v\delta$$

It is far from this to the pencil marks $B\,J$ and $F\,G$ arbitrarily given on a scrap of paper. However attractive may be the easy theory of Professor Heck, it may be pronounced unreliable in practice.

Professor Heck's Figs. 2 and 3

All the curves of compression in Figs. 2 and 3 are traced according to the law $p\,v$ = a constant, which never followed in any engine, whether small or large. They are used to measure the work just as the position of the point J is used to measure the quantities of steam; they deserve no confidence whatever. The method by which the figure is traced conduces only to a fictitious explanation of the problem. The work called effective $A\,B\,C\,D\,E\,F\,A$, is not that which the indicator would give, but only that which the author of the diagram has imagined. It is the same with the volume $G\,J$ to which this work is related. And, even in these imaginary conditions, the curve $K\,R\,H$, Fig. 3, is still very slight; the maximum abscissa R differs little from the smallest, and even if this diagram were based on a solid foundation, the numerical conclusions derivable from it should be regarded with caution; the more so since no account is taken of the thermal action of the walls.

Fig. 4

Professor Heck says: "It is proper to take the clearance steam as dry at E; if compressed along the curve $p\,v$ = a constant, it will be slightly wet at G." So he thinks that we can compress the steam along the curve $p\,v$ = a constant. But we cannot compress it as we would like; we compress it according to the law which Nature imposes and which the indicator is suitable to show; and this law is never that of Boyle in steam engines. My experiments, like those of the brothers Duchesne, demonstrate positively that from the beginning of compression the steam becomes superheated more and more, both by the work expended upon it and by the heat from the hotter walls; this latter effect is, however, small, because of the gaseous condition of the fluid in contact. By this double action, the steam becomes hotter than the metal and yields some heat up to it if there is time, that is to say, if the compression is sufficiently great; it loses some of its superheat, and if the least drop of it is condensed, condensation immediately progresses with great rapidity. It is this which shows the final hook, attributed at first to piston leaks, in which Professor Heck is still disposed to believe in spite of the irrefutable experimental proof by George Duchesne (POWER, previously cited, page 1165). As to this, I shall say a few words hereafter.

Professor Heck's Figs. 5 and 6

In all his statements, Professor Heck assumes the curve of compression to have the same geometrical form as that of expansion, which is never realized in practice. As I have already said, if that were so the question of compression economy would be solved and the hypotheses of Fig. 4 would be useless. In Figs. 5 and 6 there are applied the same arbitrary methods in choosing points, the same want of precision, lack of experimental data—in short, nothing reliable.

Figs. 7 and 8

That in ordinary cases the compression curve falls below the equilateral hyperbola is a simple fiction like the hyperbola itself. The actual fact is that, in ordinary cases, the compression is small and its effect little apparent. It tells nothing either for or against my experiments.

"In the Dwelshauvers-Dery experiments," says Professor Heck, "the engine gave a compression curve like $E\,F$ in Fig. 7; and the whole cycle for the clearance steam *must have been* of the form *E F G H L E*" there sketched. I greatly doubt that Nature obeys the mandate of Professor Heck, and that it follows the law $pv =$ a constant, to accomplish the cycle. As already stated, this law is never followed in an engine, and is applicable only to perfect gases, which do not exist, and isothermal changes of volume, which are impossible throughout an entire mass inclosed between metallic walls which have no other heat than what they receive from the working fluid.

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[*] Pages 1162 and 1163, POWER for June 28.

To pretend to separate the action of the clearance steam from the other by the method which Professor Heck has proposed, is going too far; and I could not approve it unless he based it upon experiments made with large engines as accurately as upon my small one. Our experiments on this latter have shown positively that the steam remaining in the cylinder at the end of the exhaust stroke, in contact with walls hotter than itself, was certainly free from any moisture and probably a little superheated; further, that compression raises the temperature more and more, until, with sufficient compression, it reaches and exceeds that of the walls, following which there may be produced condensation, causing a final hook in the indicator diagram. Are there plausible reasons why the size of the engine should modify these processes? I see none. There is, on the contrary, a reason why the phenomena would have time to take place in one size of engine as well as in another; it is that at the dead-center position the speed of the piston is always nothing.

"There is still some room for skepticism as to the entire absence of leakage from the experimental engine at Liège." Upon what does Professor Heck base this insinuation, the tendency of which is evidently to cast discredit upon our experiments? Upon nothing that has been published concerning this engine, but solely on what relates to the engine of Messrs. Calendar & Nicholson. I feel impelled, then, to inform him that for years, myself believing in loss by reason of the final hook in the compression curve, I did everything possible in order to reveal it, discover it, measure it and avoid it. I procured a fundamental investigation of my engine by the celebrated mechanician Walschaerts, who found nothing more to be said after numerous and careful tests. Having arrived at conviction as to the distribution valves, I wished to examine the piston. For this purpose I ran the engine as a single-acting one, with the rear head removed; holding a light in my hand and following the movement of the piston, I closely scrutinized the entire circumference, clear to the far end, and thus convinced myself that there was no piston leakage within the scope of my senses. Anyone investigating the matter with the same rigor would have arrived at the same conviction. But any reader who has never seen my engine is still capable of defying my statements and maintaining doubt. There was necessary a proof written by the engine itself, an irrefutable witness. It was George Duchesne who furnished it to me, clear, evident, leaving no more room for doubt. Instead of having the engine compress *condensable steam*, he caused it to compress *noncondensable air*, and then all the hook disappeared from the indicator diagram taken in numerous examples. This is stated and illustrated in the issue of Power above cited, on page 1165.

How does Professor Heck, who seems to believe in his theory, produced purely by his imagination, refuse to give way before a natural, patent, evident fact? If he will only repeat the experiment upon a large engine without leakage, and with sufficient compression of steam to produce a final hook in the diagram, and then will run the engine with the same compression of noncondensable air, he will see that the compression of the air does not produce any hook.

The National Engineers' Convention

> *The Convention at Rochester was the largest and best ever. The exhibits numbered 175 and the attendance close to 1000. Carl S. Pearse, of Denver, becomes president and Edward H. Kearney, of Boston, was elected vice-president.*

At 10 o'clock Tuesday morning, September 13, the twenty-eighth annual convention of the National Association of Stationary Engineers opened in the armory building at Rochester, N. Y., with a large attendance of delegates and visitors. William J. Ranton, chairman of the local committee, presided over the introductory exercises. The Rev. E. P. Hubbell offered prayer, and the chairman then presented Mayor H. H. Edgerton, who warmly welcomed the convention to Rochester. National President William J. Reynolds responded on behalf of the engineers.

The next speaker was George Dietrich, president of the Chamber of Commerce, who said the growth of the National Association of Stationary Engineers was one for pardonable pride and that its preamble merited such a name as the "Coöperative Brotherhood," because of its endeavors to be valuable in industrial life.

The response to Mr. Dietrich was made by National Vice-president Carl S. Pearse, who told of the discovery of the section of our country now known as the Empire State, alluding to the discoveries of Henry Hudson. Then came a brief review of the colonial period followed by a laudatory account of Fulton's achievement with the "Clermont," America's first steamboat.

Past National President Robert G. Ingleson said that each generation had problems peculiar to its day, and that all methods had been tried, but that the method of the present, that of organization, was the only sane one to get results that promise progress and stability.

The next speaker was Senator George L. Meade, who, after eloquently picturing the importance of the engineer's work in life and what the world owes to him, said he believed that a licensed competent engineer is a far better protection against boiler explosions than any system of inspection that could be devised. Also that the granting of licenses in all professions and trades had elevated the standard of all so licensed. He urged the engineers to stick, even for years, and success must eventually come.

Response was made by E. E. Pruyn, secretary of the New York State Association, National Association of Stationary Engineers, who interspersed the serious portions of his talk with humorous stories.

Before turning over the meeting to National President Reynolds, Chairman Ranton presented the president with a silver-mounted ivory gavel which was accepted with a few well chosen remarks.

The convention was then formally opened. The first order of business being the appointment of the regular convention committees, consisting of:

Ways and Means: F. L. Ray, William H. Sennett, O. S. Wiley, A. A. Garrett and Edward Lawler.

Auditing: John F. McGrath, C. W. Naylor and George W. Brownhill.

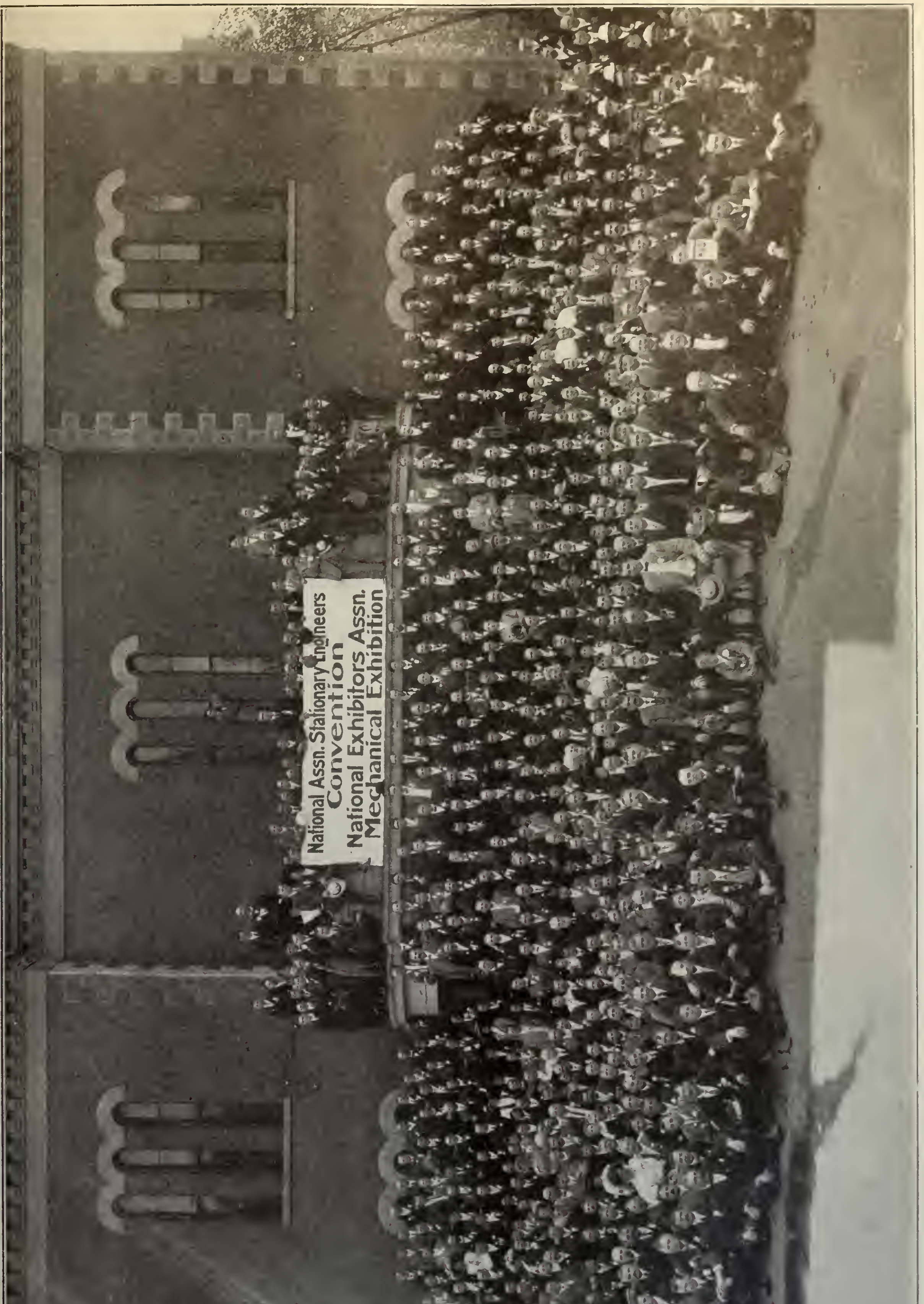
Mileage: J. Singleton, J. W. Varden, Thomas Swayne, George W. Roff and H. F. Morningstar.

Ritual: J. M. Williams, F. H. Munsberg and W. A. Lockett.

Credential: W. P. Weiman, W. V. R. Coon and J. H. Mullen.

The convention then adjourned until 2:30 p.m., and immediately the delegates and guests assembled in the exhibition hall, where National Vice-president Carl S. Pearse in a neat and complimentary address formally opened the exhibit of the National Exhibitors' Association.

The afternoon session convened at 2:25 o'clock. The first order of business was the preliminary report of the cre-


National Assn. Stationary Engineers
Convention
National Exhibitors Assn.
Mechanical Exhibition

Delegates and Visitors to the National Association of Stationary Engineers' Convention at Rochester, N. Y.

dential committee showing an attendance of 351 delegates.

Then came the annual report of the national president, which was referred to the committee on analysis and distribution.

The report of Secretary Raven showed that he had received and transmitted to the treasurer nearly $25,000. That seven new subordinate associations had been formed and that five had surrendered their charters. Two associations in Connecticut had consolidated as had also two in New Jersey. This report was referred to the auditing committee and was followed by the report of National Treasurer Forse, who reported receipts and disbursements for the year, leaving beside the protection fund a bank balance of over $15,000.

A Group of "Big Ones" at the Convention

Walter E. Sanders, Cleveland, Ohio.
195 lb.—5 ft. 10 in.
Herbert E. Stone, New York City.
267 lb.—6 ft. 2 in.
Louis F. Hoffman, New York City.
296 lb.—6 ft. 4 in.

A. C. Conway, Cleveland, Ohio.
200 lb.—6 ft. 7 in.
A. Van Vredenburgh, Pawtucket, R. I.
244 lb.—6 ft. 8 in.
Theodore Hoff, Chicago, Ill.
274 lb.—6 ft. 7 in.

P. E. Merriman, Methuen, Mass.
220 lb.—6 ft. 4½ in.
James L. Band, Ansonia, Conn.
320 lb.—6 ft. ½ in.
Chester W. Dibble, New York City.
187 lb.—5 ft. 8 in.

Secretary Raven then read the report of the educational committee which was accepted. The license committee's report covered the past year's work and was promptly accepted and the committee given a vote of thanks.

Invitations were received to visit the power stations of the Rochester Railway and Lighting Company and the Vacuum Oil Company, and the courtesies of the Masonic Club were extended to the delegates.

Permission was given a representative of the National Exhibitors' Association to present a paper to the convention, advocating a better understanding and coöperation between the National Association of Stationary Engineers and the National Exhibitors' Association. The paper was referred to the committee on ways and means. The convention then adjourned until Wednesday morning. At this session the first order of business was the final report of the credential committee giving the list of delegates as 357. An invitation from Adler Brothers to visit their plant was received and accepted, the visit to be made on Friday just after the morning session of the convention.

Proposals to amend the constitution were presented.

A committee was appointed to arrange the proposals which were similar in principle so that the convention could act upon them in an understandable manner and without loss of time, the committee consisting of C. W. Naylor, E. J. Maroney and F. L. Johnson. The committee was instructed to have such proposals printed and distributed to the delegates before their submission to the convention, and thereafter submission to a referendum of the subordinate associations.

At the afternoon session the first matter for consideration was the report of the trustees of the protection fund, which stated that it had not been necessary to expend any of the fund during the past year.

The board of arbitration reported that no cases had come before it for consideration.

Next in order was the report of the board of trustees of the *National Engineer*.

Consideration and discussion of this report was postponed, and attention was given to the report of the pension-fund committee, which was in substance:

There is in the treasury of the Ladies' Auxiliary $4000 which they desire to turn over to the National Association of Stationary Engineers for the purpose of establishing a pension fund. We recommend that this money be received from the Ladies' Auxiliary, to be held in trust by the president, vice-president, secretary and treasurer of the National Association of Stationary Engineers for the specific purpose of establishing a pension or relief fund, until such time as suitable legislation can be enacted by the National Association of Stationary Engineers.

The report was accepted and the recommendation adopted, after which the convention adjourned.

On Thursday morning the delegates listened to the report of the board of trustees. The mileage committee reported the total mileage of the 360 delegates as 249,182.

The privileges of the convention were accorded to the Ladies' Auxiliary. The national president, Mrs. Nella C. Moore, addressed the convention on the question of the pension fund. Among the comments offered was that the convention should make a decision as to the establishment of such a fund, making it part of the association work. She said that referring the matter to a referendum vote was dilatory and very discouraging to those who have for years sought to have the matter placed on a definite basis without unnecessary delay.

A motion was adopted that the president, secretary and treasurer of the National Association of Stationary Engineers be a board of trustees to arrange for the organization of a pension department in conjunction with the officers of the Ladies' Auxiliary.

The next business was the selection of the meeting place of the 1911 national convention, the cities contesting for the honor being Cincinnati, Peoria, Ill., and Mobile, Ala. Cincinnati won by 75 votes over Mobile, its nearest competitor, Cincinnati getting 185, Mobile 110 and Peoria, 6.

ENTERTAINMENT FEATURES

There were several automobile tours to places of interest through the city.

On Tuesday evening there was a theater party at the Lyceum theater.

On Wednesday evening at the convention hall an entertainment was given by the "Bunch."

On Thursday afternoon special cars conveyed the delegates and guests to Manitou beach. A baseball game was played between the engineers and supplymen, which resulted in a victory for the engineers.

The entertainment features were brought to a close on Friday evening by the usual grand ball, which, on this occasion, was a great success.

PRESENTATIONS

During the opening session of the convention, National President William J. Reynolds was presented a silver-mounted ivory gavel from the local associations of Rochester.

After the installation of officers on Friday evening, Past National President Reynolds was presented a $250 diamond ring and a past president's jewel.

The members of the New Haven Association, No. 2, of Connecticut, presented delegate Edward J. Maroney with a diamond horseshoe stick pin before his departure for Rochester.

Past President Joseph F. Carney received a handsome clock as a gift from his many friends.

The newly elected vice-president, Edward H. Kearney, was the recipient of a beautiful floral tribute from his admiring friends.

Past National President P. H. Hogan was also remembered by the testimonial-giving crowd.

Nella C. Moore, president of the ladies' auxiliary, was given a valuable cut-glass berry dish.

The George W. Lord Company donated $50 to the Ladies' Auxiliary for their pension fund.

The Seneca hotel was the headquarters of the convention. At the entertainment on Wednesday evening, the proprietor, M. E. Wolf, was tendered a loving cup by the convention guests.

OFFICERS ELECTED

The national officers elected for the ensuing year were: Carl S. Pearse president, Denver, Colo.; Edward H. Kearney, vice-president, Boston, Mass.; Fred W. Raven, secretary, Chicago, Ill.; Samuel B. Forse, treasurer, Pittsburg, Penn.; Joseph J. Gibney, conductor, Mobile, Ala.; W. H. Lockett, doorkeeper, Dallas, Tex.

J. D. Taylor was reëlected secretary-treasurer of the life and accident department and the salary fixed at $1200 for the ensuing year. James G. Beckerleg was reëlected to the board of trustees, and was retained as chairman. J. A. Kerley was reëlected to the board of trustees of the *National Engineer*.

THE EXHIBITS

The exhibition of power-plant specialties and supplies held under the auspices of the National Exhibitors' Association was, as usual, one of the most prominent and attractive features of the convention. One hundred and thirty manufacturers occupied one hundred and seventy-three booths in the exhibition hall, which adjoined the convention hall. The entrances were so arranged that the delegates were compelled to pass through the exhibition hall to gain entrance to the hall where the convention held its sessions. This was a most desirable feature from the exhibitors' point of view, and it is hoped that this plan will be adopted at all future conventions.

The exhibition hall was laid out with booths of uniform size but so arranged that two or more booths could easily be thrown into one for those who desired additional space. Wide aisles were laid out so that there was ample passage room between each group of booths. The decorations were uniform in green and white, no other decorative colors being allowed in the hall, making a most pleasing effect. The hall was of limited size so that it was necessary to use two floors for the exhibition.

The following is a list of the exhibitors and their representatives in attendance:

Acheson Oildag Company, Niagara Falls, N. Y. Represented by R. B. Clark, E. G. Acheson, Jr.

Alberger Condenser Company, New York. Represented by S. C. Ross.

Allis-Chalmers Company, Milwaukee, Wis. Represented by R. D. Tomlinson, C. R. Tatem, Rex T. Stafford, F. C. Randall, L. L. Willard, C. A. Burns, A. P. Peck, E. T. Pardee, H. L. Watson, M. H. Merrill, F. G. Bolles, F. S. Sly, W. I. Lucius. They held a guessing contest for the number of blades in a 1000-kilowatt turbine. Sixty sterling silver prizes were distributed.

American Radiator Company, Bundy Department, New York. Represented by W. E. Van Keuren, E. M. Stevens, A. S. Lessen, A. E. Jones.

American Radiator Company, Specialty Department, Chicago, Ill. Represented by E. S. Storm, F. C. Reeder.

American Ship Windlass Company, Providence, R. I. Represented by R. S. Riley, Messrs. Marble, Mallory, Alpern.

American Steam Gauge and Valve Manufacturing Company, Boston, Mass. Represented by N. H. Nickerson, R. S. Pollard. Souvenir; celluloid indicator scale.

Anderson Company, V. D., Cleveland, O. Represented by Joseph H. Meyer, R. E. Bronson.

Arrow Boiler Compound Company, St. Louis, Mo. Represented by F. A. Foster, E. M. Swindler. Souvenir, watch fob.

Ashton Valve Company, Boston, Mass. Represented by Harry Ashton, Columbus Dill, C. W. Houghton.

Austin Separator Company, Detroit, Mich. Represented by H. J. McGregor.

Best Manufacturing Company, Pittsburg, Penn. Represented by Charles K. Thomas.

Bird-Archer Company, New York. Represented by William J. Schatz. Souvenir, link-cuff buttons with official button of the National Association of Stationary Engineers.

Builders Iron Foundry, Providence, R. I. Represented by N. L. Sammis.

Cancos Manufacturing Company, Philadelphia, Penn. Represented by Harry Harmstadt. They held a prize drawing for a gold watch.

Chesterton Company, A. W., Boston, Mass. Represented by H. D. Raymond.

Chicago Die and Stamping Company, Chicago, Ill. Represented by Robert Edwards.

Claflin & Co., Charles A., Boston, Mass. Represented by Charles A. Claflin, Merrill N. Davis.

Cling Surface Company, Buffalo, N. Y. Represented by William D. Young, T. H. Clark.

Cook's Sons, Adam, New York. Represented by M. V. Peavy, George E. Tanberg, W. D. Naughton. Souvenir; pencils, balloons, court plaster.

Crandall Packing Company, Palmyra, N. Y. Represented by John M. Chapman, John Kaymel, Stephen Nauerth.

Crane Company, Chicago, Ill. Represented by J. L. Forsyth, W. L. Oswald.

Crowe Iron Works Company, Paul L., Jersey City, N. J. Represented by Paul L. Crowe.

Davidson Company, M. T., New York. Represented by John Lowe, Thomas Ellerby, Alden L. Corill. Souvenir; paper cutter.

Dearborn Drug and Chemical Works, Chicago, Ill. Represented by Herbert E. Stone, P. H. Hogan, J. N. Gregory, Dan

National Assn. Stationary Engineers
Convention
National Exhibitors Assn.
Mechanical Exhibition

Members of the National Exhibitors' Association who Were on the Job at Rochester

Delaney, J. G. Beckerleg, J. W. Harkins. Souvenirs; perfume and pocket screw driver.

Detroit Lubricator Company, Detroit, Mich. Represented by C. H. Crockett.

Dixon Crucible Company, Joseph, Jersey City, N. J. Represented by L. W. Brooks, John A. Condit, Dudley Thurston. Souvenir; pencils.

Dyna-Como Company, Boston, Mass. Represented by William I. Maude, J. D. Baldwin, D. B. Fox.

Eagle Oil and Supply Company, Boston, Mass. Represented by John L. Hamilton, ably assisted by Mrs. Hamilton. Souvenirs; pocket memo. book, good-luck coin, Perolin noise maker.

Eddy Foundry Company, R. M., Chicago, Ill.

Elliott Company, Pittsburg, Penn. Represented by Henry F. Weinland. Souvenir; pocketbook.

Ernst & Sons, Louis, Rochester, N. Y., representing J. H. Williams & Co., Brooklyn, N. Y., and F. E. Wells & Sons. Represented by August L. Hall, Louis J. Ernst, Joseph L. Ernst.

Fairbanks Company, New York. Represented by L. B. Mann, H. Jeffrey. Souvenirs; hat brush, tape measure.

Federal Metallic Packing Company, Boston, Mass. Represented by C. M. Wheaton, H. W. Rand. Souvenir; joke pocket mirror.

Ferguson Publishing Company, New York. Represented by A. C. Oliver.

France Packing Company, Philadelphia, Penn. Represented by A. W. France, J. F. Boreland. Souvenirs; pencils, stick pins.

Garlock Packing Company, Palmyra, N. Y. Represented by F. P. Morton, F. S. Mosher, F. J. Highmoor, T. C. Green, John N. Todd. Souvenir; leather bill fold.

Goade Water Filter and Purifier Manufacturing Company, Allentown, Penn. Represented by R. J. Goade.

Goulds Manufacturing Company, Seneca Falls, N. Y. Represented by F. T. Wells, H. W. Gumbert, Creed W. Fulton. Souvenirs; steel and celluloid rules.

Gould Storage Battery Company, New York.

Greene, Tweed & Co., New York. Represented by F. E. Ransley, B. M. Bulkley.

Hamler-Eddy Smoke Recorder Company, Chicago, Ill.

Hartford Steam Boiler Inspection and Insurance Company, Hartford, Conn.

Haverstick & Co., Rochester, N. Y. Represented by Frantz Haverstick, Charles Hery, Charles A. Schwonke, Frank E. Lawn, Henry J. Smith, William F. Sullivan, E. M. Keyes.

Hawk-Eye Compound Company, Chicago, Ill. Represented by H. E. Tibbles, Walter Tibbles, R. M. Rochfort, R. S. Everitt. Souvenirs; calendars, watch fob.

Hills-McCanna Company, Chicago, Ill. Represented by Robert E. Hills, Thomas A. Delaney.

Home Rubber Company, Trenton, N. J. Represented by A. R. Foley, Charles H. Swoger. Souvenirs; Foley's fly flowers.

International Acheson Graphite Company, Niagara Falls, N. Y. Represented by Archer C. Gibbons, M. A. Smith. Souvenirs; watch fob, celluloid rule.

International Text Book Company, Scranton, Penn. Represented by J. D. Jones, R. A. Davidson, E. S. Hammond.

Jenkins Brothers, New York. Represented by William G. Le Compte, H. C. McLaughlin, Billy Murray, Charles Beckwith, Frank Martin.

Johns-Manville Company, H. W., New York. Represented by B. C. McClure, E. W. Drew, H. P. Domine, Mr. McKinley.

A. C. Judd, Cleveland, O., representing American Huhn Metallic Packing Company. Represented by A. C. Judd, F. J. Davis. Souvenirs; knife sharpener, pencils.

Kennedy Valve Manufacturing Company, Elmira, N. Y. Represented by William Martin.

Kennicott Company, Chicago Heights, Ill. Represented by W. T. Runcie.

Kent Lubricating Company, Milwaukee, Wis. Represented by L. H. West, J. Morrison Gilmour. Souvenir; flag pin.

Kerbaux Company, New York. Represented by Irving D. Lyon.

Keystone Lubricating Company, Philadelphia, Penn. Represented by C. A. Hopper, T. W. Armstrong, H. L. Carpenter, A. D. Page, M. B. Urquhart. Souvenirs; clothes brush, cap.

Kitts Manufacturing Company, Oswego, N. Y. Represented by Arthur S. Light, T. N. Malcolm, Houston Hill, W. A. Kitts. Souvenir; celluloid rule.

Lagonda Manufacturing Company, Springfield, O. Represented by M. M. Sellers, James H. Johnson, J. P. Maloy, M. O. Lewis, W. J. Patterson, Hart Hill, C. F. Overly. Souvenir; watch fob.

Lake Erie Boiler Compound Company, Buffalo, N. Y. Represented by E. C. Schwingel, J. R. Pratt, George J. Juhre. Souvenirs; celluloid rule, pencil cigar.

Liberty Manufacturing Company, Pittsburg, Penn. Represented by H. A. Pastre, Karl Sixt. Souvenir; pocket book.

Lippincott Steam Specialty and Supply Company, Newark, N. J. Represented by A. C. Lippincott. Guessing contest for a complete indicator outfit.

Lloyd Manufacturing Company, Menominee, Mich. Represented by C. N. Dalrymple.

Lord Company, George W., Philadelphia, Penn. Represented by Edward McCarthy, B. L. Lord, Roy Downs, H. L. Gaskell.

Lunkenheimer Company, Cincinnati, O. Represented by Lane Thompson, assistant sales manager; W. W. Beal, A. S. Phillips, William Schmidt, Jr., E. P. Could. Souvenir; cloth cap.

Lyons Boiler Works, De Pere, Wis. Represented by W. P. Lyons.

McCrea & Co James, Chicago, Ill. Represented by M. B. Skinner.

McLeod & Henry Company, Troy, N. Y. Represented by John H. Foote, Charles N. Morgan. Souvenir; 2-foot folding boxwood rule.

McVicker Company, W. B., New York. Represented by W. B. McVicker, John McDonald. Souvenirs; card case, silver collapsible drinking cup.

Manning, Maxwell & Moore, New York, representing Ashcroft Manufacturing Company, Consolidated Safety Valve Company, Hayden & Derby Manufacturing Company, Hancock Inspirator Company. Represented by L. M. Brigham, Frank J. Corbett, Harry B. Aller, W. P. Bradbury, C. Monroe Smith.

Mason Regulator Company, Boston, Mass. Represented by Eldon MacLeod, Frank M. Morrison.

Mechanical Rubber Company, Cleveland, O. Represented by H. T. Fryant, Charles B. Iler, A. C. Conway, R. F. Valentine. Souvenirs; pipes, glass and stein mats.

Michigan Lubricator Company, Detroit, Mich. Represented by W. E. Bryant.

Mining World, Chicago, Ill. Represented by J. Winchester Holman.

Monarch Valve and Manufacturing Company, Springfield, Mass. Represented by R. E. Brown.

National Engineer, Chicago, Ill. Represented by Robert E. Larkin, L. L. Rice, John W. Lane, W. T. Mug.

Neemes Brothers, Troy, N. Y. Represented by Spencer Neemes, Oscar G. Neemes.

Nelson Valve Company, Philadelphia, Penn. Represented by Carlisle Mason, R. Bonnell, W. J. Spencer, R. E. Thomas, G. E. Huxley.

New York Belting and Packing Company, Ltd., New York. Represented by Charles A. Wilhoft, Carl P. Kock, F. E. Friedlander. Souvenirs; knife, Manual of Steam Engineering.

North Brothers Manufacturing Company, Philadelphia, Penn. Represented by F. A. Mutchmore.

New York & New Jersey Lubricant Company, New York. Represented by J. D. Kelrie. Souvenir; pipe and tobacco.

Ohio Blower Company, Cleveland, O. Represented by H. H Lind.

Ohio Brass Company, Mansfield, O. Represented by W. T. Jameson, C. N. Allerding.

Ohio Grease Lubricant Company, Londonville, O. Represented by F. W. Rogers, J. C. Ross, George A. Mull. Souvenir; postal cards.

Osborne High Pressure Joint and Valve Company, Chicago, Ill. Represented by W. L. Osborne, C. M. Schneider, W. A. Tilley.

Otis Elevator Company, New York. Represented by E. R. Pike.

Peerless Rubber Manufacturing Company, New York. Represented by Stephen Roberts, Charles Heitzmann, H. L. Brown,

J. Voltz, F. E. Bigelow, James Mason. Souvenir; watch fob.

Penberthy Injector Company, Detroit, Mich. Represented by F. O'Leary, Edwin Robinson. Souvenirs; watch fob. memo book, steel tape measure.

Peterson Engineering Company, New York. Represented by J. W. Peterson.

Philadelphia Grease Manufacturing Company, Philadelphia, Penn. Represented by T. Addison Collins, Horace A. Smith. Souvenir, postal cards.

Positive Differential System Company, New York. Represented by James A. Donnelley, Charles A. Fuller.

Power. Represented by John A. Hill, Jack Armour, William Van Kleeck, Charles N. Beard, H. K. Fisher, Fred L. Johnson, R. S. Richardson, J. D. Tighe, H. E. Cleland, Chester W. Dibble. Souvenir; identification badges.

Powell Company, William, Cincinnati O. Represented by Charles E. McFarlin, Charles Cullen, W. J. Leonard. Souvenir; watch fob.

Practical Engineer, Chicago, Ill. Represented by E. R. Shaw, A. L. Rice, C. S. Clarke, C. B. Leech, S. W. Anness, John A. Kershaw, H. W. Williams. Souvenir; memo book with directory of the exhibits.

Quaker City Rubber Company, Philadelphia, Penn. Represented by W. D. Purcell, S. B. Johnson. Souvenir; watch fob.

Restein Company, Clement, Philadelphia, Penn. Represented by A. H. Foster, D. D. Fernell. Souvenir; watch fob.

Revere Rubber Company, Chelsea, Mass. Represented by J. W. Dugeon, R. C. Whitmore, E. D. Lombard, J. W. Lawrence. Souvenirs; memo book, watch fob.

Richardson-Phenix Company, Milwaukee, Wis. Represented by Robert R. Keith, J. S. Ward, T. M. Walton, Mr. Tomlinson.

Ridgway Dynamo and Engine Company, Ridgway, Penn. Represented by R. C. Eccleston, Peter G. Rimmer.

Rochester Rubber Company, Rochester, N. Y. Represented by H. L. Dusky, F. E. Bigelow, G. N. Wilkinson.

Roebling's Sons Company, John A., New York. Represented by John W. Troescher, G. W. Swan.

Ross Valve Manufacturing Company, Troy, N. Y. Represented by William Ross, Adam Ross, J. C. Ross.

Ryerson & Sons, Joseph T., Chicago, Ill. Represented by D. M. Ryerson, J. T. Corbett, G. D. Branston. Souvenir; paper weight.

S-C Regulator Company, Fostoria, O. Represented by N. G. Copley, John F. Robertson.

Scully Steel and Iron Company, Chicago, Ill. Represented by John H. Allen, F. C. Farnsworth, W. H. Dangel.

Standard Oil Company of New York. Represented by C. A. Salisbury, C. S. Thayer, G. E. Roe, F. L. Damon.

Sterling Lubricator Company, Rochester, N. Y. Represented by John Buckley, F. C. Bulkley.

Sterling Oil Company, Rochester, N. Y.

Strong, Carlisle & Hammond Company, Cleveland, O. Represented by F. H. Evans, A. B. Pinney, C. H. Camp, J. C. Murdock, Homer Whelpley, L. P. Strong, F. S. Palmer, Charles Van de Velde. Souvenir; caps.

Southern Engineer, Atlanta, Ga. Represented by J. Willett Smith, W. S. Cushion, Lewis L. Warren, W. S. Dobie. Souvenir; cotton boll.

Squires Company, C. E., Cleveland, O. Represented by C. E. Squires. Souvenir; memo book.

Thacher & Co., George H., Albany, N. Y. Represented by C. T. Coe.

Thermoid Rubber Company, Trenton, N. J. Represented by Fred Kuhl. Souvenir; packing knife.

Trill Indicator Company, Corry, Penn. Represented by W. L. Trill, L. T. McElroy. Souvenir; hook and cord adjuster.

Under-Feed Stoker Company of America, Chicago, Ill. Represented by W. T. Johnson, W. R. Faber. Souvenirs; memo book and card case, ladies' shopping list.

Underwood & Co., H. B., Philadelphia, Penn. Represented by Edwin J. Rooksby, Edmund C. Crawford. Souvenir; book of useful hints for engineers.

Union Oil Works, Rochester, N. Y.

United States Graphite Company, Saginaw, Mich. Represented by F. J. Lee, G. N. Southard. Souvenir; pencils.

Universal Lubricator Company, Philadelphia, Penn. Represented by J. F. Lewis. Souvenir; playing cards.

Vanda Company, New York. Represented by Hudson Dickerman, Messrs. Hasenderer and Kascher. Souvenir; watch fob.

Von Kokeritz & Co., R. G., New York. Represented by J. A. Carson, Harry Wigley.

Warren Lubricant Company, New York. Represented by Albert J. Squier, M. M. Sanderson, F. P. Greenlund.

Westinghouse Air Brake Company, Pittsburg, Penn. Represented by M. H. Burchard, F. H. Parke, S. H. Mills, H. B. Bradford, H. E. Chilcoat. Souvenir; cuff buttons.

Westinghouse Electric and Manufacturing Company, Pittsburg, Penn. Represented by E. D. Kilburn. Souvenir; cuff buttons.

Westinghouse Machine Company, Pittsburg, Penn. Represented by R. A. Miller, G. D. B. Van Tassel. Souvenir; cuff buttons.

Wickes Boiler Company, Saginaw, Mich. Represented by James Milne, C. F. Wilson.

Williams Company, G. H., Cleveland, O. Represented by G. H. Williams.

Williams Gauge Company, Pittsburg, Penn. Represented by C. E. Wolff, Jr., F. E. Cox.

Williams Valve Company, D. T., Cincinnati, O. Represented by H. J. Ernst, D. T. Williams, L. W. Kramer, Charles Becker. Souvenir; combination pocket book and card holder.

Wright Manufacturing Company, Detroit, Mich. Represented by H. J. McGregor.

Ziermore Regulator Company, Johnsonburg, Penn. Represented by W. E. Zierden.

The election of officers for the National Exhibitors' Association resulted as follows: President, William D. Purcell, Quaker City Rubber Company; vice-president, Harry H. Ashton, Ashton Valve Company; secretary, Homer Whelpley, Strong, Carlisle & Hammond Company; treasurer, W. T. Johnson, Under-Feed Stoker Company of America; executive committee, Lane Thompson, Lunkenheimer Company; M. B. Skinner, James McCrea & Co.; L. B. Mann, Fairbanks Company; H. D. Raymond, A. W. Chesterton Company; H. A. Pastre, Liberty Manufacturing Company.

At the regular meeting of the National Exhibitors' Association on Thursday evening, September 15, the retiring president, William G. Le Compte, was presented with a very handsome gold watch and fob with the various Masonic emblems, Mr. Le Compte being a thirty-second-degree Mason. The presentation speech was a most happy one, delivered by Arthur L. Rice, editor of the *Practical Engineer.*

One of the other pleasant features was the presentation on Friday of a beautiful gold-headed cane to "Dick" Foley, of the Home Rubber Company, by a number of admiring friends among the exhibitors. The presentation took place at the Home Company's booth, with a neat speech by J. Winchester Holman, of the *Mining World.*

Only Partly Watt

On a visit to Birmingham, we were informed that there was running at the works of Gillott & Sons, the well known steel-pen manufacturers near that city, an old steam engine supplied by Watt & Boulton. Application for photographs and particulars brings out the information that all that is left of the original Watt engine is the spear connecting rod and the flywheel. The cylinder, slide valve and parallel motion are of modern construction, the old Watt piston and valve having been replaced in 1886 by Buckley's piston and cylindrical slide valve, with relief valves of that date. The engine in its present form is working very satisfactorily with a boiler pressure of only 26 pounds and developing 60 horsepower.

In 1902, Russia produced 2,000,000 tons of peat fuel. This quantity is more than any other nation produced in that year.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain; Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for August, 1910, *was* 170,000.

September 6	35,000
September 13	33,000
September 20	33,000
September 27	33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

The Rochester Convention

The twenty-eighth annual convention of the National Association of Stationary Engineers at Rochester, September 13-16, has gone into history as the largest and most successful ever held by the organization. There were in attendance three hundred and sixty delegates from forty-one States and the District of Columbia. There were present the representatives of one hundred and thirty manufacturers who occupied one hundred and seventy-five booths in the exhibit hall. Beside these there were also friends and guests who brought the total number attending the convention considerably above the one-thousand mark.

The approximate cost of this convention estimated from the reports of the treasurer and mileage committee was $13,000, which, of course, does not include the individual expenses of the delegates nor the sums appropriated by the subordinate associations for convention purposes.

What it costs the manufacturers to display their products in the booths, to distribute souvenirs and the customary, as well as the unusual, hospitality that was occasionally extended may not be so easily guessed, nor is it of much importance, as it is generally understood that the exhibitor gets full value in the advertising and publicity attendant upon such occasions.

But does it pay the association to hold annual conventions? As the initiative and referendum method of legislation which was adopted some years ago has reduced the necessary work of the convention to the accepting or rejecting the reports of its officers and committees, to fixing the amounts that shall be paid to delegates and officers, and the election of officers, the question does it pay often comes to the front.

In the preamble to the constitution it is stated that preference shall at all times be given to the education of engineers. While the social events which occupy a large portion of the delegate's time, and the examination of the mechanical exhibits coupled with the duties attending the convention sessions are in a broad sense educational, there are many who favor either biennial conventions or such changes in the constitution as will permit of representation by a much smaller number of delegates elected from State associations.

Either of these changes, it is argued, would leave in the treasury at least $10,000 which could be profitably expended in helping the weaker associations, in employing lecturers and in other ways developing and improving the educational features of the organization and for which it was founded.

In the convention the question of paramount importance is the selection of a vice-president for the ensuing year. To be sure this is framed about by excursions to points of interest, by attendance at entertainments, dinners, social gatherings and the selection of a city in which to hold the next convention, but the *piece de resistance* is the election of this officer.

Is it worth the price?

License Laws

We put a gate at the head of a flight of stairs so that the baby cannot fall down and cause anguish and dismay all through the family. We did not do this because we suspected that the baby had any deliberate intentions of barking himself up. We did it to protect him from his ignorance.

The child is father to the man—and frequently brother too. That is to say, like Peter Pan, some of us never grow up. We need "gates" in the form of life lines at the seashore; bells, barriers and flagmen at the railroad crossing; automobile speed laws, and don't-blow-out-the-gas signs as well as a thousand and one warnings not to do this or to be careful about that, all to protect us from ourselves.

The enactment and enforcement of license and inspection laws constitute a "gate" to protect the public and the mentally infantile power-plant owners and operators from their own ignorance. With thorough license laws in force, none but those with the proper mental equipment would be allowed to hold positions as engineers. We have specified only proper mental equipment, for it is perfectly safe to leave the matter of proper physical equipment entirely to the man who does the hiring.

It is sure that he will not take on any one-armed firemen or blind engineers, any more than a person who had a pig in an alley to catch would get a man with parenthetical ( ) legs to do the trick. Would that the employer could be as thoroughly depended upon to hire men properly qualified mentally.

When every State has a good license

and inspection law or when there are thorough Federal laws—and we believe that either the former or the latter are surely coming—we will have to cast about for a new stock topic for editorial comment. In a way, we shall regret this, for the present one has been with us long and ever active.

On page 1739 in this issue a correspondent expresses a line of logic which is naive. We are not going to answer a single sentence for we feel that anything which we might say would be quite superfluous. He makes us think of the two men who had been watching a free street parade which was given by a circus. One of the men was a pessimist. When the other man asked him what he thought of the parade, he replied that it was no good and ought to be prohibited. He claimed that his wife and children would see it and then want to go to see the circus, and there is where they would get him. In fact, it was nothing but a scheme to get money.

So with the correspondent just referred to, he thinks that license laws are nothing but a scheme of some men to get the jobs of others.

Economic Engineering

During the past summer, New Yorkers have witnessed the spectacle of a sixteen-story office building, which had been erected about twelve years, being torn down to make room for a still taller building. Apparently the trend of modern practice is to discard anything which can be replaced by a more efficient substitute, insuring an increased return upon the investment.

This practice is not by any means confined to the building trades; in fact, its greatest field of application is to be found in the generation of power. Here, engines, which only a few years ago were considered to have approached very closely the goal of perfection, are now being replaced by new prime movers which will meet the increasing demands for power with a higher efficiency. Boilers, mechanical stokers, auxiliaries, oiling systems, etc., are subjected to the same treatment. But not alone is the equipment affected, the methods of operation are also undergoing a transformation; the tendency, especially among the larger plants, being to combine scientific results with practical operation, to systematize methods, and to effect a suitable system of keeping records and costs, so that the performance of the plant may be known at any time.

With the exigency for such methods a new field in engineering has been created, employing what may be termed the "economic engineer." The duties of this class of experts are to carefully investigate the conditions existing at the plant under consideration, view the proposition from all sides, devise schemes whereby the equipment and methods of operation can be improved and made more efficient, and submit a report to the owners, setting forth such recommendations as will accomplish the desired result. The economic engineer is in a much better position to produce an effective solution of the problem than the superintendent or the chief engineer, because with the latter it is often very hard to break away from long established customs, while the economic engineer is free from local prejudice and has a vast amount of data, collected from various other plants, to aid him in making intelligent recommendations.

Another Chance for Investors

The latest institution to philanthropically offer a magnificent chance for riches to the widows and orphans and hard-working mechanics, is the National Mechanical Power Company, 1548 East Sixty-third street, Chicago, Ill.

The perpetual-motion scheme which this company is propagating is the old, old one of falling weights on the periphery of a wheel, the weights being returned to their starting place at a point on the wheel less distant from the center. Exactly the same number of weights are ascending and descending at all times, but owing, of course, to the "new principle" which has been discovered by the inventor, the wheel is enabled to give out an excess of power over that required to raise the weights, limited only by its diameter and the number of weight systems employed.

The operation is explained as follows: "Each wheel has two rows of pockets, the inner row, called the short side and the outer row, the long side. The wheel is made to revolve by falling balls. When a ball has passed up the short side to the top, it rolls out into a triangular conduit, which is fast on the side of the wheel, and rolls down into the outer row of pockets. After it has reached a point near the bottom of the wheel it passes through another conduit back into the inner row of pockets and is elevated to the starting point, thereby making the *gravity* greater than the *levity*."

It is needless to say that a working model of this device has not yet been produced, although various mysterious patterns and parts lay about the office of the concern, upon the walls of which are plastered numerous blueprints showing the "simple principle" on which the thing works.

The excuse for not exhibiting a working model is said to be totally in the interests of those who are about to invest in the stock of the concern, as after the model has been exhibited the stock will, of course, jump three or four hundred per cent. Therefore, it behooves all the innocent, nontechnical parties who have not yet been "soaked," to get in on the ground floor immediately.

Why Use Compression?

The subject of compression is not new. It has been discussed for years by engineers, engine builders and college professors, but today the question is very little nearer being settled than it ever was.

Many thinking men have taken a pad of paper and a pencil, and after plotting diagrams, some of which could not be obtained in actual practice with the steam and exhaust valves properly set, have undertaken to prove that compression is advantageous.

Other thinking men, men with engines before them, have not wasted their time with pad and pencil, but have experimented with their engines and found that with valves set to give a late steam admission and practically no compression the steam plant could be operated on less coal than when the engines were operated under high compression.

Men are a great deal like sheep—they are prone to follow a leader. Someone has decided that compression is necessary for smooth running and for the highest efficiency, and the idea has been generally accepted by engineers. They have taken the teachings of others as facts without questioning the accuracy of the statements, although at the same time they have had the means of proving or disproving, to their own satisfaction at least, that compression is, or is not, economical.

Engineers who have taken interest enough in their work to make tests, although demonstrating to their own satisfaction that compression is expensive to maintain, and have proved it by the difference in the fuel bill of their own plants, have received more or less criticism from others holding a different view.

But the strange part of it all is, that, while column after column has been published in the technical journals, criticizing the practice of not using compression, not one of the critics, so far as we know, has verified his contention by proving from actual experiments on a real engine that his theory is correct. And so the question is juggled from time to time and nothing definitely accomplished.

A few experiments in valve setting would do much to prove or disprove the contentions of those holding different views on the question, providing the tests were impartially and carefully made.

The engineer, or the man who pays the bills, does not care much for the paper and pencil end of the argument; but they do care about the money that can be saved in operating a steam plant, and would much rather see figures obtained from actual operation. It would seem that the proper place to carry out these experiments is in the laboratory of the technical institution. When this is done, some interesting and reliable data will be obtained.

Inquiries of General Interest

Condensing Water Required

I have a compound noncondensing engine which I wish to run condensing. The boilers evaporate about 1000 cubic feet of water per day. How much water will be needed to condense the steam?

G. E. B.

The quantity of injection water will vary with the conditions; such as the temperature of the exhaust, the temperature of the water, the degree of vacuum and whether a jet or a surface condenser is used. Under ordinary circumstances the amount of injection water required will probably be 30 times the weight of the steam exhausted, which in this case will be 30,000 cubic feet, or about 19,000,000 pounds.

Independent Condenser

What is an independent condenser and in what manner does it differ from any other?

J. R.

When the term independent is used it refers to something that acts independently of other machinery. As applied to condensers it is a misnomer as it is the air pump to which reference is made, and while it may be operated by itself independently of the engine, it is called independent to distinguish it from one operated by the engine it serves.

Capacity of Coal Bunker

A coal bunker is 137 feet long, 12 feet wide and 15 feet deep. How much soft coal will it hold level full?

M. B. C.

There are in the bunker

$$15 \times 12 \times 137 = 24{,}660$$

cubic feet. The weight of loose soft coal varies from 45 to 55 pounds per cubic foot. Taking 50 pounds as the average weight, 40 and 44.8 respectively will be the space occupied by a short and a long ton, and the bunker will hold

$24{,}660 \div 40 = 614$ short tons.

$24{,}660 \div 44.8 = 550.5$ long tons.

Crank-pin and Piston Travel

Which travels faster, the crank pin or piston of an engine?

A. M. D.

The crank pin moves over the semicircumference of a circle, while the piston moves through the diameter of the same circle, hence the pin travels faster.

Cost of Exhaust Steam Heat

Does it cost more to heat with exhaust steam than with live steam?

E. A. D.

Not necessarily, nor usually. An exhaust steam-heating system may be so designed that there will be little or no back pressure in the engine cylinder and all the heat got from it will be practically clear gain.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Steam Pressure and Hight of Water Column

If the steam pressure in a boiler is 80 pounds, how high will it force water in a one-inch pipe attached to the bottom of the boiler?

B. L. S.

If the water in the pipe is of the same temperature as that in the boiler it will be forced to a hight of 200 feet.

Horsepower of Boilers

What is the proper rule for computing the horsepower of a boiler?

J. A. S.

The "Centennial Rule," which requires that a boiler shall evaporate 30 pounds of water into dry steam from feed water at 100 degrees under a gage pressure of 70 pounds per square inch for each horsepower. It is equivalent to the evaporation of 34½ pounds of water from and at 212 degrees. There should be 10 square feet of heating surface in a boiler and ¼ to ⅓ square foot of grate area in the furnace for each horsepower.

Key Seat in a Split Pulley

Where is the right place to put the key in the hub of a split pulley?

F. R. T.

Keys are not necessary in split pulleys in which the bore of the hub fits the shaft properly. If a key seat is desired, it is usually cut at the middle of one of the halves of the hub and a feather is used instead of a key.

Engine Horsepower

What is the indicated horsepower of a 10x18-inch engine running 150 revolutions per minute; 100 pounds initial pressure per square inch in the cylinder cutting off at quarter stroke and exhausting into a heating system against 10 pounds back pressure?

C. W. D.

The mean forward pressure of the steam in the cylinder is practically 0.6 of the initial pressure or 69 pounds absolute per square inch. The back pressure is

$$10 + 15 = 25$$

pounds absolute, which, deducted from the 69 pounds absolute forward pressure, leaves

$$69 - 25 = 44$$

pounds mean effective pressure per square inch in the cylinder. The piston speed is 450 feet per minute and the area of the piston is 78.54 square inches. The horsepower of an engine is expressed by the formula

$$\frac{P.A.S.}{33{,}000} = \textit{Horsepower}$$

where

$P =$ Mean effective pressure:

$A =$ Area of piston;

$S =$ Feet of piston travel per minute.

Substituting

$$\frac{44 \times 78.54 \times 440}{33{,}000} = 45.72$$

indicated horsepower.

Safety Valve Rule

What is a safe rule for finding the pressure at which the safety valve should blow on any boiler?

J. G. R.

Calculate the bursting pressure of the boiler and set the valve to open at one-sixth of that pressure.

Meaning of the Term "Battery"

What is meant by the word "battery" when applied to boilers?

F. J. H.

It means that two or more boilers have a setting common to all and discharge steam into the same main steam pipe.

Evaporation per Pound of Coal

If a boiler evaporated 10.25 pounds of water per pound of coal containing 13,900 B.t.u., how much would have been evaporated per pound of coal containing 14,200 B.t.u. per pound?

J. C. D.

As it was the heat that evaporated the water a coal having more heat units would evaporate water in proportion to the excess in heat units. Hence

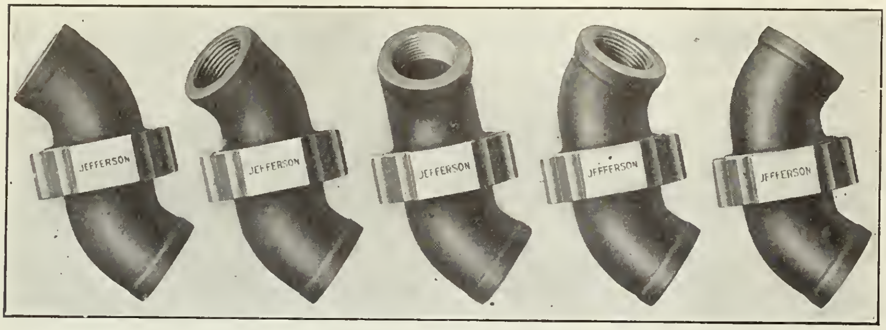
$$\frac{14{,}200}{13{,}900} \times 10.25 = 10.465$$

pounds of water would under the same conditions have been evaporated for each pound of coal containing 14,200 B.t.u.

New Power House Equipment

Jefferson Swing Union

The new swing union, made by the Jefferson Union Company, Lexington, Mass., consists of two 45-degree elbow parts with a spherical ground iron-to-iron seat. The two ends are connected by a nut of either brass or malleable iron. These unions can be made with the regular Jefferson brass-to-iron seat, which consists of a narrow brass wrought-metal ring sunk in a channel in which it fits tightly. A lip of iron protects the brass ring from contact with the fluid, and also from injury, should the pipe be screwed in too far. For close work the Jefferson swing union can be equipped with a special notched nut.



Various Positions of Jefferson Swing Union

The accompanying illustration shows the union in various positions, the joint always being tight regardless of the angle of connection.

By using an iron-to-iron seat these unions can withstand the high temperature of the exhaust from gas or gasolene engines.

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

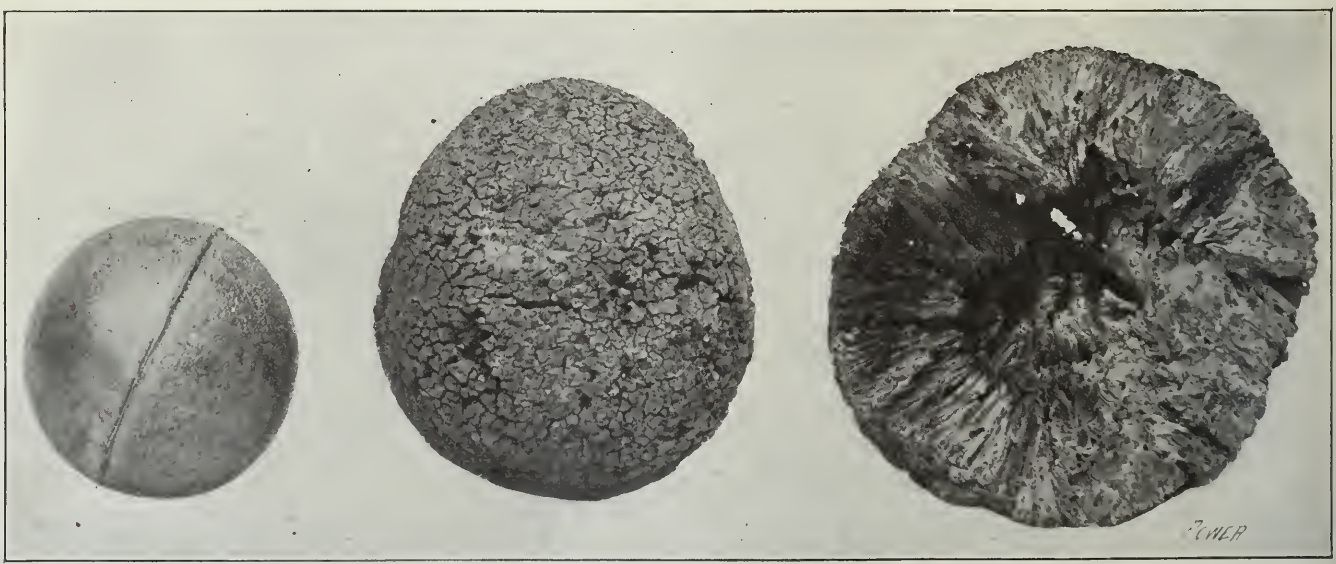
Metallic Boiler Cleaner

In our issue of August 3, 1909, brief mention was made of a new boiler cleaner placed upon the market by the Connecticut Metal Boiler Cleaner Company, of New Haven, Conn. Since then we have come into possession of more definite information upon the subject.

The device consists of a number of metallic balls contained in a cast-iron basket which is suspended just below the water line in the boiler. The composition of the balls is zinc, antimony, aluminum, bismuth, copper, lead, tin and mercury. Each ball does not contain all these ingredients, nor are the proportions the same. With zinc (which is frequently used in marine work to prevent scale) as a basis, the balls are divided into eight groups, each containing zinc alloyed with various combinations of the other metals. It is claimed that the other metals hold the zinc in solution for different periods under various conditions of pressure and temperature; hence, a ball of the first group may be disintegrated in, say, three weeks, one of the next group a week later, and so on, the action of all the balls in the basket extending over a period of several months. The quantity of the metal compound and the proportion of the ingredients entering into its composition depend upon the horsepower of the boiler and the quality of the water. Only in extreme cases is the group containing mercury used.

It is claimed that the compound will prevent the formation of new scale and will loosen old scale by setting up a galvanic action which causes bubbles to form between the boiler plate and the scale, thus loosening the latter.

The accompanying photograph shows one of the metal balls, first in its original state before being placed in the boiler; next, its condition after having been in the boiler for about three weeks; and finally its appearance after having been in the boiler for five weeks.



Before and After Being Placed in Boiler

A personal canvass among the users of these metallic balls has met with a number of very favorable testimonials, and a few that were unfavorable.

The Hopper Steam Turbine

The object of this invention is the production of a steam-turbine engine in which effective and simple means are utilized for obtaining an efficient operation for a wide range of loads. Means are also utilized for automatically varying, in accordance with the load demand, the effective number of operating steam supply nozzles.

The steam passages are so arranged that the steam may traverse the entire system at an average velocity without being restricted at any point. The area of the nozzles gradually increase in width, and the movable and fixed vanes in depth. Steam in passing through them is prevented from acquiring its full theoretical velocity from the perpendicular pressure used by its action on the vanes.

The turbine shown in the accompanying illustrations is of the multistage type in which steam is admitted at the middle and flows in opposite directions to the exhaust openings.

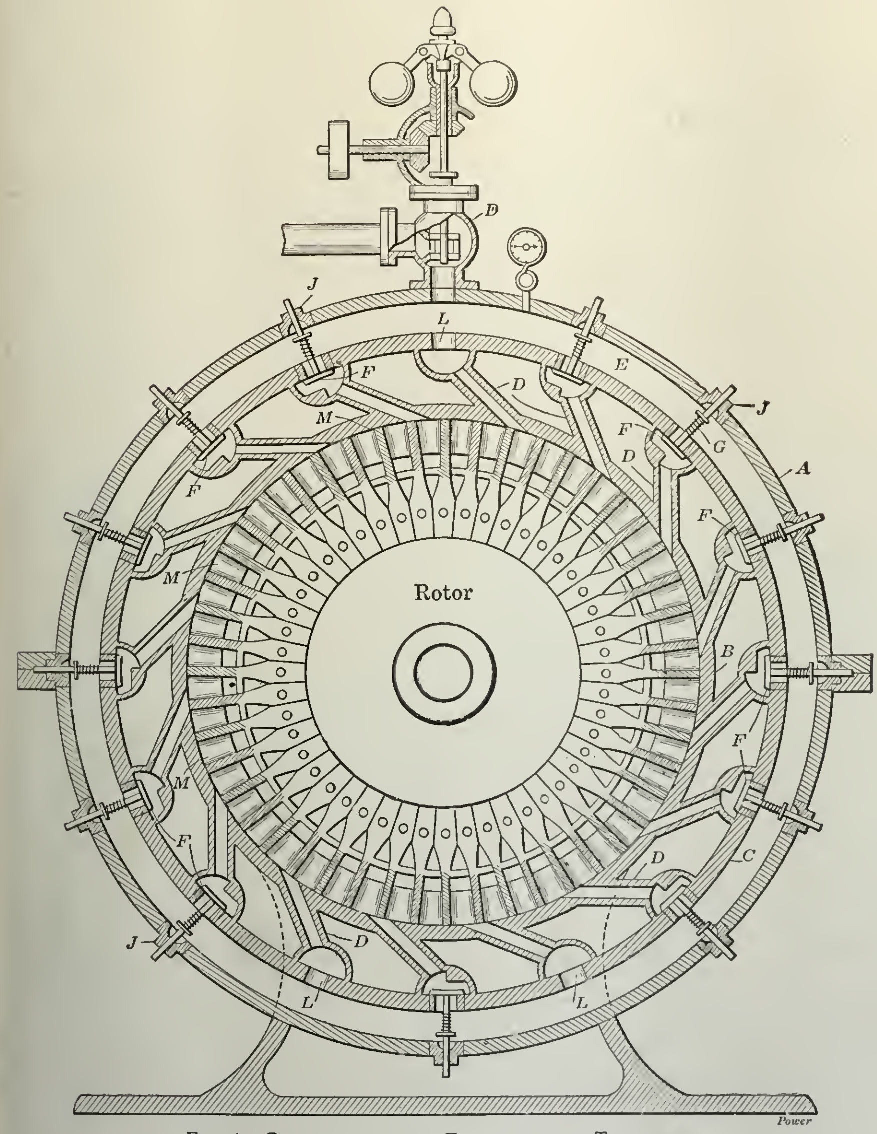


Fig. 1. Cross-sectional Elevation of Turbine

Fig. 1 shows a sectional elevation of the turbine. Fig. 2 is a longitudinal section and Fig. 3 shows an edge and face sectional detail view of the vanes. Fig. 4 presents a detail of the vanes showing the enlargement of the steam areas between them, also a top view showing the curved form of the vanes. Fig. 5 shows an enlarged sectional view of one of the nozzle-controlling valves, also a face view of a valve seat.

The turbine comprises a series of rotors rigidly mounted on a shaft which extends through the heads of an exterior casing, and is journalled in suitable bearings formed integrally with the heads. An annular steam chest *A* surrounds the casing *B* and is divided by an annular ring *C*, from which extends divergingly downward a series of steam passages or nozzles *D* that communicate with the inner chamber of the casing *B*. Between *A* and *C* is an annular space *E* that is supplied with steam by a suitable governor through an opening on the top of of the casing. This governor controls the operation of the valves *F*, which are adapted to open or close independently by a varying steam pressure, and are so arranged that steam in the chamber *E* opens the valves *F* against the action of a spring *G*. The valve rests against a seat threaded into the ring *C*. On the stem of the valve is a lock nut *E*, Fig. 5, by which proper tension is given to the spring. The plugs *J* are of larger diameter than the valve seats. They are threaded into the steam chest and are provided with a stuffing box *K*. The three nozzle openings having no controlling valves, permit an unobstructed flow of steam from the chamber *E* through the respective nozzles with which they are connected. Under a very light load these nozzles would supply all necessary steam to the turbine. and the throttling governor would regulate its speed by the admission of steam to the chamber *E*.

As the load on the turbine increases, the steam pressure in the chamber becomes greater, and the nozzle-controlling valve having the least spring tension will open, admitting more steam to the tur-

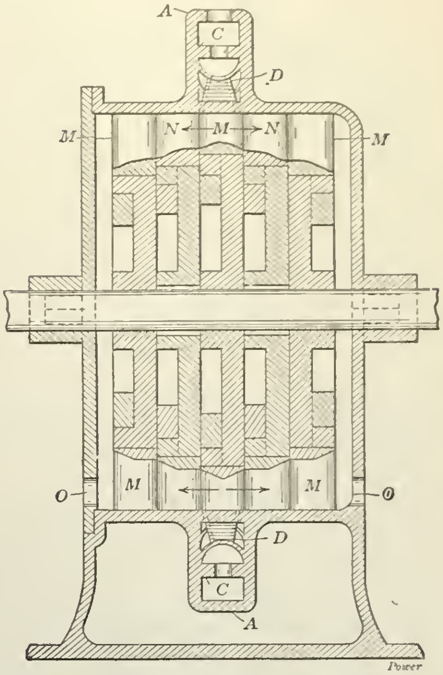


Fig. 2. Longitudinal Sectional View of Turbine

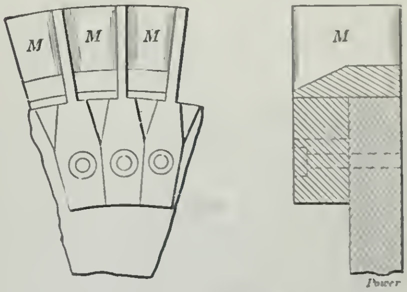


Fig. 3. Detail View of Blades

bine; a further increase of the load opens another valve in a similar manner and so on until all the nozzles are supplying steam and the turbine attains its maximum power. Should the amount of load decrease, a proportionate number of the valves would automatically close, permitting only a sufficient number of active steam-supply passages to economically operate the engine.

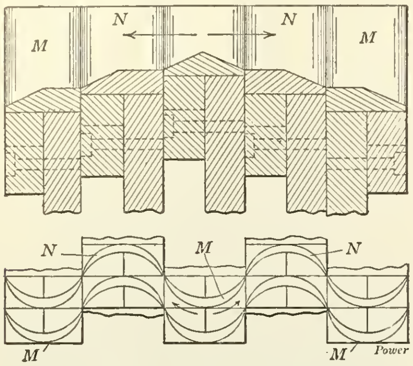


Fig. 4. Vane Design

Steam is expanded in the nozzles *D*, its pressure energy thereby being partly transformed into velocity energy, and is made use of by passing it in either direction through the vanes *M* of the rotors, as indicated by the arrows in Figs. 2 and 4 Between each pair of rotors is inserted a set of stationary guides or vanes *N*, Fig. 4, for the purpose of receiving and reversing the steam, discharged from a rotor, and directing it at the proper angle into a succeeding rotor; the vanes *M* of the rotors and *N* of the stators are made with reverse curves, as shown in Fig. 4. As the passages for the steam in both directions are the same, a description of its flow in one direction will answer for both.

The steam is not expanded down to exhaust pressure in the nozzles, so the lower surface of the space between the vanes of the central rotor, which first receives its impulsive force, diverge from the center outward, increasing the port in depth and area, which proportionately increases the steam expansion and velocity. It then passes to the oppositely curved blades of an abutting stator, entering openings between them of the same enlarged area, and travels one-half the distance through them on the same plane, when the blades increase in depth as described for the rotor vanes, further increasing the steam expansion and velocity, and also directing it in the second rotor at a proper angle.

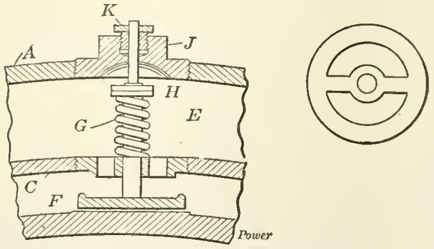
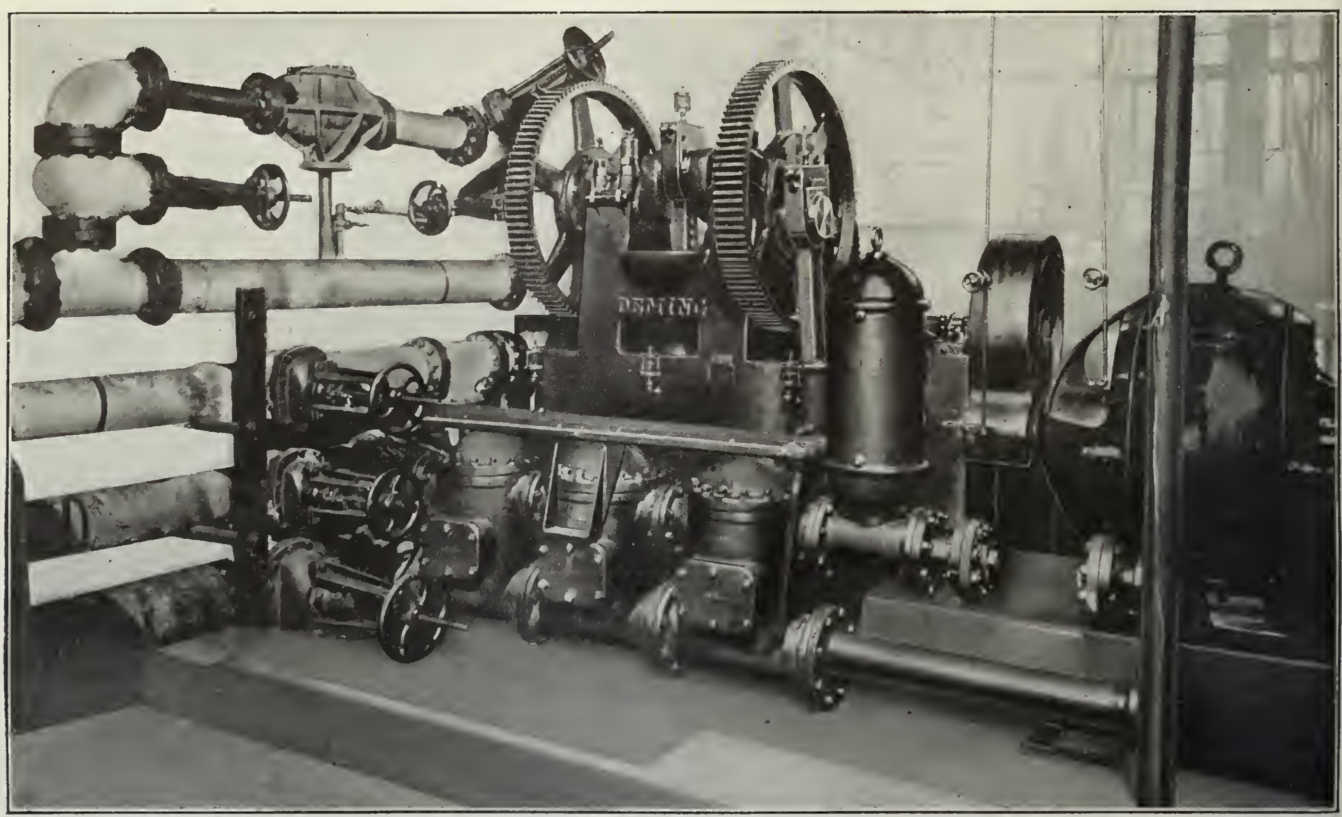


Fig. 5. Enlarged View of the Nozzle-controlling Valve

A sufficient number of rotor and stator stages are employed to utilize all the available impulsive and reacting energy in the steam. Each set transforms into velocity an equal portion of the total pressure energy, and passes the balance in the form of pressure along to succeeding sets to the end, when the steam is allowed to escape into an exhaust chamber *O*, Fig. 2, from which it may be piped to any convenient point. The exterior ends of the rotor blades are finished to a running fit with the interior of the casing, and likewise the rotor sides adjacent to the stators. The rotor and stator vanes may be dovetailed in a projecting flange and otherwise secured by bolts as shown, or any suitable method may be employed for securing them.

A Notable Feed Pump Installation

Two triplex motor-driven feed pumps, built by the Deming company, Salem, O., have recently been installed in the plant of the Eastman Kodak Company, Rochester, N. Y. These pumps supply 5000 horsepower of boilers, the pressure on the delivery side of the pumps being 140 pounds per square inch. Each takes its supply from the feed-water heater under a head of from 3 to 6 feet and handles the water at a temperature of 210 degrees. The larger pump, shown in the accompanying illustration, has 9x10-inch cylinders and runs on an average at 25 revolutions per minute. It is fitted with



Deming Motor-driven Triplex Feed Pump

a Mason combination pressure-regulating valve connected between the discharge and suction, and the motor is of the variable-speed type. The smaller pump runs at 30 revolutions per minute and has 7x8-inch cylinders. These pumps have all the characteristics of the Deming construction, including bored crosshead guides, machine-cut gearing and bronze boxes with wedge and screw adjustment at the crosshead ends, and marine type babbitted boxes at the crank end. The valve chambers are separate castings bolted to the cylinders, making all valves readily accessible for repairs. This style of outfit, mounted on a cast-iron sub-base, makes a desirable combination where economy of space is essential and electric power is available.

Evening Technical Courses at Pratt Institute

The Pratt Institute, at Brooklyn, N. Y., is conducting a series of evening technical courses, adapted to the requirements of men who are engaged during the day. The classes meet on Monday, Wednesday and Friday, from 7:30 to 9:30, and from September 28 to March 24. The courses comprise practical mathematics, technical chemistry, industrial electricity, elementary electricity and practical mechanics, electrical machinery, electrical design, mechanical drawing, machine design, mechanism, steam and steam engines and strength of materials. For the course upon steam and steam engines, examinations in arithmetic only are required. The course is intended primarily for young men who are anxious to fit themselves to become competent stationary engineers, or who are preparing to pass examinations to obtain an engineer's license. It covers the study of steam and the steam engine and steam boilers, fuel and steam generation, properties of steam, condensing and noncondensing engines, advantage of single, compound, triple expansion, etc. Laboratory practice includes valve setting, indicator work and the making of engine tests for power, steam consumption and efficiency, pump tests, gas-engine styles, etc. The tuition fee is only $15 for the season of six months. The Pratt Institute has an excellent equipment both of apparatus and instructors, and by taking a number of these courses successively a man who is obliged to earn his living in the daytime can acquire a knowledge not only of drawing and mathematics, but all the technology which is essential or useful in his vocation.

Brooklyn Trade School

The Brooklyn Evening Technical and Trade School gives a number of free courses to young men of ability who have their evenings at their disposal. A two-year course in industrial chemistry, which may be followed by a third year in quantitative analysis; electrical engineering, preceded by necessary mathematics; steam engineering, qualifying men to set up dynamos and engines, and test and regulate them, also elementary civil engineering, may be taken up by the students.

I. U. S. E. Hold Convention in Denver

During the week of September 12 to 17, the International Union of Steam Engineers held its annual convention at Denver, Colo. Matt Comerford was reelected as president and J. G. Hannahan, of Chicago, was elected secretary. Further particulars will be published as soon as available.

SOCIETY NOTES

The National Association of Manufacturers, of which John Kirby, Jr., is president, has removed its general offices from 170 Broadway to 30 Church street, one of the twin Hudson Terminal buildings in New York City. The association's growth during the past few years necessitated more commodious quarters than could be obtained at the Broadway address. The editorial and advertising departments of *American Industries,* the association's official magazine, and of *Export American Industries* (English and Spanish editions), have also removed to the new location.

At a meeting of the executive committee of the National Gas and Gasoline Engine Trades Association, held in Racine, Wis., August 31, arrangements were completed for the next convention of the association to be held in that city, December 12 to 15 inclusive. The local committee at Racine has arranged for suitable entertainment and a very cordial invitation is extended to anyone who is interested in any form of internal-combustion engines to attend the convention. Special arrangements will also be made for the entertainment of ladies who accompany members or visitors.

PERSONAL

John B. Milliken, controller of the Crocker-Wheeler Company, of Ampere, N. J., has accepted the position of treasurer of the Yale & Towne Manufacturing Company, of Stamford, Conn. Mr. Milliken's headquarters will be in New York.

Arthur Fritsch was recently appointed by the Pawling & Harnischfeger Company, of Milwaukee, as manager of the Chicago office, Monadnock block, succeeding W. E. Kreamer, resigned. Mr. Fritsch was formerly connected with the engineering and sales department of the Allis-Chalmers Company.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago. Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

BUSINESS ITEMS

The Bigelow Company, New Haven, Conn., has just sold to the Pacific Mills, Lawrence, Mass., for use in its worsted department, twenty 175-horsepower Manning boilers. This order makes a total of over 11,500 horsepower of fire-tube boilers sold to this concern by the Bigelow company during this year.

The Pawling & Harnischfeger Company, of Milwaukee, designer and builder of traveling electric cranes and hoists, announces the appointment of Arthur Fritsch as manager of its Chicago office in the Monadnock block, succeeding W E. Kreamer, resigned. Mr. Fritsch was formerly connected with the engineering and sales department of the Allis-Chalmers Company.

The H. W. Johns-Manville Company has found it necessary, owing to the great increase of business in the vicinity of Atlanta, Ga., and Rochester, N. Y., to open offices in those cities. The Atlanta office is located in the Empire building, in charge of W. F. Johns, who has been traveling that territory for the company for a number of years. The Rochester office is located at 725 Chamber of Commerce, in charge of H. P. Domine, formerly with the Buffalo branch of the company.

While the Joseph Dixon Crucible Company, of Jersey City, N. J., is the oldest and largest graphite concern in the world, there are no doubt many more or less recent users of graphite who may not be fully acquainted with its line. The name Dixon has always been associated with flake graphite, and it is flake graphite alone that the Dixon company provides for lubrication. The word "flake" seems to give some the impression that the Dixon graphite is coarse. It is true that a large flake graphite known as Dixon's No. 1 is prepared, but in addition to this there are two finer grades, that known as No. 2 and that indicated by the number 635. This last, while a flake graphite, is ground to an impalpable powder. Under the microscope its flake formation is evident, but to the naked eye the flakes are not appreciable. The No. 2 graphite is about one-half way between the No. 1 and No. 635. These three grades (referring to size of flake) cover every lubricating requirement that occurs in the power house and machinery field, from heavy, slow-moving bearings to the very finest and closest set ones. The Dixon company is always glad to take up the lubricating problem with any who may desire to avail themselves of their experience.

NEW EQUIPMENT

North Bay, Ont., will install new waterworks.

Oakland, Ill., will install a waterworks system.

Harlowton, Mont., will install a waterworks system.

The city of El Centro, Cal., will erect a pumping station.

Strathroy, Ont., will install new waterworks machinery.

Baker Bros., Toledo, Ohio, are erecting a new power house.

The town of Rippey, Iowa, voted to issue waterworks bonds.

Collyer & Keyworth, Gardner, Mass., will install a steam boiler.

Arlington, Tenn., contemplates installation of waterworks system.

New Liskeard, Ont., is in the market for waterworks equipment.

The Geo. E. Belcher Company, Stoughton, Mass., will install a boiler.

The Earle Fruit Company will erect a packing house at Turlock, Cal.

Alice, Texas, contemplates issuing $35,000 waterworks and street bonds.

L. O. Koven & Bros., Jersey City, N. J., will build a new power house.

The St. Croix Gas Company has been granted a franchise at Hudson, Wis.

The Rensselaer (N. Y.) Water Company will enlarge its pumping station.

The Range Power Company, Chisholm, Minn., will install a steam plant.

Swift & Co., Jersey City, N. J., will erect a $150,000 cold-storage warehouse.

The Modern Power Company will locate a large new plant at Winnipeg, Man.

The Montreal (Que.) Hydro Electric Company will locate a plant at Montreal.

The Purity Ice Cream Company, Watertown, S. D., will erect a new plant.

Franklin, N. C., will vote on issuance of $24,000 bonds for waterworks system.

The Center (Tex.) Light and Ice Company will increase the capacity of its plant.

Orillia, Ont., will spend $30,000 for new electrical equipment for power purposes.

The Chapin-Sacks Ice Company, Washington, D. C., will erect additional buildings.

Gomery Brothers, Allentown, Penn., will erect a cold-storage and ice-making plant.

The British Columbia Electric Railway is extending its power plant at Vancouver, B. C.

John I. Beggs, of Milwaukee, Wis., will erect an electric power house at Menosha, Wis.

City of Maysville, Ky., will vote on issuance of $50,000 bonds for an electric-light plant.

Regina, Sask., will buy large quantities of electric machinery for its light and power plant.

The Washington County Hospital, Hagerston, Md., will install boiler, engine and pumps.

The C. C. Garrigas Machine Company, Bristol, Conn., will install boilers, engine and pumps.

The Henderson (N. C.) Power Company will rebuild its electric-light plant, recently burned.

The city of Harlan, Ky., is having plans prepared for the construction of an electric-light plant.

J. B. Thompson, Gilbert, Minn., will be granted a franchise to install an electric-light plant.

The Straight Electric Company, Des Moines, Iowa, is considering the erection of a plant at Dows, Iowa.

H. F. Pierce, Bay City, Mich., has been granted a franchise to erect a lighting plant at Crosby, Minn.

The U. S. Bobbin and Shuttle Company, Manchester, N. H., is in the market for an engine and boiler.

The Farmers Creamery and Produce Company will erect a creamery and boiler room at Newell, Iowa.

The Astoria (L. I.) Heat, Light and Power Company will erect new meter house, pump house and boiler house.

The power house and electric-light plant of the Ruby Gulch Mining Company, at Zortman, Mont., were burned.

The Hospital for Children and Training School for Nurses, San Francisco, Cal., will erect a new boiler room.

Thos. D. Miller, of New Orleans, La., has been granted a franchise to install a gas plant at Cedar Rapids, Iowa.

The Canadian Coal Corporation, Salmon Harbor, N. B., will need a large amount of steam and electrical equipment.

The city of Toledo, Ohio, is preparing to extend the city filtration plant. Additional pumps and engines will be installed.

W. F. Turner and J. W. Barker have been granted a franchise by the town council, Atkins, Ark., for an electric-light plant.

The city of Pittsfield, Mass., is considering the installation of a new pumping station. Electric power pumps will be installed.

The Hydraulic Power and Manufacturing Company, Niagara Falls, N. Y., contemplates making extensive additions to its power plant.

The Western Canada Power Company, New Westminster, B. C., is in the market for large quantities of electrical equipment of all kinds.

Bids will be received until September 29 by the United States engineer, Wheeling, W. Va., for two 115-horsepower boilers, two steam-driven air compressors.

The Navy Department, Bureau of Supplies and Accounts, Washington, D. C., will open bids October 4, for 30 testing generators, 10 electric heaters as per Schedule 2890 and air compressor, Schedule 2905.

NEW CATALOGS

The Babcock & Wilcox Company, 85 Liberty street, New York, Chas. C. Moore, Pacific coast manager, 99 First street, San Francisco. Cal. Catalog. Stirling boiler. Illustrated 22 pages, 8x10 inches.

Buffalo Steam Pump Company, Buffalo, N. Y. Catalog No. 228. Vacuum pumps, jet, barometric and surface condensers. Illustrated, 62 pages, 6x9 inches.

Saginaw Manufacturing Company, Saginaw, Mich. Catalog. Gilbert wood split pulleys. Illustrated, 40 pages, 6x9 inches, paper.

Wright Manufacturing Company, Detroit, Mich. Catalog No. 12. Steam traps, water columns, exhaust heads, filters, etc. Illustrated, 44 pages, 6x9 inches.

Jeffrey Manufacturing Company, Columbus, Ohio. Bulletin No. 42. Coal-handling and mine equipment. Illustrated, 48 pages, 6x9 inches. Booklet No. 28. Conveying machinery for handling stone, sand, etc. Illustrated, 28 pages, 3½x6 inches.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

SALESMEN—We have an A-1 proposition for high-grade salesmen. Address Box 247, POWER.

WANTED—Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," POWER.

WANTED—Live, hustling salesmen to call on engineers to sell a widely advertised oil filter; good commissions. Address Box 279, POWER.

Power and The Engineer

Vol. 32 NEW YORK, OCTOBER 4, 1910 No. 40

SEE the chief!

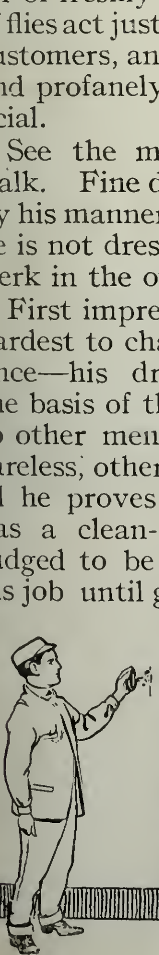
Is that the chief, he in the faded, soiled blue jeans and jumper with the crumby old peak cap on his head?

Yes, that is the chief. He has just finished eating his lunch in that corner saloon where they give the largest "scoop" in the city for five cents; where they put up a free hot lunch all day; where sawdust saturated with tobacco juice and stale beer sends up a pungent aroma which mingles with the nutritious whiffs from the "hot-dog" cauldron and the smoking platter of freshly fried crabs; where the multitudes of flies act just as though they were regular cash customers, and where the conversation is frank and profanely emphatic even if it is not beneficial.

See the manly (?) swagger of the chief's walk. Fine dignity! The chief seems to show by his manner that he is proud of the fact that he is not dressed as well as the poorest paid clerk in the office upstairs.

First impressions are the most lasting and hardest to change. A man's personal appearance—his dress and manner—is invariably the basis of the first impression which he gives to other men. If a man looks slovenly and careless, other men will accept him as such until he proves himself to be otherwise. If he has a clean-cut, snappy appearance, he is judged to be straightforward, alert and onto his job until greater intimacy serves to reverse this decision. From your own experience you all know this to be so. How often, after turning a man down, have you not remarked "I didn't like his looks"?

Of course we know that the engineer is a busy, important man, and that his work often necessitates his getting a bit greasy and mussed up. Many reason in this way, "If I clean up I'll only get dirty again; few people see me, and, anyhow, what's the use!"



If we eat, we only get hungry again. But, whether a man eats or not, as far as he is concerned, things will be all the same a thousand years from now, so what's the use! Never heard anyone reason that way, did you?

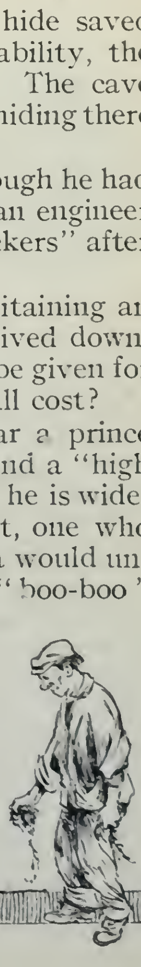
Some of the most trivial incidents often have far-reaching effects. It is said that the simple circumstance of a spider weaving its web across the opening of a cave into which a Scotch patriot had just crawled to hide saved the patriot's neck, and, in all probability, the cause for which he was fighting. The cave didn't "look" as though Bruce were hiding there and so the trackers passed it by.

Because he did not "look" as though he had some *quality* hiding within, many an engineer has been passed by when the "trackers" after brains came his way.

Why handicap yourself by manitaining an unprepossessing appearance to be lived down, when a favorable impression may be given for very little effort and at a very small cost?

An engineer doesn't have to wear a prince albert, mauve spats and gloves and a "high lid" to convey the impression that he is wide-a-wake and onto-his-job. In fact, one who showed up to work in such regalia would undoubtedly be transferred to the "boo-boo" house in short order. We will not give a list of "what to wear" or instructions on "how to trim the beard." We don't think it necessary, anyhow. There is lots more that could be said along this same line but don't wait—get busy and,

LOOK YOUR PART.



Plant of the Bigelow Carpet Company

By B. S. Walters

A new central station plant furnishing power for the two mills of the Bigelow Carpet Company which are located at some distance apart, and each of which was formerly operated by its individual power plant at a considerable sacrifice in economy.

The Bigelow Carpet Company, of Clinton, Mass., consists of two separate mills; one located on the Worcester & Nashua division of the Boston & Maine railroad and known as the "spinning mill," where wool scouring, spinning and the preparation of the yarn is accomplished; the other mill, known as the "Axminster weaving mill," where the dyeing and hacking of the yarns and the weaving of rugs and carpets is carried on, is an eighth of a mile distant from the spinning mill. Until recently each mill contained its individual power plant, and all current for lighting was furnished by the weaving mill. The company, after careful investigation by their engineer, Charles T. Main, of Boston, decided that considerable economy in operation could be secured by concentrating the power-generating units into one plant; consequently, plans were immediately drawn and a new power plant erected. A further consideration concerning the erection of a central power plant was the anticipated extension of the weaving mill.

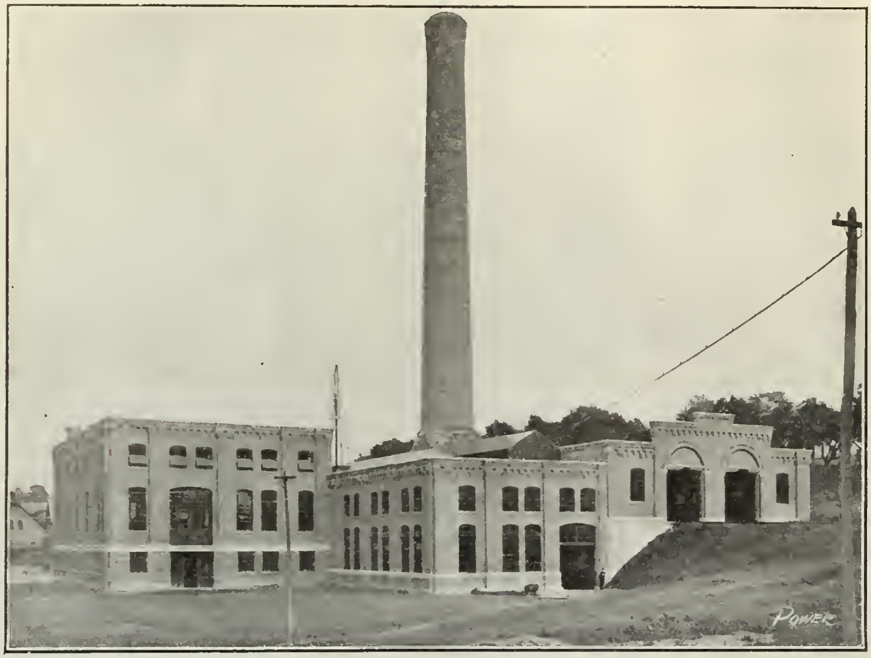


Fig. 1. Exterior View of Plant

The new plant, an exterior view of which is shown in Fig. 1, is located in a lot about halfway between the two mills, where advantage was taken of a slope of the ground for the arrangement of a coal pocket. A spur track from the railroad was laid on the top of the embankment, and a trestle was extended from this into the power house. The tracks enter the coal pocket 18 feet above the level of the boiler-room floor, and the coal is dumped from the cars to the bunker floor, where it is loaded by hand into industrial cars and run through doors in the 20-inch fire wall which separates the coal pocket from the boiler room. Each opening is equipped with a vertical rolling steel door and in front of each opening is a pair of two-ton scales with the registering device inclosed in a dust-proof case.

The coal pocket is built of reinforced concrete with structural-steel columns, brick walls and steel trusses to support the roof. It is 71x197 feet, and has a maximum capacity of about 5000 tons. It is divided in the middle by a reinforced-concrete fire wall extending to the level of the tracks.

The chimney, which is located in the coal pocket on the side next to the boiler room, is built of red brick, has a 10-foot flue, and rises 200 feet above the boiler-room floor. The boiler room, which adjoins the coal pocket, has a floor area 50x197 feet and a clear hight of 23 feet 6 inches to the bottom of the trusses. The boilers face the coal pocket, there being a firing space 20 feet deep between the boiler front and the dividing wall. All the firing is done direct from the industrial cars, no coal being dumped upon the boiler-room floor, also all the coal is accurately weighed upon being brought from the coal pockets.

The boiler equipment consists of eight old boilers, which were taken from the Axminster mill plant, and six new boilers with space available for the future installation of two additional boilers. The boilers are of the horizontal, return-tubular type, 84 inches in diameter, with 180 three-inch tubes 19 feet long, and all built by the D. M. Dillon Water Works, of Fitchburg, Mass. Fig. 2 is an end view of one of the boilers showing the piping at the back of the battery.

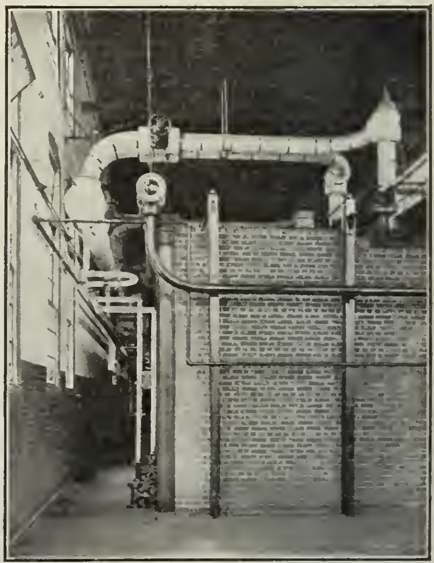


Fig. 2. Piping at the Back of Boilers

The engine-room equipment consists of a 26x48x42-inch Rice & Sargent, horizontal cross-compound condensing engine running at 120 revolutions per minute, which is direct connected to a 750-kilowatt Allis-Chalmers generator; a 20x36-inch Rice & Sargent simple noncondensing engine, running at 150 revolutions per minute and direct connected to a 300-kilowatt Allis-Chalmers generator; a 12x12-inch simple noncondensing Shepard engine, direct connected to a 50-kilowatt Allis-Chalmers generator; each of the foregoing generators being of the three-phase, 60-cycle, 600-volt type. A 12x12-inch simple noncondensing Shepard engine is direct connected to a 50-kilowatt, 120-volt direct-current generator, furnishing excitation current to the main generators. There is also a 50-kilowatt motor-generator set for excitation purposes. Space is provided in the engine room for an additional unit of 750 kilowatts capacity. All the engines are supplied with oil by a "White Star" gravity oiling system.

A general view of the engine room is shown in Fig. 3.

There are two steam pressures used at the plant: a high-pressure system of 125 pounds gage providing steam for the main engines and auxiliaries, also supplying steam to the dye house; and a low-pressure system of from 5 to 10 pounds gage, supplying steam to the feed-water heater and the heating system of one mill. There is also a vacuum system in which the vacuum is maintained at the proper point to insure the desired temperature of the overflow water going to the dye house.

The pump-room equipment consists of two fire pumps, one of 1500 and the other of 1000 gallons capacity, each made by the International Steam Pump Company; three 14x8x12-inch duplex boiler-feed pumps for regular service and one 8x5x12-inch vertical piston duplex pump for night and Sunday service, all manufactured by the Warren Steam Pump Company, and an 8x7x12-inch duplex piston pump for supplying cold water to the plant. The boiler-feed and fire pumps are equipped with Locke governors and the service pump with a Fulton tank governor. A 3000-horsepower Kelley-Berryman feed-water heater is also located in the pump room. The boilers are provided with Williams feed-water regulators, which are a necessary appliance in this plant, owing to the irregular demands for steam in the dye house. All the hot water supplied to the boilers is measured by Hersey water meters, there being one meter for each battery of boilers.

As the supply of water is limited, its economical use for condensing and manufacturing purposes was very carefully considered when designing the plant. The suction of the boiler-feed pumps is connected to the overflow line from the condenser, the water in which is heated to about 95 degrees Fahrenheit. This line also connects with the dye house and has an overflow through an elevated loop through which any excess water is discharged into the pond from which it is drawn to the condenser.

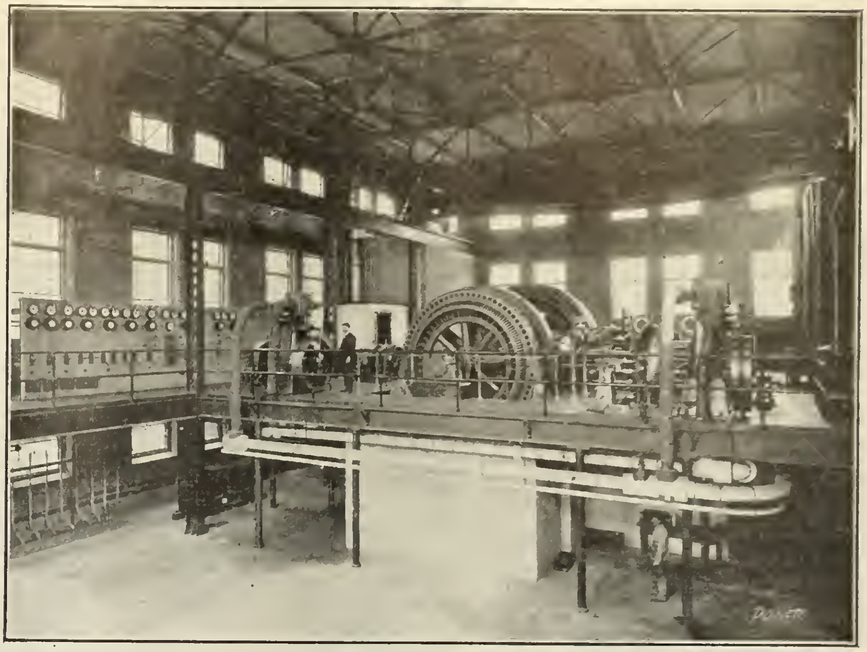


Fig. 3. Engine Room, Including Switchboard Gallery

A test which was recently conducted on two of the boilers showed the following results:

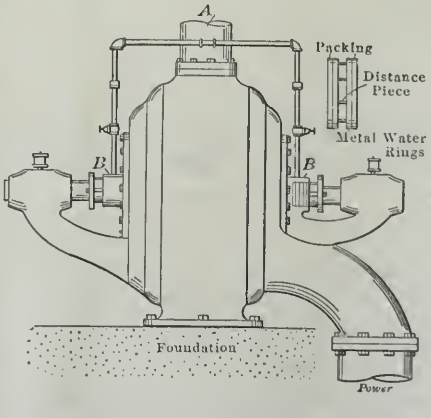
BOILER TEST ON TWO 84-INCH DILLON BOILERS.

Duration of test	10 hours, 20 minutes
Method of conducting test	Alternate
Kind of furnace	Brick, with Dillon shaking grates
Grate surface, square feet, 2 × 46.7	93.4
Water heating surface, square feet, 2 × 2,741	5,482
Ratio of heating to grate surface	58.7 : 1
Heat value of dry coal per pound, B.t.u	14,829
Moisture in coal, per cent	1.28
Ash and refuse in dry coal, per cent	6.85
Dry coal consumed per hour, pounds	2,235
Dry coal per square foot of grate surface per hour, pounds	23.9
Water evaporated per hour, corrected for quality of steam, pounds	20,470
Factor of evaporation	1 1657
Equivalent evaporation per hour, from and at 212 degrees Fahrenheit, pounds	23,860
Equivalent evaporation per hour from and at 212 degrees Fahrenheit per square foot of heating surface, pounds	4.35
Steam pressure, gage	109.1
Temperature of feed water entering boiler, degrees Fahrenheit	93.2
Temperature of escaping gases, degrees Fahrenheit	388
Moisture in steam, per cent	2.66
Boiler horsepower developed	715
Water apparently evaporated under actual conditions per pound of coal as fired, pounds	9.28
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of coal as fired, pounds	10.52
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of combustible, pounds	11.45
Efficiency of boiler and grate, based on dry coal, per cent	69.5

Continuous readings were taken on a CO_2 recording machine and showed the CO_2 to range from 6 to 15 per cent., with an average of 12 per cent. for the test.

Water Seal for Centrifugal Pump

Centrifugal pumps cause more or less trouble by drawing in air through the stuffing box around the shaft. Different methods of water sealing have been tried by engineers with varying success. The method adopted by the Chattanooga (Tenn.) Power and Light Company, which is illustrated in the accompanying drawing, has given satisfaction.



Water Sealing a Centrifugal Pump

The outlet pipe is tapped at *A* and from this point a branch pipe is run to the bearing on either side, as shown at *B*. In place of the regulation method of packing the stuffing box a combination is used, consisting of one ring of ordinary packing placed next to the inner side of the stuffing box, followed by two rings of brass each 1¾ inches thick and spaced by distant pieces 1 inch apart. On the outside of this combination of packing rings another piece of ordinary ring packing is fitted next to the stuffing-box gland. Water from the small pipe flows into the space left between the two metal rings, thus producing a water seal around the shaft.

In the case cited the engineers do not recommend that the stuffing box be screwed up so tight that no leakage will occur; on the other hand, they prefer that a slight leakage take place so that the soft packing will not deteriorate too rapidly. The idea works out well in the present instance.

The Predecessors of Watt's Engine

By H. W. Dickinson

Salient points in the history of the atmospheric engine of Thomas Newcomen, which did good work for a period of more than fifty years.

Most educated persons and probably very many engineers would, if asked who James Watt was, reply at once that he was the inventor of the steam engine, and while this is in a sense the truth, it is not the whole truth, because Watt commenced as the improver of an existing apparatus —to wit, the atmospheric engine of Thomas Newcomen.

In this article I propose to give the salient points in the history of this remarkable engine, which did yeoman service for more than 50 years, and leave the subject at the stage when Watt's attention was drawn to it and when he began his improvements.

The genesis of an invention is always interesting, but of this, the most momentous discovery of its age and the ultimate source of the material progress of the last two centuries, it must be regretfully owned that very little is known. All that can with certainty be said is that in 1710 or 1712 the Newcomen engine was in actual operation.

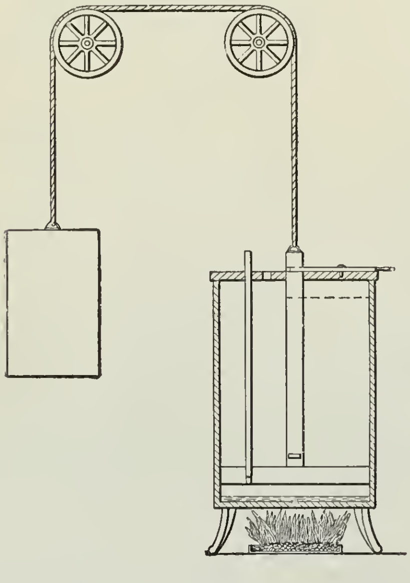


Fig. 1. Denis Papin's Engine—1690

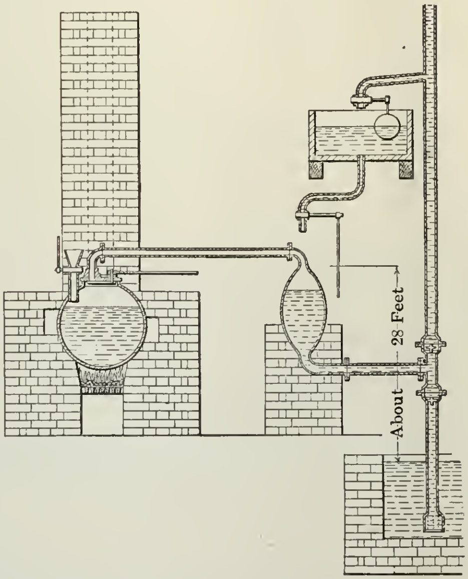


Fig. 2. Thomas Savery's Engine—1698

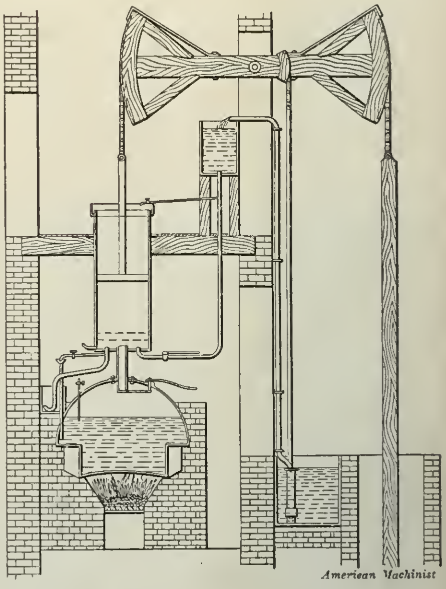


Fig. 3. Newcomen's Engine—1712

The Work of Papin and Savery

For quite 50 years, scientific and practical men, notable among them Denis Papin and Thomas Savery, had been engaged in trying to turn to practical account the discovery of Evangelista Torricelli that the atmosphere had weight, a fact that he had deduced from his observations of the hight that mercury will stand in a barometer tube. Then was given for the first time the true reason why water follows the bucket of a suction pump. It was not a great step from this to imagine that if one could remove the pressure of the atmosphere from under the bucket, the pressure of the air on the upper side would force the bucket down and, by attaching a cord to the bucket, lift a weight.

Papin got so far in 1690 as to scheme a closed vessel or cylinder with a piston, in which was a nonreturn valve, as shown in Fig. 1. A little water on the bottom of the cylinder was boiled and when the steam had forced out the air through this valve, and when the cylinder had cooled again, the piston would descend when the catch was released. But he was unable to devise means for repeating the motion even had it not been altogether too slow.

Savery went on another track. His apparatus had a boiler with a hand-controlled valve admitting steam to a receiver provided with foot and head valves. By external condensation of the steam by a stream of water outside, he could draw water from a depth of about 20 feet, and then shutting off the condensing water force it to a hight depending on the steam pressure, all as shown in Fig. 2. The necessity for this pressure was a fatal defect at this stage of the mechanic arts, for no means of construction which would stand more than a very moderate pressure were known; it had to be a plumber's job, in fact. However, Savery managed in 1698 to get a patent for 14 years on his engine; indeed, it was a master patent, so widely was it worded: that is, "for raising of water and occasioning motion to all sort of millworks by the impellent force of fire." He was granted a prolongation of his patent for 21 years, making 35 in all, so that it did not expire till 1733. The invention, for the reason given, met with but scant success. The curious in these matters will notice that this apparatus is practically identical in principle with that brought out in 1872 by Cyrus H. Hall, and known as the pulsometer.

The Newcomen Engine

And now Newcomen comes on the scene, for we are assured by a contemporary writer that: "Mr. Newcomen was as early in his invention as Mr. Savery was in his," but the latter being nearer to the court obtained a patent first. Newcomen's invention was totally different from Savery's, yet the latter's claim fully covered it, and this accounts for the fact that Newcomen never did get a patent, repeated statements to the contrary notwithstanding. The writer already quoted says that Newcomen was glad to come in as a partner of Savery. For about 12 years, therefore, Newcomen, who was an ironmonger, assisted by John Cawley or Calley, a glazier, was perfecting the invention at his home in Dartmouth, Devon.

In its essentials the plant, shown in Fig. 3, consisted of a boiler, after the style of a brewing copper, above which was a thin brass cylinder provided with a piston attached by a chain to one end of a beam or gigantic pump handle, from the other end of which hung the buckets and pump rods that balanced the piston. Steam just above atmospheric pressure was introduced into the cylinder, and when the piston had got nearly to the top of its stroke a jet of water was turned on inside the cylinder, the steam was condensed and the atmosphere forced down the piston, thus raising the pump bucket. The grand feature of this invention was that no matter from what depth water was to be raised, steam at atmospheric pressure only was required.

So far this was a combination of known elements, but success would not have resulted from it without two other beautiful and original inventions. The first was a valve gear which opened and shut the steam and injection valves at the right moments; the second was a "snifting" valve whereby the injection water and the condensed steam, together with the small amount of entrained air, were removed from the cylinder. This valve gear was the first invention in the world's history by which a nonliving agent had been schemed to repeat its own operations automatically an indefinite number of times—in its results surely a most pregnant discovery. Nevertheless, it seems to have excited the attention of the scientific men of the day very little.

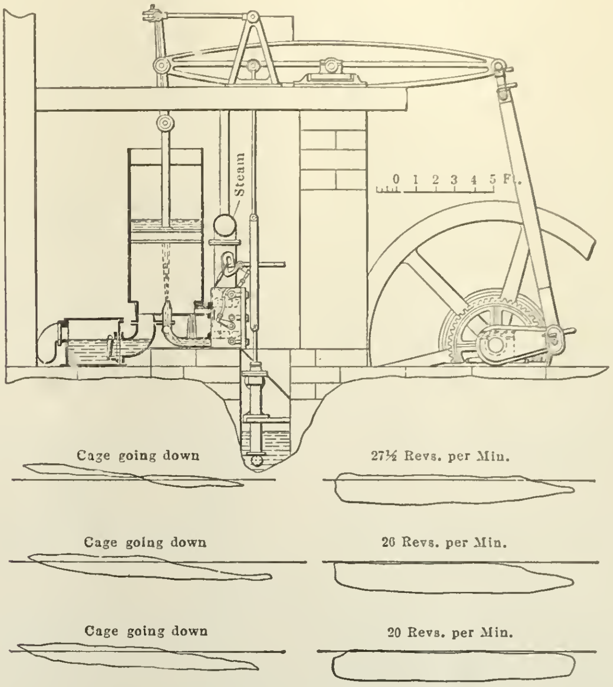


FIG. 5. ELEVATION OF NEWCOMEN ENGINE SHOWN IN FIG. 4 WITH INDICATOR CARDS TAKEN JUNE 20, 1901

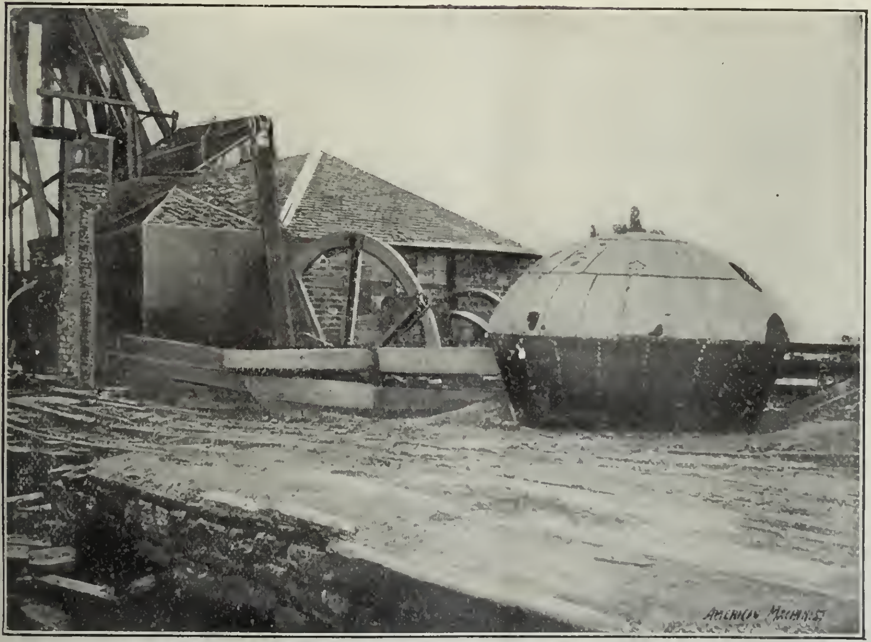


FIG. 4. A NEWCOMEN ENGINE THAT IS STILL DOING DUTY

From contemporary prints and published notices we are able to piece together with some show of probability the subsequent career of this invention. The earliest known representation of the engine is an engraving, dated 1719, of an engine erected in 1712. It is not quite certain, however, that the engine shown was the first to be erected, because proposals were made in 1710-1711 to put down an engine at Griff, in Warwickshire. Even if not at once acted upon, they were soon after, because an engine was in existence at Griff a few years later. Difficulties were encountered in the construction and working of these pioneer engines, but these were overcome by skilled mechanics from Birmingham.

THE HUMPHREY POTTER STORY A MYTH

It is said that Newcomen at first condensed like Savery, by water on the outside, and that the jet was only found out by the accident of a leaky piston. To understand this it should be remarked that water was used on the top of the piston as packing—another ingenious idea.

We are assured by Dr. J. T. Desaguliers in his "Experimental Philosophy" that the self-acting valve gear was schemed by a boy named Humphrey Potter because he wanted to be released from turning the cocks by hand so he could go and play—a most improbable tale. Several individuals of the name of Potter appear in connection with the Newcomen engine during the subsequent 15 years of its existence; indeed one went as far afield as Königsberg, in Hungary, to erect an engine, and was there believed to be the inventor of it. No doubt one of these engineers added an improvement, and hence the story.

The snifting valve was a neat way of overcoming "wind logging." From the bottom of the cylinder an eduction pipe led to a cistern of water where the end was closed by a poppet valve. When steam entered the cylinder the condensed water ran out by gravity and the air followed through this valve, giving rise to the noise which gave it its name.

SIZE AND COST OF THE OLD ENGINES

A rough idea may be formed of the space necessary at this period to develop a given power when it is noticed that the engine shown in the above mentioned engraving had a cylinder 21 inches in diameter by 7 feet stroke and from the quantity of water raised developed 5½ effective horsepower. The cost, too, was enormous. For an engine not much larger than this, built in 1725 in Scotland, the cylinder alone cost £250, and the total expense, excluding the engine house and pit work, exceeded £1000. As this must be multiplied by 3 or 4 to obtain the value at the present day, it will be seen that it was no light undertaking to erect an engine in the year of grace 1725. The engine met a long-felt want, so that, notwithstanding all this, it was adopted at collieries up and down the country, to such an extent that when the celebrated engineer John Smeaton turned his attention to the subject he found, in 1769, no less than 57 engines, one with a cylinder as large as 75 inches diameter, in use in the Newcastle district alone. The average duty of these engines was 5.6 millions of pounds of water raised 1 foot high by the expenditure of one bushel (84 pounds) of coal. By judicious proportioning and better workmanship Smeaton obtained a duty of 9½ millions. The most celebrated engine of his construction was that at Chacewater in Cornwall, completed in 1775. By that time, however, Watt, in partnership with Boulton, had begun the construction of his improved engine and as it would "fork" as much water as the atmospheric engine with about one-third of the consumption of fuel, the doom of the latter was sealed.

It did not go under without a struggle for, at the pit mouth where coal is cheap, these engines long survived. Indeed one is actually at work at the present day at a colliery near Rutherglen in Scotland. It is shown in Figs. 4 and 5, for the latter of which I am indebted to the *Proceedings* of the Institution of Mechanical Engineers. It is a rotative engine and from the illustrations it will be noticed that the wooden beam has been replaced by one of iron, and the flexible connection to the piston by a piston rod and parallel motion. As one would expect, the beam was the weakest part of the engine.

The long life of these veterans is in marked contrast to the engines of today, which find their way to the scrap heap before they have had a chance to become venerable.

Types of Ammonia Compressor

By F. E. Matthews

Of the two types of ammonia compressor, double- and single-acting, the former is usually a horizontal and the latter a vertical machine. Typical examples of each type are presented.

Ammonia compressors are divided into two principal classes; double-acting and single-acting. The former type is most commonly horizontal, although frequently of vertical construction. The single-acting type is almost exclusively a vertical machine. Each type has its own followers among builders, and under certain conditions possesses some advantages over the other. While there is much variation in details of design among the various builders, the accompanying illustrations, Figs. 1 and 2, are the most characteristic of the general types. Fig. 4 shows a modification of the vertical single-acting machine which may be said to be typical of the compressor of small capacity.

Vertical Single-acting Compressors

The accompanying illustration, Fig. 1, giving a lateral elevation in section of a characteristic vertical single-acting ammonia compressor, shows the relative arrangement of compressor and engine cylinders as well as the principal details of design. In this type of compressor the vaporized refrigerant enters the compressor near the bottom, passes up through the suction valve, located in the compressor piston, during the downstroke, and is compressed and discharged through the discharge valve, located in the safety head, during the upstroke of the piston. The compressor may be water jacketed or not; popular preference, however, is in favor of the water jacket, and most machines are so built.

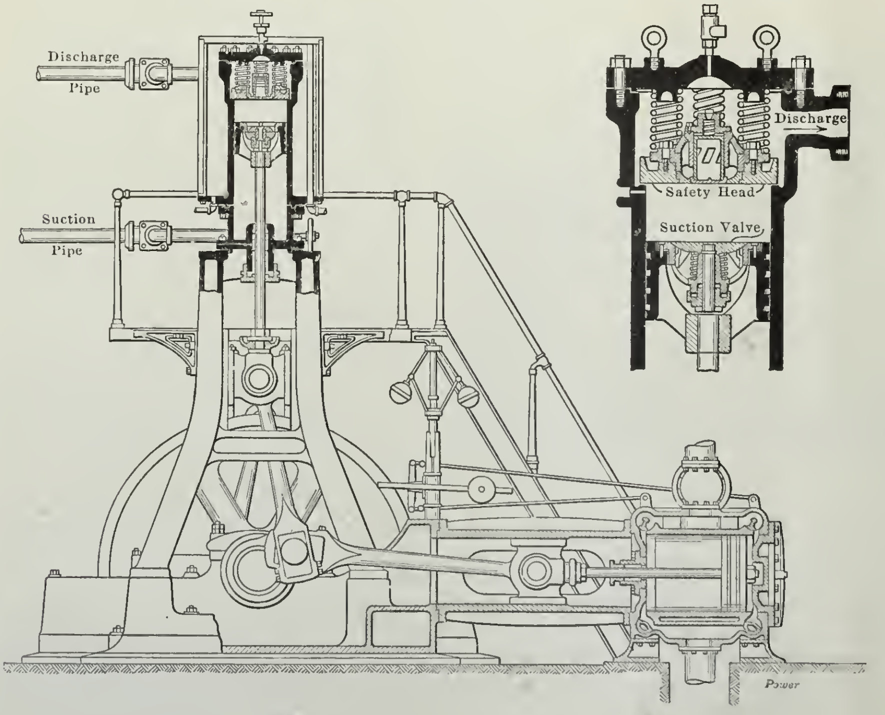


Fig. 1. Sectional View of Vertical Single-acting Ammonia Compressor

Vertical compressors possess the advantage of requiring less floor space than horizontal machines and the disadvantage of being less accessible for repairs. The inaccessibility of suction valves located in compressor pistons is offset by the unmistakable advantage offered by this type of machine in that these suction valves can be made of generous area, and the inertia of the valve tends to hasten its closing promptly as the piston reverses at the lower end of its stroke, thus preventing opportunity for gas to escape from the cylinder during the time required for the acting of stationary valves (unless they be heavily spring loaded, which tends to prevent the back pressure in the cylinder from quite reaching the hight of that in the suction line from the coolers). Inertia also tends to open the valve immediately when the piston begins its downward stroke, giving full opportunity for the cylinder to fill. The spring below the suction valve should be of such strength as to almost balance the weight of the valve, so that its inertia may act promptly at each end of the stroke.

Vertical single-acting compressors are usually provided with a "safety head" which is normally held securely to its

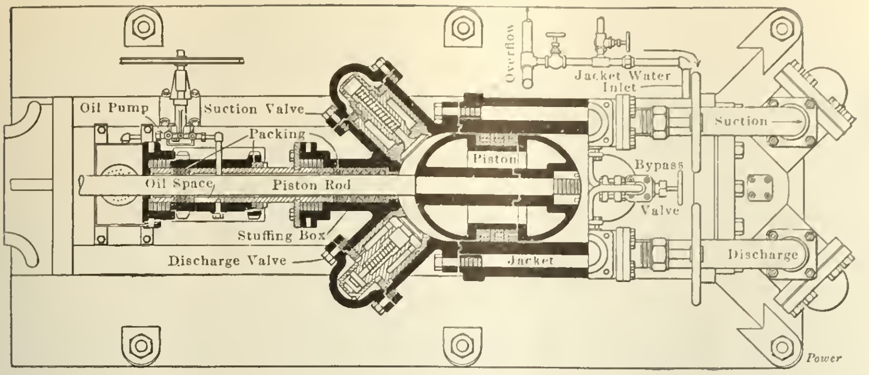


FIG. 2. PARTIAL SECTION OF HORIZONTAL DOUBLE-ACTING AMMONIA COMPRESSOR

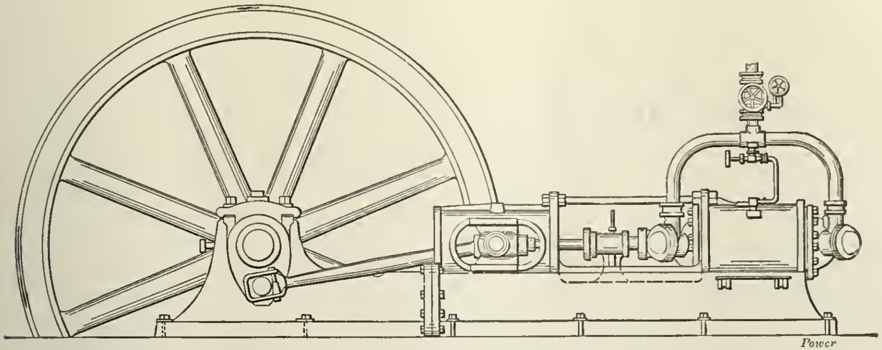


FIG. 3. TYPICAL HORIZONTAL DOUBLE-ACTING AMMONIA COMPRESSOR

seat by strong springs, but which, in the event of abnormal quantities of liquid ammonia or broken parts entering the cylinder above the piston, are pushed back, compressing the springs and thereby saving the machine from the strains that would otherwise occur. One of the principal advantages claimed by the advocates of the single-acting compressor is that the use of the safety head allows the compressor pistons to be operated with less clearance than would be practicable in the case of double-acting machines, a condition which insures a more complete expulsion of the gas.

THE HORIZONTAL DOUBLE-ACTING MACHINE

Fig. 2 represents a horizontal half section of a characteristic horizontal double-acting ammonia compressor. The right-hand portion of the cut shows the exterior of the head end of the compressor-cylinder valve housings, suction and discharge connections and valves. The remaining portion of the cut shows the details of construction of the compressor cylinder, water jacket, piston suction and discharge valves, double stuffing box and means of lubricating the piston rod. The outer wall of the water jacket is formed by the main frame casting, which is bored and fitted with a working cylinder liner consisting of a straight sleeve forced into place by hydraulic pressure and bored to the required size. The valves in this type of compressor are arranged radially to the hemispherical cylinder heads. The piston rod is provided with a primary stuffing box where it enters the compression cylinder. The packing in this box is tightened by a primary packing nut which carries a long sleeve, the other end of which is provided with a secondary stuffing box and packing nut. The main stuffing box, containing the bulk of the packing, withstands the high pressure of the ammonia in the compressor cylinder. The small stuffing box at the end of the sleeve is provided with sufficient packing to withstand the pressure of the oil circulated by the oil pump through the hollow sleeve surrounding the piston rod, in order to insure constant lubrication of and to maintain an oil seal on the main stuffing box.

The general appearance of the horizontal double-acting compressor cylinders just described is shown in the longitudinal elevation, Fig. 3.

INCLOSED CRANK-CASE COMPRESSORS

In addition to the two principal types of compressor previously described, the inclosed-crank type is deserving of mention because of the great number of such machines of small capacity now being installed. Details of design of this type of compressor are even more varied than those of the machines already described, and it is difficult to point out a single design that can be said to be more characteristic of the type than another.

In the illustration, Fig. 4, the refrigerant vapor enters the compressor cylinders through suction valves located in the cylinder head. Valves so located cannot be made of so liberal dimensions as

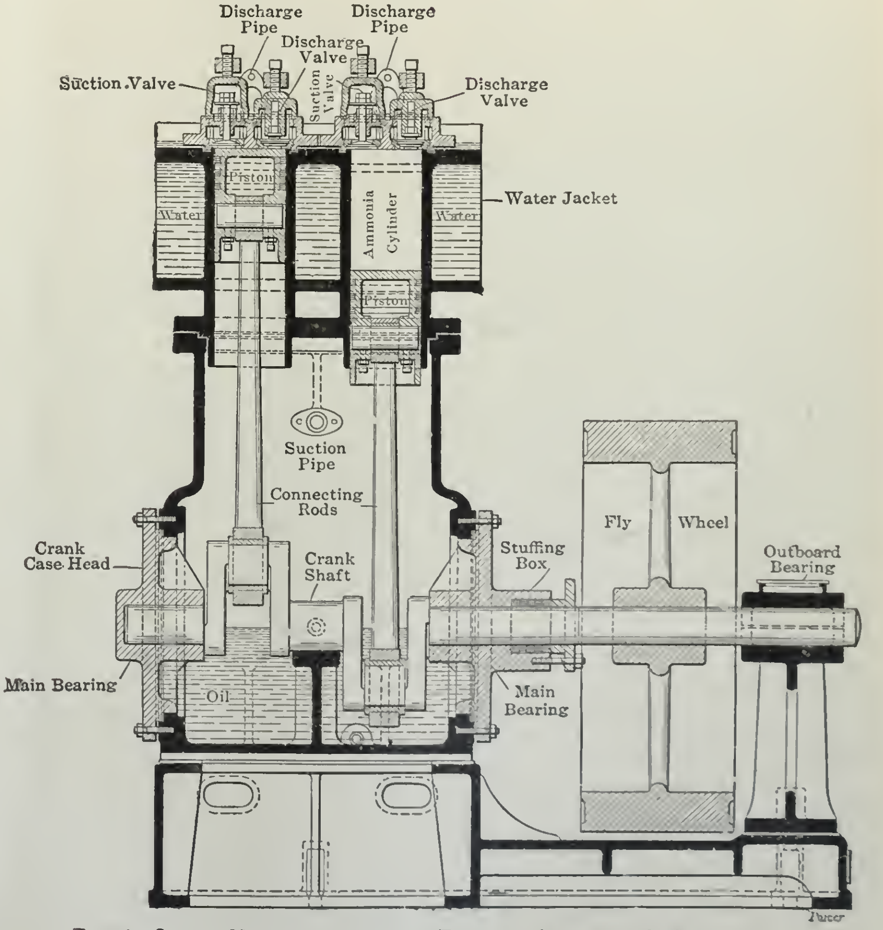


FIG. 4. SMALL VERTICAL INCLOSED TYPE OF AMMONIA COMPRESSOR

those located in the compressor piston, and the assistance which inertia offers in the way of opening and closing suction valves located in the piston cannot be realized. To offset this disadvantage, oil from the crank case is much less likely to be carried over into the condenser and low-pressure side of the system.

Machines of the inclosed type are especially adapted to use where little, or only inefficient, attendance is available. The details of construction which make less attention possible in this type of machine are principally the stuffing box and the main-bearing lubrication. The crank case being filled with oil to the center of the crank shaft, and the outboard bearing being usually ring oiling or provided with a compression grease cup, little attention to lubrication is necessary. There are no reciprocating piston rods to pack, the only stuffing box required being on the crank shaft, where it is always well lubricated and not subject to such extremes of temperature as are the pistons in other types of machine. The cylinder, however, must be cooled, and the water jackets for this purpose are plainly shown in Fig. 4.

The Theory of the Cooling Tower

By Joseph H. Hart

The author shows how it is possible for water to leave the tower at a temperature of 25 to 30 degrees below that of the air and explains how the cooling effect is obtained by absorption, conduction, convection and radiation.

Cooling towers operate as the result of two separate and distinct conditions, which make toward their cooling effect, and these are very often in opposition. Cooling towers are used to-day solely for the purpose of cooling water, which has become heated by passage through either a steam or ammonia condenser and is hence unfit for further use in this device unless it is itself in turn cooled off, in which case, however, it can immediately be used over again in the condenser with full effect. This, then, covers the entire scope of the cooling tower, to effect a saving in the water consumption, to the extent to which this is influenced by temperature variations in the water itself in both the steam and ammonia condensers. Of course, there are a number of small additional applications in industrial developments, in which similar conditions hold, where cold water is used for cleansing or purifying purposes, or even for the condensing feature in a number of other applications, but all of these are comparatively unimportant, and the chief use of the cooling tower is in condenser work in steam-power and refrigeration plants.

Here, the saving is confined to a diminution in the water consumption in the plant, and to a saving under some circumstances in the power consumption as well. In extreme cases where the water is extremely bad for feed-water purposes, cooling towers have been installed with a total increased power consumption as the result of their use, but with a greatly increased efficiency when all the determining factors have been considered. Again, cooling towers can be installed with considerable satisfaction with merely a power saving in view, if the pumping duty is large and the water circulation very unusual. This is quite possible, since there is invariably a saving in water consumption and consequently a saving in power on this account, as well as in the performance of the actual pumping itself. In a general way it can be said that wherever the water is bad or difficult to obtain, it pays to consider the possible installation of a cooling tower. Again, the question of mere size may enter and prove a determining factor, since the larger a plant is, the greater becomes the absolute importance of small economies.

So much for the position of the cooling tower from a commercial viewpoint in its present-day development. Granting the position of the cooling tower and its duty as well defined, many questions arise as to the absolute values of the results attainable and to the reliability or uniformity of the results attained under wide variations of temperature and hygrometric conditions. A consideration of this question brings us into close contact with the ultimate theory of action of the cooling tower itself, and here it can be said that absolute values along these lines have not as yet been obtained. A cooling tower depends for its action upon the cooling effect of the air on the water, and this is accomplished through two entirely different agencies. Water can be cooled by air in direct contact with it, if the air is cooler than the water, by the ordinary processes of conduction and convection. This is the agency that performs the cooling duty in the closed type of cooling tower to a much greater extent than is imagined. The cooler the air is, under these circumstances, the more efficient is the tower in the performance of its duty. A closed type of tower operates under a forced draft, induced by means of fans, and the air almost immediately on entrance to the tower, loses its hygrometric cooling effect, since water is vaporized into dry air with astonishing rapidity.

The hygrometric cooling effect, so called, depends upon the absorptive power of dry air for water vapor when brought in contact with a water surface and the consequent cooling effect upon the water due to the evaporation of a portion of it and the abstraction of the latent heat of vaporization, which is required for this event, from the remaining water. The absorptive power of air for water vapor depends not upon the air itself, but upon the water vapor present in the air at the time and upon its temperature and pressure. The air acts really as an agent for the conveyance of the water vapor present in it, and is important only as its temperature and pressure affect the temperature and pressure of the water vapor conveyed in its midst. This action, however, is quite extensive, since the two materials are quite intimately mixed, and any variation in the temperature and pressure of the one immediately produces a pronounced effect upon the other, due largely to the latent heat capacity of the water vapor when saturated and subjected to small temperature variations.

For a simple understanding of the fundamental problem, the air should be eliminated, or not considered as present during the cooling process as carried on in the tower. The air is actually eliminated in one or two commercial processes involving the same phenomena, with much more pronounced cooling effect. This is actually the case in the so-called vacuum ice-making process, where the vaporization is accomplished by forced variation in the pressure and degree of saturation of the water vapor, with a consequent cooling effect of the water sufficient to transform it into ice. The cooling tower, therefore, is not burdened by the performance of a large duty, from a cooling point of view, at least when the possibilities are considered in other developments utilizing the same agency. However, conditions are essentially different to the extent that the conditions are forced or produced by mechanical means in the one case, whereas natural conditions are used in the other. The utilization of forced draft in the development of the closed type of tower, is a step in the direction of utilizing mechanical agencies in the development of the cooling tower for increased efficiency, and it is an interesting question in regard to the extent to which such developments can be carried with this object in view.

Nothing has been attempted in this line, except the forced-draft development, and this has undoubtedly resulted in a diminished mechanical efficiency, although it has placed the cooling tower of the closed type upon a much higher plane, at least from a reliability and commercial-efficiency standard.

The fact is that ordinary air loses its absorptive power for water vapor within a distance of from 2 to 4 feet after entrance into the cooling tower and from then on it produces its cooling effect solely and simply by conduction, convection and radiation. Saturated water vapor exists in the air in amounts ranging from several grains to the cubic foot at freezing temperatures to an amount as high as 250 grains to the same space at the temperature of boiling water. Its amount is dependent absolutely on the temperature, and its absorptive power at any temperature for water vapor is almost directly proportional to the amount of vapor it possesses at any temperature, to that which it should possess at that temperature if it were saturated. Thus it can be readily seen that ordinary air, under any hygrometric conditions, possesses an enormous capacity for absorbing water vapor, provided only that it is heated sufficiently, and in no case need this exceed that of boiling water. The presence of the air does slightly influence the amount of water vapor present in the space occupied by it, but its effect is comparatively slight in comparison to that of the other factors, and it can be neglected in most calculations in this work.

Thus it can be readily seen that these two cooling agencies act in some cases in direct opposition in their effect in the cooling tower. The cooling effect, due to absorption of water vapor, is proportional more or less directly to the temperature, and the higher the temperature the less is the effect due to the absorption of heat by the processes of conduction, convection and radiation. In some extreme cases, notably of the closed-tower type, where the amount of air furnished can be regulated at will, the cooling effect, due to absorption of water vapor alone, can often be carried so far that the water will leave the tower at a temperature from 25 to 30 degrees below that of the air which produced the cooling effect. Again, it can be said, that the effect due to conduction and convection varies slightly with the temperature, whereas the variation in absorptive power for water varies greatly with the temperature. Hence, the hotter the air is, the greater will be its cooling effect due to absorption, and the less its effect due to conduction, etc. In the example shown the cooling action by conduction, at least during a portion of the process, was undoubtedly negative, and the total cooling effect might possibly have been greatly increased, if the conduction had been almost eliminated or the absorption increased by raising the temperature of the ingoing air.

Both of these modifications are comparatively easy to attain in actual practice. The elimination of the conducting feature can be accomplished by reducing the size of the tower, in the closed type, or cutting down the path of the air in contact, or by speeding up the rate at which the air traverses the tower. In the open type of tower, on the other hand, the same result can be accomplished by installing fans for diminishing the time of air transit, since the length of path in this type is quite small in comparison to the closed tower. On the other hand, conduction plays a most important part in the closed type of tower under adverse hygrometric conditions, such as are often met with under normal atmospheric conditions. Thus, when it is raining, or the atmosphere is saturated with moisture, the cooling effect due to absorption of water vapor is absolutely negligible, and the cooling of the condenser water can be accomplished by conduction alone. Of course, this is carried out in practice by speeding up the fans, and thus increasing the amount of air that passes through the water in a given interval of time. Also, air often exists in a condition approximating this, and combinations, in which the air possesses small absorptive power, but quite low temperatures, are common, and by this means the closed type of cooling tower can be operated under all temperature and hygrometric conditions with a degree of uniformity and reliability in the amount of cooling produced that cannot be met with in any of the open types of towers now on the market.

Thus, the question of the possible heating of the air before advent into the cooling tower is of great importance. It can be accomplished with considerable ease in many installations as they exist in actual operation and with minimum changes in construction and operative details. The majority of cooling towers have been placed on the roofs of buildings, in general to economize space and to allow greater access to the air. This is almost invariably the case in installations of the open type of tower, since they depend for their supply of air on natural agencies. However, they are often placed in this location for an entirely different reason. The hygrometric condition of the atmosphere varies at all seasons of the year very greatly with the hight above the surface of the ground and on this account more satisfactory operation is obtained with the cooling-tower installation on the roof of the building, since the moisture content of the air is much less here. If then the air supply is drawn from the interior of the building for such a tower it will invariably be much warmer and hence possess a much greater absorptive power and water-cooling effect. This is quite possible in most power-plant installations, since the air could be readily drawn from the loft above the boiler or engine rooms, and would be well suited for the purpose.

So much for a possible suggestion in regard to an easy economy in the operation of the closed type of cooling tower. Many other interesting possibilities present themselves in regard to constructive details with the object of further increase in efficiency. Of course, the statement that air loses its absorptive power for water vapor in from 2 to 4 feet in its passage through the tower depends on ideal conditions existing in regard to the distribution of the water in the space traversed by the air. These are not met with in practice and the operative distance can be increased to at least three times this amount with satisfaction on account of the poor distribution of the water, the often high but variable speed of the air, and the interaction of these two effects. A wide and uniform distribution of the water is not so necessary in the closed type of tower as in the open one, since the length of air path in the closed type is much greater. In actual practice, a wide and uniform distribution of the air is not always easily attainable. This is especially the case in the open type of tower, since the air currents cannot be regulated at all, and they tend to redistribute the water and further cause it to be piled up on the leeward side of the distributing devices or pans, so that this feature in turn affects the efficiency of the distributing device, and augments the evil. This shows another reason why the open type of tower may prove unreliable in its cooling action even though hygrometric and temperature conditions are favorable. In some types of open towers, in the case of a high wind, fully 60 per cent. of the surface of the distributing pans may be absolutely dry, showing the inefficiency of this device under some circumstances.

The object of this article was, however, not to consider the relative merits of the two types of towers, but to consider the theory of the cooling tower in the abstract. Both towers possess special advantages, at least from a commercial point of view. The open type can be installed at a much lower first cost than can the closed type, and its cost of maintenance, at least in regard to operative expense, is much less. The closed type, on the other hand, is a more permanent structure, and more reliable in the performance of its duty. Sufficient has been said to show the relative positions of the two types of tower now on the market, and further to show that the theory of these has not been fully worked out and that interesting possibilities present themselves for future work in this field, and let us hope that progress will be more rapid than it has been in the past.

The Experience of Ryan and Casey

By T. T. Roe

Bordering on the open prairie on the northwest side of the city, stood the yards and planing mill of Olson & Co. The street in front of the plant was a dirt road, impassable when wet, save by using the flat rails of the horse-car line, over which occasionally passed an ancient car drawn by a team of fat steeds. A few unfenced cottages were scattered about and from these rambled together children, chickens and goats.

The mill was a small affair, employing but a dozen men. The power department was in one narrow room and consisted of a 60-inch by 16-foot boiler, having one wall as part of the building, the opposite one being run up to support the floor above, which was used as a drying room. In front of the boiler and not more than 6 feet from it was a center-crank engine of the single-valve type, exhausting into a small closed heater. A small steam pump completed the outfit. On the right of the boiler was a drop door opening into a little coal shed. On the left was a glass door that led into a small narrow room provided with a bench and vise and a door opening outside. To reach the top of the boiler one went outside and climbing a rickety ladder opened a 24x24-inch door and crawled over the rear arch. Few had the curiosity to crawl through this opening and encounter the dust of ages. That the plant had run in this manner a number of years without being destroyed by fire was more astonishing than the failure of a butt-strap triple-riveted joint and could be attributed only to an all-powerful streak of luck.

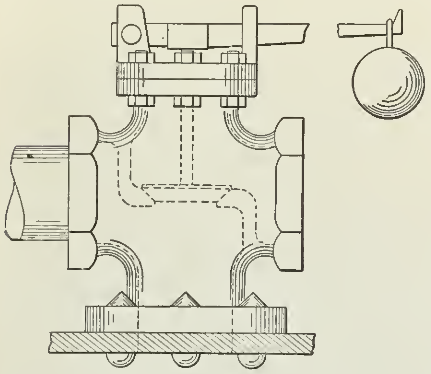


Fig. 1. Original Ball and Lever Valve

The engineer in charge was a husky old man with a warlike mustache directly in the rear of a black dudeen that defied continually the smoke ordinance. That he was a fine mechanic no one ever denied and, indeed, Ryan himself admitted the fact frequently and casually as one would comment on the weather. He was engineer, superintendent, repair man and the boss of the entire outfit.

The fuel consisted of shavings and the supply was inexhaustible. The operation of firing was very simple and consisted of opening the large firebox door, taking a big scoop full of shavings and packing the furnace full. If Ryan now and then used a few shovels of stock coal it followed that the shaving pile increased proportionally and with an eye for saving fuel he troubled the coal pile but little.

To this peaceful scene drove Casey, a riveter, and by grace of membership to a political club, a city boiler inspector. The chief had complained about inspections being overdue and Casey, scouting in this section, saw the stack afar off.

At the office of the mill his rude voice scattered the peaceful owners who promptly referred the matter to Ryan. The certificate was found and proved to be three years old. At a glance Casey knew that Ryan would not assist in the "shaking-down" process and loudly demanded that they pull the fire at once or stand a fine, whereupon Ryan proceeded to enter the telephone booth and call up his friend, the mayor, and report this red-haired intruder. This action, as intended, caused Casey to cool down and it was agreed the boiler would be ready for inspection two days later.

On the day set, Casey arrived with a helper as husky as himself. The boiler was cold and filled with water, the pressure used being 70 pounds. The water test was 105 pounds, Ryan attending to the safety valve.

Now, Casey was out for trouble and bound to find something on which to base a kick. But the tubes were tight, the seams and rivets showed no leaks and the fire sheets and heads were good. The brickwork was all right, and the water column, gage cocks, etc., were beyond criticism. The grin of superiority on Ryan's face and the air of tolerance with which he glanced at Casey's endeavors to get back at him caused Casey to determine that a kick would be forthcoming. Failing to find cause, he went to the outer door and climbed the ladder leading to the top. Through clouds of dust, he proceeded to the dome. Here he found a tee ball-and-lever safety valve as shown in Fig. 1. Now, in this city, the law called for spring-loaded safety valves and did not allow ball-and-lever valves to be used. Here was his chance to kick and he loudly proclaimed that if within three days the boiler was not provided with a pop safety valve, Mr. Ryan would have to give up his license. He tore out a slip of paper and wrote out a formal notice, serving it on Ryan in the presence of the owners. Ryan handed the paper to the owner and remarked that he had better throw it away and get the mill going, as they were behind on orders. The blood of Casey boiled and be retorted fiercely, whereupon Ryan told him to remove his shield and he would toss him over the roof. Casey left, vowing he would personally escort Ryan to the workhouse upon his return.

Fig. 2 shows how Ryan converted the 4-inch lever valve into a 4-inch pop valve. The spring was an ordinary draw-bar spring picked up on the railroad. Tension was produced by drawing down on the upper nuts and the valve relieved at 70 pounds, closing at 66 pounds. The valve was reduced to 3/16 inch where it touched the seat, thus affording prompt closure, which would not have been the case if the lip had been wide.

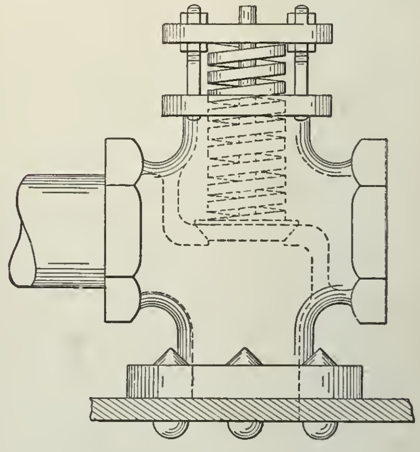


Fig. 2. Valve as Altered

Upon Casey's return three days later, armed with a warrant to arrest everybody in sight, he was told to see the new pop valve and as he crawled on top, the valve happened to blow off. Finally he viewed the valve and with curses descended. Tired of the man, Ryan threw him through the window into the slack coal pile, relighted his pipe and resumed the work of passing the shavings into the furnace. Casey arose and to his helper's inquiry if he would run Ryan in, responded: "Lave him alone, Dan, he's a bull of a man."

Coal and Ash Handling at Large Plant

By Warren O. Rogers

> *The Birmingham Railroad, Light and Power Company has equipped its power plant with an unusually complete system for handling the coal and ash for the 17,000 horsepower of boilers installed. The plant is one of the largest in the South.*

The method of handling coal and ash at the power plant of the Birmingham Railroad, Light and Power Company is somewhat different from that used in any other large station, as far as the writer is aware. The thirty-one boilers, aggregating over 17,000 boiler-horsepower; naturally consume considerable coal and consequently produce a large amount of ash.

Fig. 1 shows the layout of the plant in cross-sectional view, also the ash tunnel between the two rows of boilers and the coal pockets on the outside of the boiler house. Fig. 2 shows a view of the coal-handling apparatus. The coal is delivered in the regulation-size coal-carrying car and emptied through the bottom hoppers into the coal pockets, which are arranged as shown at *A*, Fig. 1. These pockets extend the entire length of the boiler room and each pocket is provided with a gate operated by a handle from a walk-way alongside the belt conveyer.

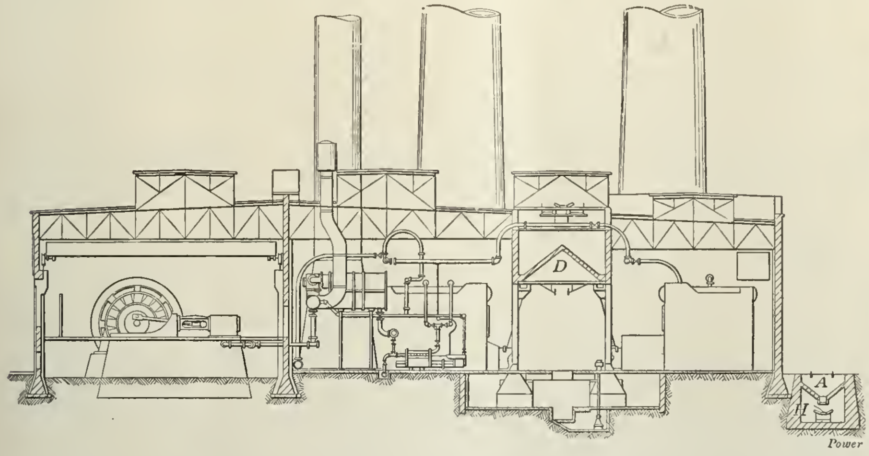


Fig. 1. Vertical Section through Plant

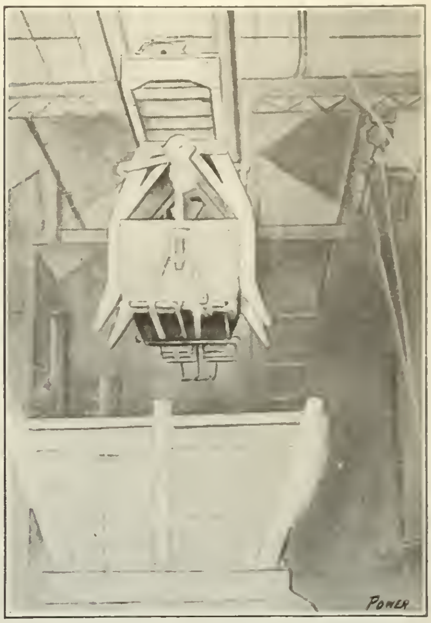


Fig. 3. Ash Skip

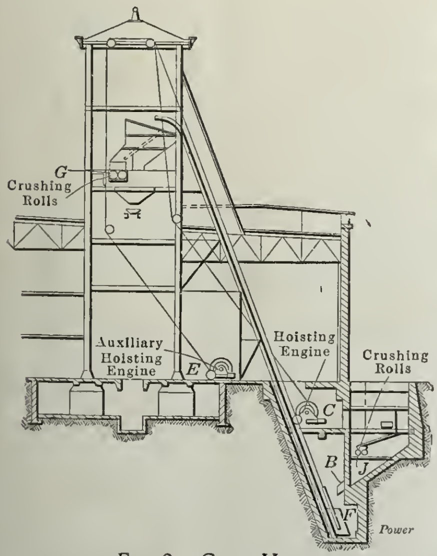


Fig. 2. Coal Hoist

Under these pockets is a belt coal conveyer, which takes the coal away from the pockets as the attendant allows it to feed through the bottom. The coal is delivered to a set of crushing rolls and passes through them into a small concrete storage bin. The outlet of this bin is fitted with a swing gate, which fits in the trough *B*, Fig. 2, and is operated by the hoisting engineer who is stationed at *C*.

Over the boilers is located the main coal bin, the construction being of concrete, with the bottom shaped as shown at *D*, Fig. 1, so that the coal supply will freely pass to the coal chute running to each stoker. Over the bin is placed a belt conveyer, operated by a motor; above the conveyer at the receiving end is a second set of crushing rolls.

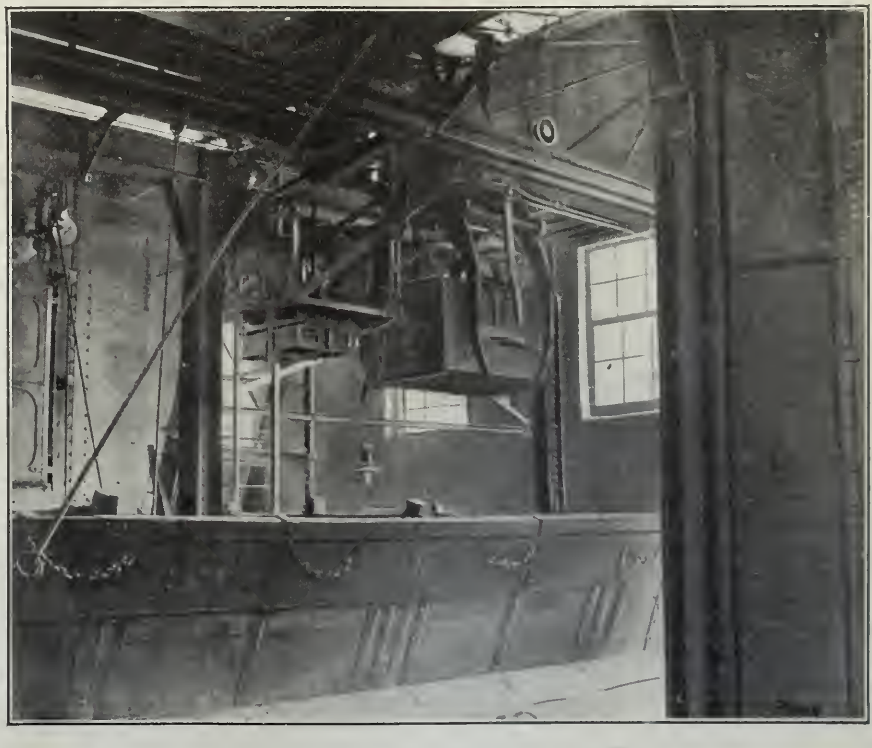


Fig. 4. Ash Car, Skip and Traveling Ash Hoist

From a point above the crushing rolls, *G*, Fig. 2, to a point in the pit, is an incline hoist on which a coal skip is operated by means of a steam hoisting engine, shown at *C*. An auxiliary hoisting engine is placed at *E*.

When coal is required in the upper bin the operator starts the underground conveyer *H*, also the lower crushing rolls *J*, with the skip *F* in its lowest position, as shown in Fig. 2. The gate of the chute *B* is then opened, which allows the coal to fill the skip. The gate is then closed and the skip of coal hoisted to the top crusher rolls *G*, which can be used or not, depending upon whether the lower rolls *J* are in working order or not. The coal, after being delivered to the top conveying belt, is carried to the proper point over the coal bin and discharged where desired. The upper conveyer is housed on the roof of the boiler house, as shown in Fig. 2.

Handling the Ash

The method of handling the ash is also out of the ordinary. In the center of the boiler room is placed a standard-gage track, which serves a double purpose, that of allowing loaded freight cars and empty ash cars to run into the boiler room for the removal of ashes.

Between the rails of this track are open spaces which permit hoisting the ash skip from the basement by means of an overhead traveling hoist. Beneath the boiler-room flooring is a sub-basement. Two rows of posts support the track and boiler-room floor. Between these posts is a trench in which the empty skips are lowered ready for filling.

The ashes from the boiler furnaces fall into an ashpit, the bottom of which is at such a hight that a common ash barrel can be wheeled alongside and loaded with ashes, which are hoed out by hand. The ashes are then dumped into the nearest skip, of which there are a number, and when full the operating boy with his traveling hoist lifts the skip by means of hooks which engage in the chains shown at each end of the skip, Fig. 4. The skip is held in place by arms, which engage with lugs, one arm being placed on each side of the skip. This prevents accident to the workmen as the loaded skip is hoisted and conveyed from one end of the boiler room to the dumping point. The four legs of the ash conveyer, shown in Fig. 3, are for the purpose of steadying the loaded ash skip and also to cause it to always hoist into a certain position so that the catch arms and lugs will always engage with each other. When the load of ash is hoisted and secured in position, it is run to the far end of the boiler room, where ash cars are waiting, Fig. 3. Fig. 4 shows a side view of the hoist, skip and ash car.

Natural Draft and Chimney Design*

By H. G. Brinkerhoff

A consideration of the factors affecting the principal dimensions of chimneys, in which a few prevailing erroneous impressions are corrected. A number of constants used in practical chimney design are also given.

Nearly every treatise on chimneys and natural draft contains the following statement: "The natural draft obtainable from a chimney depends upon the column of gas inside the chimney being lighter than the air outside." This, however, is not the only cause in producing draft, as a wind blowing over the chimney produces a considerable suction without the inside air or gas being heated at all. This fact is important, but as the wind is such an uncertain element, it cannot be included as a factor when calculating the dimensions of a chimney.

There are two principal dimensions to be determined, the hight and the cross-sectional area of the chimney. A proper hight is of the greatest importance as this produces the pulling force, which must be sufficient to overcome the friction through the grates, flues and chimney.

A hight sufficient to produce a strong, vigorous draft involves considerable expense, but this can be offset by the use of fewer boilers. It is becoming to be appreciated generally that there is economy in running with fewer boilers due to the following causes: There is less expense involved in banking the fires at night; with an intense fire the gases are more completely burned to CO_2; a better circulation is set up; and less smoke is produced.

It has come about in practice that the purchaser wishing to secure 1000 boiler horsepower is furnished with 10,000 square feet of heating surface or ten square feet per horsepower, which is a safe factor to use with ordinary draft. But whether the purchaser obtains 1000 horsepower or twice that amount is purely a matter of supplying heat enough which, in turn, is practically limited only by the draft necessary to burn the requisite amount of coal.

The boiler horsepower does not need to be considered when determining the hight of a chimney, although with rare exception this is now erroneously being done. It is universally taught that draft varies with the square root of the hight, but this use of the word "draft" applies only to the capacity, which is largely a matter of area. Chimneys produce a draft which is directly proportional to their hight, that is, a 200-foot chimney will produce a draft pressure which is twice that of one 100 feet high.

Many papers have been written on the subjects of chimneys and drafts, generally developing abstruse or speculative theories. The most helpful and practical one that I have come across is an article by T. F. J. Maguire, a summary of which is as follows:

The loss of draft in boiler settings is placed at 0.3 inch for water-tube boilers, increasing to 0.4 inch with 50 per cent. overload. Of course, this is intended to be used with judgment to suit the construction and baffling of the particular boiler under consideration. The writer believes that horizontal return-tubular boilers would require the same allowances, and the vertical types about 0.15 inch.

Maguire allows 0.1 inch per 100 feet of length for circular iron flues and 0.05 inch for each right-angled turn. If square or rectangular in shape, the foregoing should be increased 25 per cent.; for brick flues it should be increased 30 per cent.

DRAFT PRESSURES REQUIRED IN FURNACE.

Kind of Coal.	Pounds of Dry Coal Burned per Square Foot of Grate per Hour.						
	15	20	25	30	35	40	45
	Draft in Inches of Water.						
Eastern bituminous coals	0.12	0.16	0.20	0.27	0.34	0.42	0.52
Western bituminous coals	0.15	0.20	0.25	0.33	0.42	0.52	0.65
Semi-bituminous coals	0.15	0.20	0.28	0.37	0.48	0.60	0.80
Anthracite buckwheat, No. 1 and larger	0.45	0.70	1.00				
Anthracite buckwheat, No. 2 and No. 3	0.75	1.30					

Let these factors be applied to a concrete case where 25 pounds of soft coal are burned under horizontal return-tubular boilers with two right-angled turns in the breeching.

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*Abstract from paper delivered before the National Association of Cotton Manufacturers, at Portsmouth, N. H., September 17, 1910.

Furnace draft for 25 pounds soft coal	0.2
Horizontal return-tubular boilers*	0.3
Fifty feet of steel breeching	0.05
Rectangular shape requiring 25 per cent. additional	0.0125
Two right-angled turns	0.1
Round-brick chimney 150 feet	0.15
Brick surface 30 per cent. additional	0.045
Total draft required	0.8575

*The boiler resistance increases with the velocity. Therefore, if 25 pounds per square foot of grate area develops a considerable amount above rating, add a third to the boiler resistance. In this case it would be 0.4 for the horizontal return-tubular boilers instead of 0.3.

This is approximately 0.9 inch, and as 0.6 inch can be obtained with 100 feet of stack, then a 150-foot stack would provide the required pressure.

It may be pointed out that no allowance has been made for the effect which an economizer would have on the draft. But as the economizers with straight open passages usually have a greater free area than that of the flue, it can be counted as flue length.

The selection of a suitable area for a chimney is as frequently a matter of guesswork as the determination of the hight. Some have a pet rule of making the cross-sectional area proportional to the grate area regardless of the fact that if twice as much coal were burned in one case as in the other, it would call for twice the flue area, because of one having double the volume of gas. As a high chimney costs so much more in proportion than one of large area it is common fallacy to curtail on the former and increase the latter. Thus, for 1500 horsepower the owner provides himself with a stack of a hight suitable for only 1000 horsepower and uses an area good for two or three times this amount. Then he wonders why he has difficulty in getting 1500 horsepower and has to have recourse to artificial draft. The theoretical area of chimney required for 1000 horsepower with a stack only 100 feet high is 7 square feet under ordinary conditions. In my own practice, I usually allow 20 square feet per 1000 horsepower of boiler installation. For 2000 or more horsepower I reduce this allowance about 10 per cent., increasing the reduction on very large plants to perhaps 20 per cent., and keep 16 square feet as a safe limit. For smaller plants, say, 500 horsepower, I would allow 12 square feet, which is then more than three times the theoretical area on the moderate basis of a hight of 100 feet.

If the chimney is over 100 feet in hight, the theoretical requirement of 7 square feet is reduced, but I never use less than 16 square feet of area.

The least allowable area should give the best draft. Were the 20 square feet per 1000 horsepower to be increased to 30 square feet it would mean, in a travel of 150 feet up the stack, together with a flue length of 30 feet to reach its port, that there is 645 square feet of additional surface to be heated to the same temperature as the gases with exposure to radiation loss, as well as increasing the air leaks.

An unutilized excess of draft pressure is bad and the boilers should be fired to as near the limit as possible, but at times there will be necessarily a surplus suction produced by the wind. That this excess is common is shown by the operation of the automatic regulation of dampers which continually cuts off 25 to 50 per cent. of the available draft throughout the entire day. Of course, the damper regulation effects a saving in preventing an undue excess of air through the grates, but it should, as far as possible, produce a more intense fire and by heavier coaling enable the plant to run with fewer boilers.

In the matter of draft regulation, it is much better to have a regulator that operates slowly than to have one very sensitive which will open and close on a slight variation of pressure, thereby letting in an excess of air one minute and cutting off the air supply the next minute. With such a sensitive damper, it is only necessary to limit the range of damper opening, so that it does not shut tightly nor open to its fullest extent. The regulating damper should be as near the chimney as possible, and if it were placed at the top of the chimney, it would be even better, as it would then hold back that much more heat. As operated in many plants, when the damper is closed at the port, the brick lining or core loses much of its stored-up heat and the chimney cools off; then when the demand comes for more steam, it is some time before the chimney recovers enough lost heat to produce a vigorous draft. The condition becomes worse where the regulation is applied to dampers in the boiler uptakes, for during the time the damper is closed the flues have cooled as well as the chimney.

The Economical Use of Lubricants

By William M. Davis

> *The efficient and economical use of lubricants depends principally upon the selection of the proper lubricant for the particular service in which it is to be used, obtaining the lubricants at the lowest market price and a well ordered system of handling, distributing and applying the lubricant.*

Lubricants may be divided into three general classes according to their forms: fluid, plastic and solid. To the first class belong the various oils, to the second class the greases, and to the third such substances as graphite, soapstone, mica, etc.

Where the speed is high and the pressure great, oils are the most satisfactory. They form an elastic coating to the metal which keeps the contact surfaces apart and carries away the frictional heat. Another advantage is that they can be had in almost any desired grade or density, and do not become rancid nor gummy. In the early days of engineering, the oils were all of animal or vegetable origin. These, although good lubricants, had their disadvantages as they were expensive and had a tendency to become rancid. In recent years these have been almost entirely displaced by mineral oils which are, in many respects, better lubricants and are not subject to the disadvantages of the animal or vegetable oils.

The greases are more suitable on slow-moving machinery where the pressure is not great, but if used indiscriminately they will produce a noticeable increase in the friction load.

Greases may be divided into two classes: the lime and potash soaps which have a high melting point and those of a tallow base which have a low melting point. The first are made by forming a small amount of fatty oil into a soap by means of lime water, caustic potash or other alkali, and mixing it with a large amount of petroleum oil. Such greases may be made in any degree of density and have a melting point of from 140 to 180 degrees Fahrenheit. The greases having a tallow base are composed of a large percentage of tallow combined with an alkali and are brought to the desired density by means of vaseline-petroleum oils. These greases have a low melting point, usually from 116 to 120 degrees Fahrenheit.

The solid lubricants, such as graphite, soapstone, etc., have a limited field of use, being used principally as a filler for fibrous piston-rod packings, etc.; although a certain form of graphite has shown in laboratory tests to have a high lubricating value with a low coefficient of friction.

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*Abstract from paper delivered before the National Association of Cotton Manufacturers, September 15, 1910.

There are three essentials to be looked after in obtaining economical lubrication; first, the selection of such lubricants as will maintain cool bearings, have a low friction loss, etc.; second, obtaining these lubricants at the lowest market price; third, their economical use.

The selection of proper lubricants depends upon the class of machinery on which they are to be used. On light-running and high-speed machinery the light-bodied oils will give the best results; for slow-speed machinery, the heavier oils will be better. On slow-speed engines where the oil is fed from cups, a heavy oil should be used; and for high-speed engines, where continuous oiling systems are employed, a light-bodied oil should be used.

Where the plant is small, the most satisfactory method of purchasing lubricants is to buy from some reliable oil firm, but where the cost of lubrication runs into thousands of dollars, as is the case with many large companies, the best plan is to purchase on specifications, stating clearly just what is needed and awarding a contract for a year's supply to the lowest responsible bidder. Specification buying is fair both to the consumer and to the dealer.

The usual physical tests for gravity, flash point and viscosity, and in the case of cylinder oils, a chemical analysis for the percentage of fatty oil will give anyone who is familiar with lubricants a very close indication of the relative values of different oils. If one wishes to go further it is often possible to make actual service tests which will show any difference in the lubricating property of two or more oils. If it is desired to make a service test, select a bearing which is running under constant load and speed. Place a thermometer in the bearing so that the bulb rests on the shaft, and maintain a constant feed of oil. Have another thermometer placed in the room near the bearing, but out of any drafts, and note the room temperature.

Commence the test when the engine is started, note the rise of temperature at frequent intervals, and continue until the temperature of the bearing ceases to rise. Every bearing will in the course of a few hours reach a point where the heat is radiated as fast as it is generated. Deducting the temperature of the room from that of the bearing will give the temperature due to friction. If the engine runs in the daytime only, the bearing will cool off during the night; then repeat the test the next day with another oil. When making such tests it is well, after finishing one test and before commencing another, to wash out the bearing with gasolene.

Although it has been found in experimental work that the coefficient of friction often decreases with a rise in temperature, yet in every-day practice it is safe to assume that the oil which will keep the bearings the coolest is the best lubricant.

If it is desired to ascertain the lubricating value of two or more cylinder oils, take one oil and feed it at a given rate for a few days, then remove the cylinder head and wipe the surface with a piece of soft white paper. If a stain of oil is found it is evidence of good lubrication; if there is no stain and a liberal amount of oil has been used, it indicates either that the steam is very wet or that not enough fatty oil has been used in compounding the lubricant.

Next to the problem of buying, efficient and economical lubrication depends upon the methods of handling and applying the lubricant. It is a waste of money to fit engines with a continuous oiling system unless necessary precautions are taken to recover the oil used.

Another cause for loss often occurs in wiping up with waste. In most large plants it will pay to install some make of oil- and waste-saving machine, by means of which the oil is extracted and filtered and the waste washed, dried and used over again.

Receiving, storing and distributing lubricants are other important factors. If the plant is a large one, it will pay to provide storage capacity so that the oils can be bought in tank-car lots. But if it be a small plant where only a few barrels of each kind of oil are used per month, the oil should be kept in tanks so arranged that the barrels can be emptied into them by gravity. Care should be taken to see that the barrels drain out thoroughly.

It is customary in a large plant to have someone in charge of the oil house, to receive and store the lubricants, and to deliver them to the various engine rooms and departments, keeping a record of the amounts distributed.

No oil should be issued except on a requisition signed by the chief engineer, master mechanic or department foreman. In a small plant where the amount used does not warrant keeping a man especially to look after the lubricants, the oil house may be placed in charge of the general storekeeper.

At the end of each month the quantities of lubricants issued should be totaled, and their cost figured and charged to the individual departments using the oil. By comparing monthly reports, it can be seen at a glance whether the oil bill is increasing or decreasing and in which departments the differences have taken place.

New Power Plants for San Diego, Cal.

The San Diego Consolidated Gas and Electric Company, of San Diego, Cal., is adding to its plant four batteries of 500-horsepower Babcock & Wilcox boilers, which are equipped with the latest improvements and will furnish steam for a 2000-kilowatt Curtis turbine of the horizontal type driving a General Electric alternating-current generator. A portion of the old engine house is now being taken down and a new one erected, which will be equipped with an electrically operated traveling crane. Provision is being made for the extension of the power plant in the near future. It is intended to put in a double tunnel for the circulating-water intake and discharge. The tunnels are to be 5 feet in diameter and built of reinforced concrete. The discharge tunnel will be on top of the intake tunnel and both will be under the low-tide mark and about 1500 feet in length. The foundations for the turbine and auxiliaries and the engine house are now being put in, and the work is being done under the personal superintendence of H. M. Reilley, construction engineer, of Billesby & Co., of Chicago.

The San Diego Electric Company are also having plans drawn for the construction of a new power station to cost ultimately about $225,000. The structure will be one of the most complete and uptodate of the kind on the Pacific coast.

When completed, the new power plant will be about 200 feet in width by 300 feet in length. It will cover an area approximating 60,000 square feet, and will generate about 35,000 kilowatts.

The boiler room will have eight batteries of six oil-burning boilers each, which will require three large smokestacks. Glass and louvered skylights will furnish light and air to the workmen below. The engine room will be 75 feet in width by 300 feet in length, and from floor to floor will measure more than 60 feet. It will be amply lighted by six windows, each 50 feet wide and 50 feet high. An electrically controlled traveling crane, having a lifting capacity of 65 tons will run the full length of the room. A 10-ton crane located under the switchboard gallery will care for all motor-generator machinery. The switchboard gallery will be above the entire floor and will afford a clear view of the room at all times. As soon as the working plans have been completed the actual work on the building will be begun.

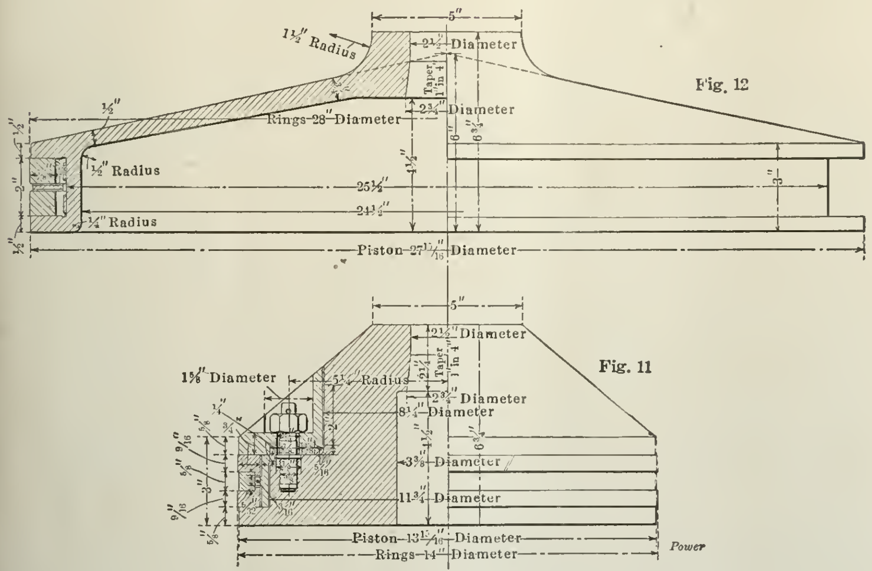
High Speed Compound Engine Design

By J. D. Speedium

The second instalment of a series of articles on High-Speed Engine Design in which the pistons, valves, guides, piston rods, crossheads, connecting rods and valve gears are considered.

PISTONS

The details of the high- and low-pressure pistons are shown in Figs. 11 and 12, respectively. As is the usual custom the high-pressure piston is fitted with junk rings, and is made of cast iron. The low-pressure piston is of wrought steel having a tensile strength of not less than from 76,000 to 84,000 pounds per square inch. For the latter piston, one type of ring is universally used, and is known as the "Rowan" ring; this is shown in Fig. 12, and consists of two ordinary ramsbottom-type rings of cast iron, turned to the exact size of the cylinder. These two rings are fitted into one wide groove, and between them is a steel wave spring which keeps them against the piston flanges. This prevents rattling and rapid side wear so often noticeable in high-speed engines when fitted with ordinary ramsbottom rings, unless made very deep in section. Behind each ring are placed other wave springs, which press the rings against the cylinder walls. As the pressures outward and sideways are adjustable independently of each other, the arrangement makes a very efficient piston packing for all low-pressure pistons; but for very high pressures and particularly where superheated steam is used, a restrained type of ring is absolutely necessary. The "Rowan" type of packing ring does not require a junk ring on the piston. This tends toward simplicity, and makes a light piston, but as it is necessary to make the weight of the high-pressure piston equal to that of the low pressure, for balancing purposes, there is no difficulty in arranging for a junk ring on the former. In the case of triple-expansion engines the intermediate pistons are usually fitted with rings of the "Rowan" type and no junk plates are fitted.



FIGS. 11 AND 12. HIGH- AND LOW-PRESSURE PISTONS

The designs of pistons here shown represent general practice. The low-pressure piston being of steel can with safety be made very thin, particularly as it is conical in shape. This makes a very strong form, and at the same time assists drainage. Wrought steel is the material generally used for these pistons, but cast steel is coming into common use. The following table gives the usual thickness of wrought-steel pistons:

TABLE 4. THICKNESS OF WROUGHT STEEL PISTONS.

Diameter of Piston.	Thickness of Plate near Boss.	Thickness of Plate near Outer Edge.
16 inches	$\frac{9}{16}$ inch	$\frac{5}{16}$ inch
20 inches	$\frac{5}{8}$ inch	$\frac{3}{8}$ inch
24 inches	$\frac{11}{16}$ inch	$\frac{7}{16}$ inch
30 inches	$\frac{3}{4}$ inch	$\frac{1}{2}$ inch
36 inches	$\frac{15}{16}$ inch	$\frac{9}{16}$ inch
48 inches	$1\frac{1}{8}$ inch	$\frac{11}{16}$ inch
60 inches	$1\frac{3}{8}$ inch	$\frac{3}{4}$ inch

High-pressure pistons are usually of a conical form, the depth overall being made the same as that of the low-pressure piston to enable piston rods of the same length to be used for both high- and low-pressure cylinders. To prevent the possibility of studs working loose, they are fitted with collars, or carried through the piston and riveted over on the under side, the nuts being secured by means of split pins. Packing rings of many types are in use, but the only successful ones are those of the restrained type. The design shown in Fig. 11, and known as the "Admiralty" ring, is extensively used and is simple. In this design the spring rings are prevented from being steam expanded, which causes excessive friction and wear, by lips which engage with a solid bull ring. As a junk ring is fitted, the spring rings can be made of deep section; consequently they will wear for some time before adjustment is necessary, and even then the junk plate can be closed a little to take up the play.

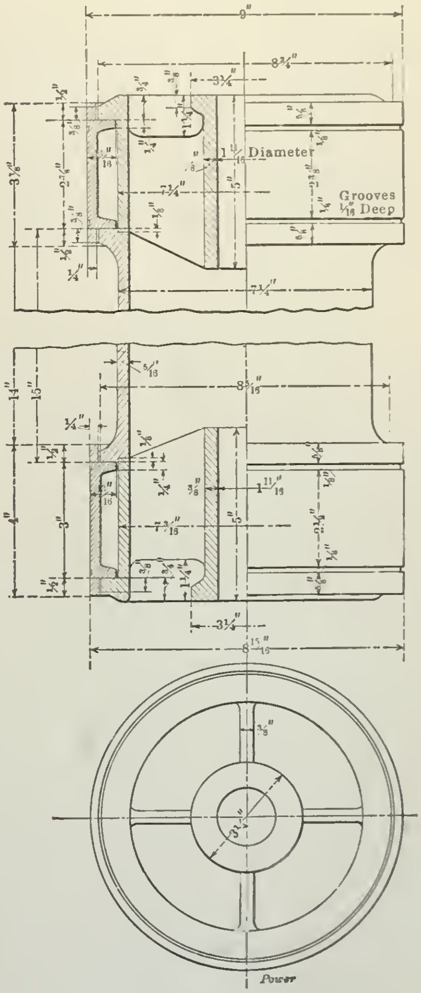


FIG. 13. HIGH-PRESSURE VALVE

PISTON VALVES

The type of piston valve most commonly used is the one fitted with solid rings as shown in the detail drawings, Figs. 13 and 14. This design makes a very simple, light and frictionless valve, which is

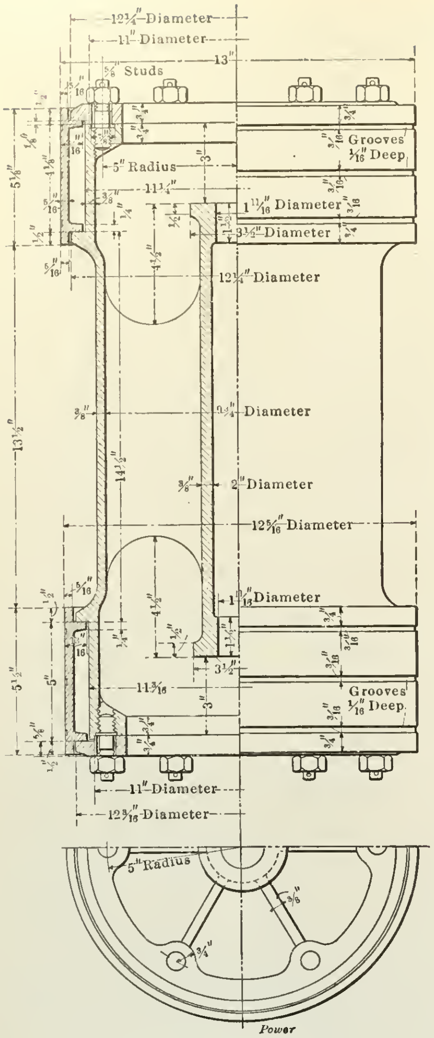


FIG. 14. LOW-PRESSURE VALVE

suitable for any condition as regards pressure or temperature of the steam. It may appear at first sight to be a very uneconomical design of valve, and also one which would readily wear and allow excessive leakage to take place. That might be the case if the engines were of the horizontal type, and the valves worked in a horizontal position, but as high-speed engines are all of the vertical type, and must of necessity be accurately built, the solid rings prove very efficient. It is, of course, essential to have the valve gear truly alined, as side pressure on the valves would cause wear. Solid valve rings will work for several years without appreciable wear, often not exceeding 0.003 inch after being in service for four or five years.

Fig. 13 represents the high-pressure valve. The body is built up in three pieces, held together by the valve spindle, sufficient clearance being allowed in the grooves for the rings to float without being slack. Both rings and valve are made as light as possible, which reduces the effect of inertia and relieves the valve motion. It will be noticed that lips are shown at both ends of the valve rings. These provide a ready means for altering the steam or exhaust lap without making any structural changes in the valve body. Valves are made of similar design to that shown in Fig. 13 in sizes up to about 12 inches diameter.

In Fig. 14 is shown the low-pressure valve. In this case the valve body is made in one casting, loose junk plates being fitted at each end to secure the valve rings. Studs are used for securing the junk plates, and they are locked in a similar manner to those on the high-pressure piston.

As the exhaust steam has to pass through the center of the valves, the diameter is usually determined by the area required for the steam. This should always be at least 10 per cent. in excess of the port area.

Grooves are turned in the rings as shown, and are necessary to retain the lubricant.

VALVE DIAGRAMS

In Fig. 15 are shown the valve diagrams for determining the necessary laps and angle of the eccentrics to give the required cutoff and compression, etc. Referring to the high-pressure diagram, it has already been shown that a mean cutoff of 61.5 per cent. is required to develop the maximum load; therefore the valve must be set to suit. As previously explained, it has been found advantageous for economy, to set the valve with negative lead, so as to obtain a late release with a reasonable compression. In this case a negative lead of ¼ inch top and ⅛ inch bottom will be found satisfactory, and a travel of 5 inches will give sufficient port opening. Roughly speaking, a travel equal to twice the width of the steam port will be found sufficient. The speed of steam through the slots in the liner at minimum opening to steam should not exceed:

High-pressure cylinder...	8,000 feet per minute
Intermediate-pressure cylinder................	9,000 feet per minute
Low-pressure cylinder....	10,000 feet per minute

In the case of compound engines the speed for low-pressure cylinders should be taken as above for the intermediate cylinders.

For exhaust, the valve usually fully opens the port; consequently the speed through the slots in the liner is less than through the cylinder ports.

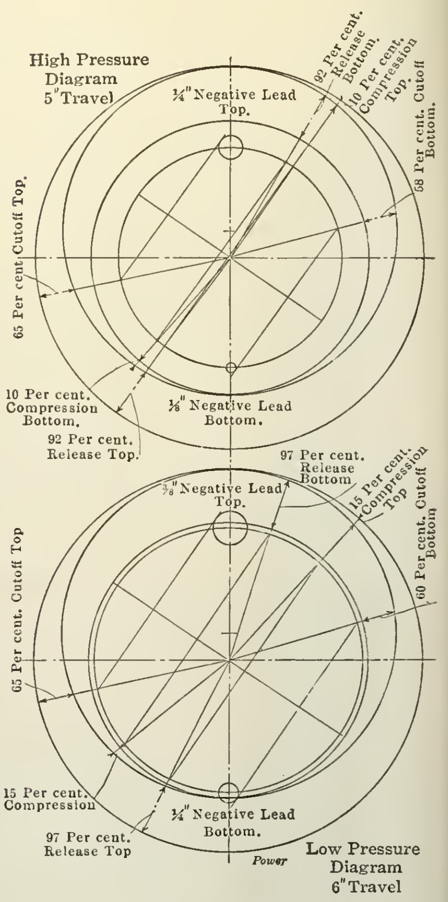
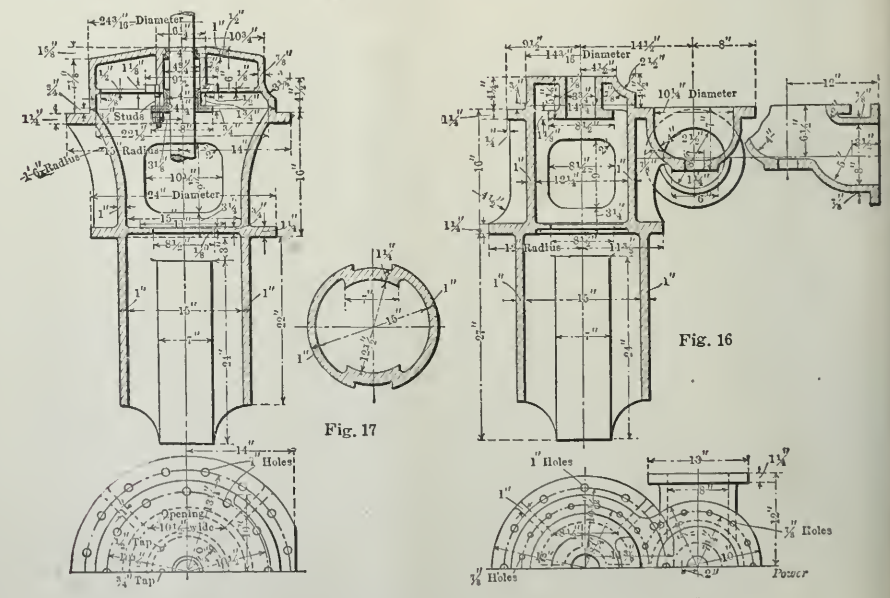

High Pressure Diagram 5" Travel
¼" Negative Lead Top.
92 Per cent. Release Bottom.
10 Per cent. Compression Top.
58 Per cent. Cutoff Bottom.
65 Per cent. Cutoff Top.
10 Per cent. Compression Bottom.
⅛" Negative Lead Bottom.
92 Per cent. Release Top.
97 Per cent. Release Bottom
15 Per cent. Compression Top
60 Per cent. Cutoff Bottom
⅜" Negative Lead Top.
65 Per cent. Cutoff Top
15 Per cent. Compression
¼" Negative Lead Bottom.
97 Per cent. Release Top
Power
Low Pressure Diagram 6" Travel

FIG. 15. VALVE DIAGRAMS

A compression of about 10 per cent. is required in the high-pressure cylinder, and as will be seen from the dia-



FIGS. 16 AND 17. HIGH- AND LOW-PRESSURE CROSSHEAD GUIDES

gram, this gives a release at 92 per cent., which is very satisfactory. The steam and exhaust laps can now be measured and the valve design completed.

In the case of the low-pressure diagram a travel of 6 inches will be found necessary, and to give approximately an equal distribution of power between the two cylinders a cutoff of about 62.5 per cent. will be required. Negative leads of ⅜ inch at top and ¼ inch at bottom will be suitable, and as the engine is condensing, a mean compression of 15 per cent. may be allowed, which gives release at 97 per cent. of the stroke. It is always a convenience to draw the valve and piston diagrams in conjunction with one another as shown in Fig. 15, as the setting can be read off at a glance.

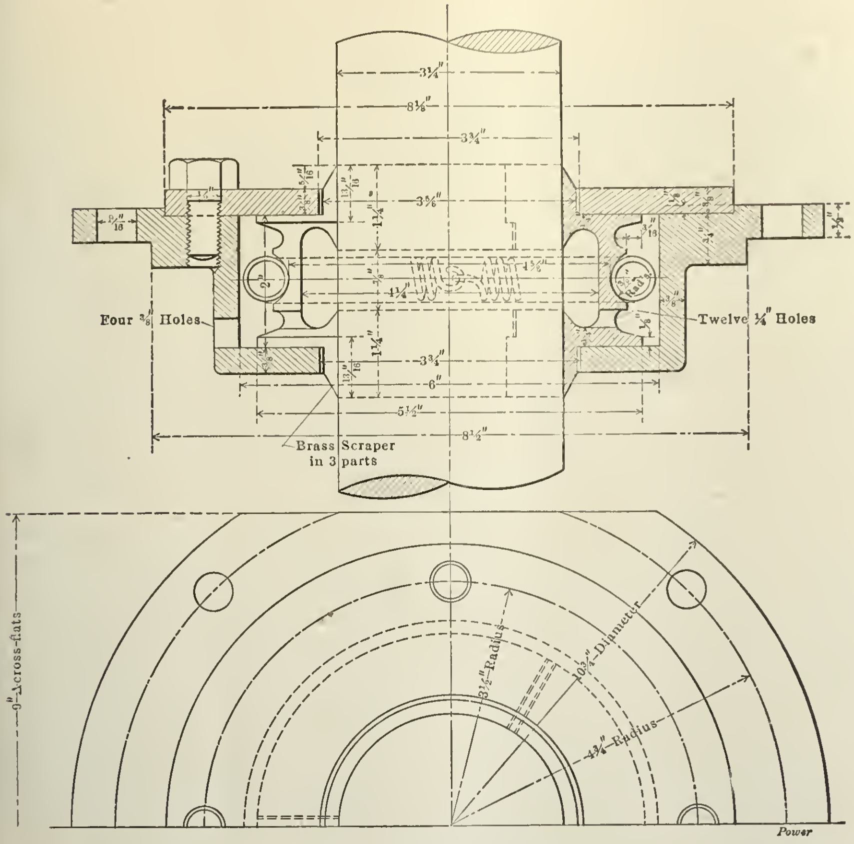


Fig. 18. Scraper Gland

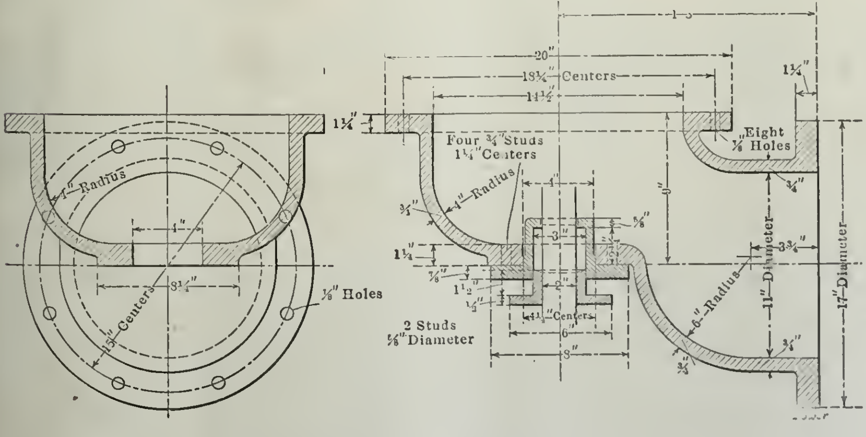


Fig. 19. Exhaust Branch

Crosshead Guides and Distance-piece Castings

In Fig. 16 is shown the distance-piece casting which supports the high-pressure cylinder, and also forms the bottom cylinder cover. It will be seen that the high-pressure exhaust branch is made in one with this casting, making a very compact and neat arrangement. This branch has to assume a rather peculiar shape in order to make room for the valve-spindle stuffing box above the top of the frame. With the bottom cover is formed a flange to which is attached the metallic packing.

The crosshead guides are of the bored type and of such a length as to allow the slippers to overrun 2 inches at each end of the stroke. The tubular section makes a very stiff construction, and as the cylinder-cover portion, which forms a spigot fitting the cylinder barrel, is machined from the bored guide, the guide is self-centering with the cylinder. On a level with the top of the frame is fitted another stuffing box or scraper gland, for the purpose of preventing water from the glands getting down into the crank chamber, and also to prevent oil from the crank chamber passing up the piston

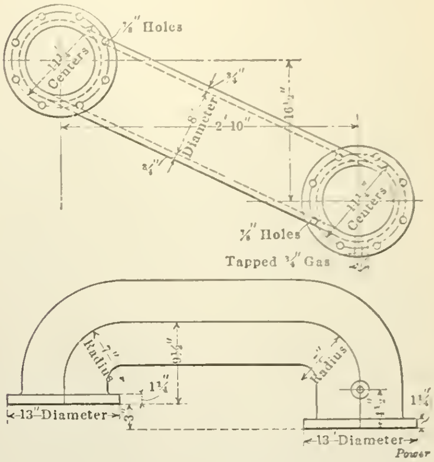


Fig. 20. Receiver

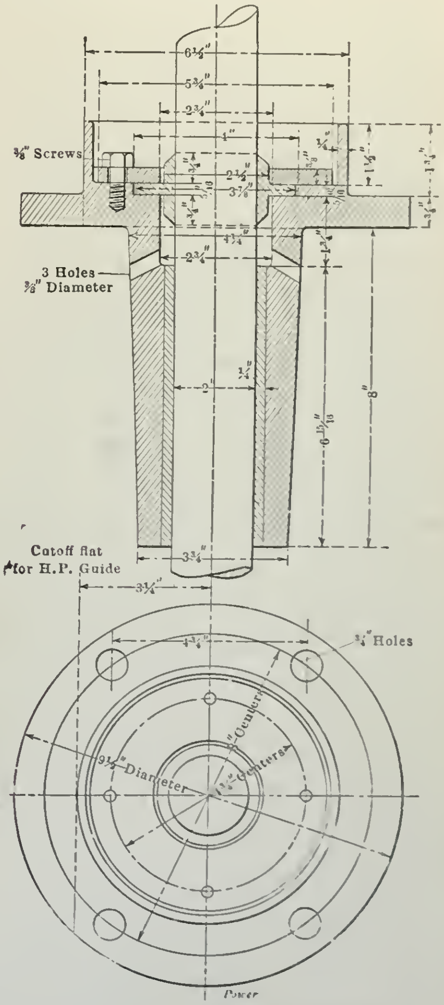


Fig. 21. Valve-spindle Crosshead Guide

rods. A detail of this gland is shown in Fig. 18. This gland is very necessary on the low-pressure side; otherwise the crank-chamber oil would be drawn by the vacuum into the cylinder, through the stuffing box. To prevent this as much as possible, it is usual to make the distance between the low-pressure stuffing-box gland and the scraper gland at the top of the frame at least one inch greater than the stroke. By this means, the portion of the piston rod working in the cylinder never enters the scraper gland. Doors of ample size are cast in the distance piece to give access to the two glands just referred to.

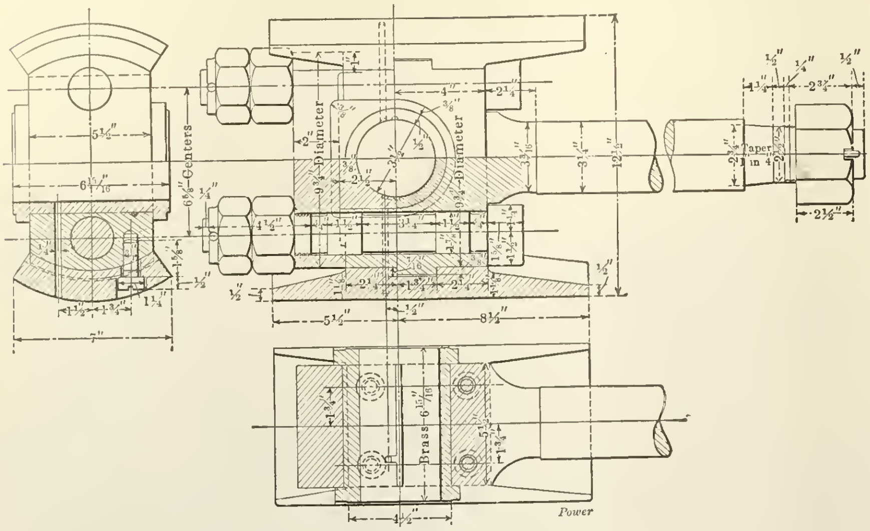


Fig. 22. Piston Rod and Crosshead

In Fig. 17 is shown the low-pressure crosshead guide, distance piece and bottom-cover casting. In this case the exhaust branch is a separate casting, and is shown in detail in Fig. 19. Owing to the larger diameter of cylinder, this arrangement is rendered possible by making the cover portion slightly less in diameter than the bore of the cylinder. Soft-packing stuffing boxes are generally used for both the low-pressure piston rod and the valve spindle.

The receiver between the cylinders is detailed in Fig. 20. This consists of a plain pipe making as short and as straight a connection between the cylinders as possible. The pipe is lagged with nonconducting material, and is covered with planished-steel sheets. All pockets where it would be possible for condensed steam to lodge are avoided and the only place where water could collect between the cylinders—the bottom bend in the receiver pipe—is drained, usually being connected to a steam trap. Reheaters for receivers have been found useless and are never fitted.

In Fig. 21 is shown the valve-spindle crosshead guide. This consists of a cast-iron bracket fitted with a brass bush. As the spindle is lubricated by oil under pressure, two holes are drilled just above the bush to allow the oil, as it escapes past the spindle, to flow back again into the engine base. A brass bush which acts as a scraper is fitted at the top of the guide to prevent oil passing up the spindle, and at the same time to prevent water from the gland getting into the crank chamber. The valve-spindle crosshead guides are alike for both high- and low-pressure cylinders.

Piston Rod and Crosshead

Fig. 22 shows in detail the piston rods and crossheads. These are usually forged in one piece, for engines up to 500 horsepower, but for larger sizes the rod is made separate from the crosshead, and a double-ended connecting rod adopted.

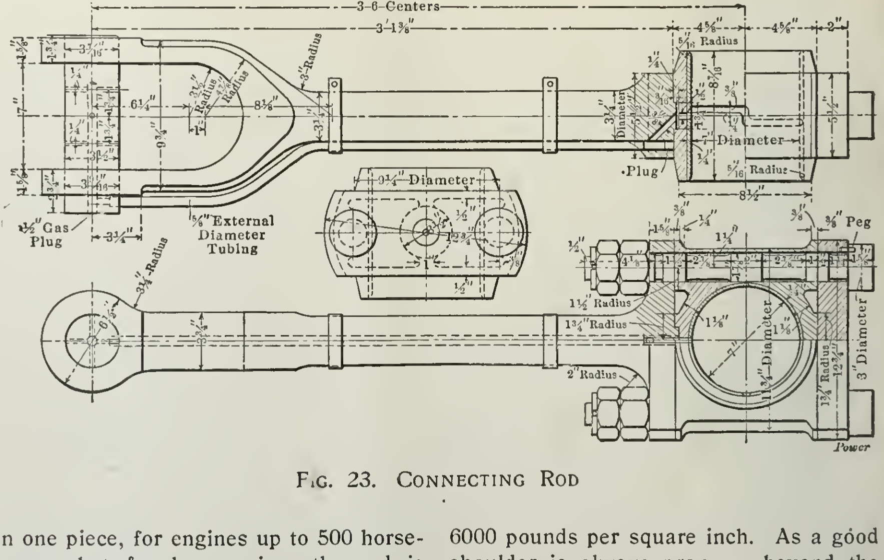


Fig. 23. Connecting Rod

The piston rod and crosshead are made of Siemens Martin acid steel and the slippers of cast iron. The latter are secured to the crosshead by cheese-headed screws made of delta metal or brass; the latter metal saves the guides in case a screw should come loose. The usual method of locking these screws is to cut out a small notch in the slipper near the head, and afterward rivet over the screw head into this notch. As adjustment is rarely necessary during the life of the engine, owing to the efficient system of lubrication, this method of locking the screws answers very well. To prevent any shearing strain coming on the screws, a feather is formed on the crosshead, which fits into a recess in the slippers.

The bushes are usually made of gun metal or phosphor bronze, and adjustment is provided by means of a cap and bolts. It will be seen that a lip is provided at each end of the cap plate to butt against the crosshead. This has been found necessary in order to prevent distortion of the crosshead when tightening up the bolts. Liners are rarely fitted to the bushes, and the nuts are usually hammered home solid, to prevent the possibility of their slacking back due to vibration when running. In addition, it is usual to fit an ordinary nut and a locknut, the latter being of the "castle" type.

The crosshead slippers are not quite central with the pin, this departure from ordinary practice being necessary to make the crosshead nuts accessible with a spanner. For attachment to the piston, a conical end is provided, in addition a good shoulder, and the nut is secured by a substantial cotter. The stress allowed at the bottom of the thread on the piston nut rarely exceeds 7000 pounds per square inch, and on the crosshead bolts 6000 pounds per square inch. As a good shoulder is always prov beyond the cone, to prevent the piston being jammed on the rod, the stress on the body of the rod usually does not exceed 3200 pounds per square inch, and as the rods are short, owing to the shortness of the stroke, they are very stiff.

The crosshead slippers are made of ample proportions, a pressure of 40 pounds per square inch being rarely exceeded. The same remark applies to the crosshead pins on which a pressure of

900 to 1000 pounds is the standard practice. The crosshead is made very compact, with a view to keeping the diameter of the guide small, and reducing the weight to a minimum. A very strong and solid crosshead is the result.

CONNECTING RODS

The connecting rods are detailed in Fig. 23. These are made of Siemens Martin acid steel and are of the ordinary "marine" type. The crosshead pins are made of low moor iron or mild steel low in carbon, having a tensile strength of from 48,000 to 56,000 pounds per square inch, and case-hardened and ground. The most usual method of securing the pins to the rods is by shrinking, but a few makers force them in by hydraulic pressure. Shrinking is preferable, however, as it makes a tighter job owing to the small section of metal in the jaw. The crank-pin bushes are usually made of brass, and lined with antifriction metal. For large engines, cast-steel bushes are often used, and a few makers use cast steel for all sizes. The method of locking the bolts is the same as for the piston rod. The crank pins are made of liberal proportions, the pressure per square inch not exceeding 400 pounds under ordinary working conditions.

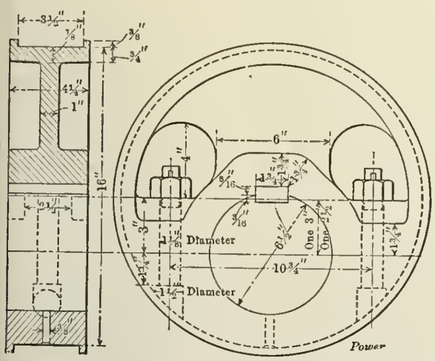


FIG. 24. ECCENTRI

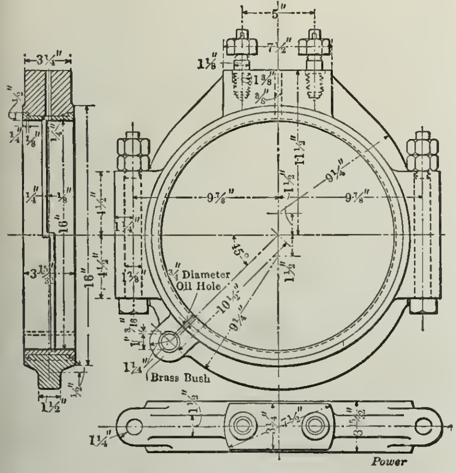
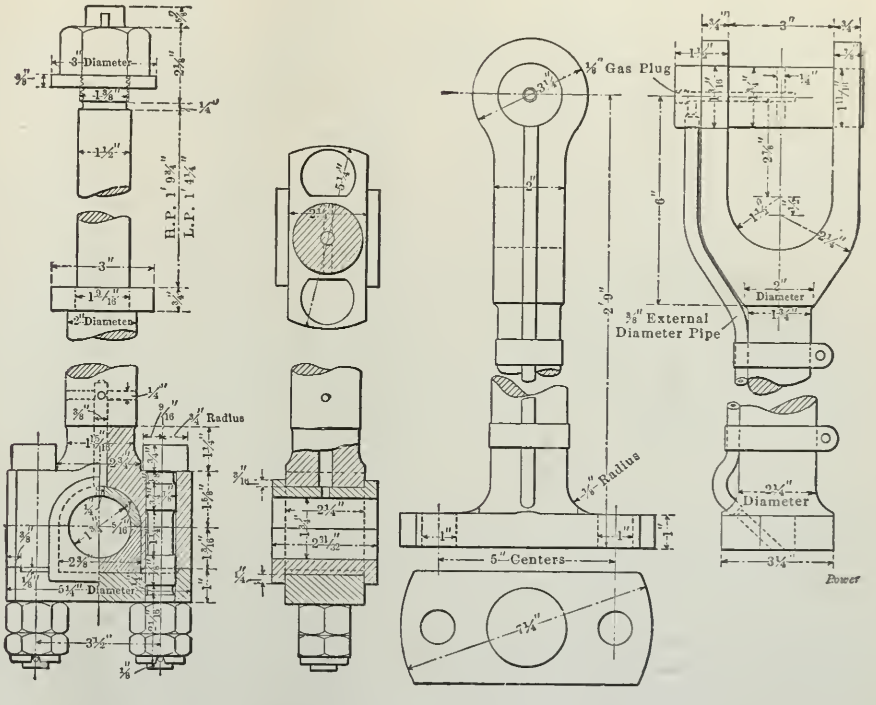


FIG. 25. ECCENTRIC CLIPS



FIGS. 26 AND 27. VALVE SPINDLE AND CONNECTING RODS

The crank pin is lubricated by a hole drilled through the crank shaft, which connects with a groove in the main bearings, and a similar groove is cut around the crank-pin brasses. The groove is staggered so as to cause equal wear over the crank pin. To lubricate the crosshead pin, a pipe is led up the side of the rod as shown, and soldered into the rod and pin at the respective ends. In addition, two clips are fitted round the pipe, to prevent it from vibrating and working loose at the joints. It will be noticed that the crosshead slides are lubricated from the pin through a central hole, which communicates with a groove in the crosshead bushes; and by means of an additional hole through the crosshead and slipper, the oil is led onto the guides.

VALVE GEAR

The valve gear illustrated in Figs. 24 to 27 is of the simple eccentric type, but of generous proportions. The eccentrics shown in Fig. 24 are of cast iron, in halves and bolted together, no cotters being used throughout the engine except as a locking device for nuts. They are made very wide to provide a large bearing surface, and are lubricated by oil under pressure from the nearest main bearing, by a hole drilled through the crank shaft. The eccentric clips, Fig. 25, are also made of cast iron, and lined with antifriction metal. An oil groove is cut inside the eccentric clip to distribute the oil around the eccentric, this groove, which is of the staggered form, being always in communication with the oil hole in the eccentric. For locking purposes, the eccentric-clip bolts are provided with lock nuts and split pins, while the studs for securing the eccentric rod to the clips are prevented from slacking by means of a collar.

The valve spindle and eccentric rods are shown in Figs. 26 and 27, respectively. These are of stiff proportions and may appear to some to be excessively strong. It may also be thought that such heavy parts are liable to cause trouble by vibration or by their inertia effect, but on the contrary they do not give any trouble, and weak gears are liable to give out and break down. The valve is securely gripped between the nut and washer on the spindle. Below the washer, the spindle is increased to 2 inches diameter and is thus large enough to form a guide. The crosshead end is of precisely the same construction as the piston rods, being fitted with gun-metal or phosphor-bronze bushes, made adjustable by a steel cap and two bolts.

The eccentric rod, Fig. 27, is fitted with a case-hardened pin at the crosshead end, and the lubrication is arranged in a similar manner to the main connecting rod. The crosshead guide, however, is lubricated by a hole drilled up the spindle from the crosshead end, two small holes drilled from the outside leading the oil onto the guide.

The valve spindles are usually made of material similar to the piston rods, and the eccentric rods are made of material similar to the connecting rods. The reason why a different steel is used for the piston and valve rods is to provide a harder material for working through the stuffing boxes, etc. If soft steel were used, it would wear rapidly.

Noiseless Circulation of Water

By William Kavanagh

> *In many hotels and apartment buildings the noise of the water pump is extremely annoying. The pneumatic system of circulation illustrated herewith has been used successfully to eliminate the noise.*

The circulation of water in large hotels and apartment buildings is receiving considerable attention from engineers and others connected with their erection and equipment.

In the old method a tank was placed on the roof of the building, the water being pumped up to it, from which it circulated by gravity to various points of outlet. Under this system an accidental lowering of the water in the roof tank below the outlet pipe would cut off the circulation of water to every point.

Air Circulating System

The system illustrated herewith may be found in operation in some first-class houses. With it, it is almost impossible to be without at least some water, in some part of the building. Under the old system of water circulation the noise of the pump destroyed the comfort of those in the building. Under the pneumatic system of water circulation the noise of the pump is not transmitted along the water lines.

The illustration shows the entire plant in duplicate for a building requiring the noiseless circulation of water. The erection and operation will be easily understood by noting the illustration. The two tanks *A A* are of a size sufficient to supply a building of 300 rooms. In this case each tank is 10 feet high and 5 feet in diameter, built out of boiler iron and made to withstand whatever pressure is necessary to force the water to the highest point. The pumps *P P* take filtered water from the filtering tanks *F F F*, and discharge it into the pressure tanks *A A* through the connection *D*. Pipes *C C* extend down to within 2 feet of the bottom of the tanks.

The Westinghouse air compressors, shown at *E E*, are connected to each tank, the governors of the pumps and compressors being controlled similarly to the method used with hydraulic elevators. Pressure gages are attached to each tank, as shown at *G*.

Operation

The system being without water or pressure, the pump governors will be wide open and the pumps, when started, will fill the tanks to a hight indicated by the dotted line at *M*. When the air compressors are started, the air pressure in the tanks *T T* will force the water up through the pipes *C C* to the manifold *H*, to which are connected the various pipe lines throughout the building.

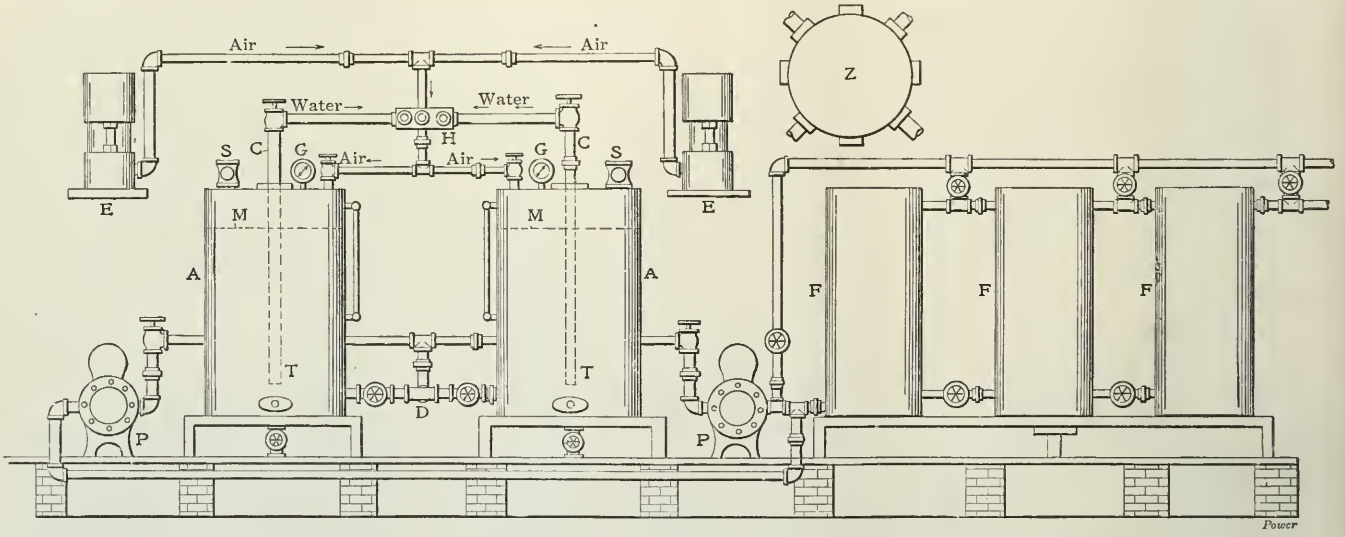
The amount of air pressure on the tanks is determined from the hight of the buildings, plus the frictional resistance offered by the water in the pipes; the governors are set slightly above this pressure, which keeps the pumps in motion and free from condensation, thus enabling the pumps to start the moment water is drawn in any part of the house.

Should both pumps fail, the air compressors would continue to circulate the water until the water had lowered sufficiently to expose the ends of the pipes *C C*. Should the air compressors fail, the pumps would continue to force filtered water through the system, but accompanied by the old objectionable noises. As the tanks, etc., are in duplicate there is but little danger of a failure. This duplicate arrangement enables the pumps and air compressors to be packed and repaired; the tanks can also be cleaned without disturbing the water circulation in the building.

The plan view *Z* is of the header *H*, showing how independent lines can be taken from it. Each line is fitted with a valve and can be shut off and repaired independently of any other line. Manholes are provided in each tank to enable them to be cleaned out, and a by-pass water connection is usually made around the filters, to be used whenever the filters need cleaning.

In some plants the filters are not by-passed, but a large open or closed tank, capable of storing one or more days' supply of filtered water, is employed, and, instead of the pumps drawing water from the filters direct, they are supplied from the storage tank. This is undoubtedly the best plan, because it affords a slow filtration of the water, and the extra supply is a very commendable feature, since the mains supplying water to the building might fail. In fact, a large number of important buildings are now equipped in this manner, a supply of water being retained in tanks, which, in case of emergency, can be used when the supply tank is used in conjunction with the filters. They are elevated above the storage tank, a float being used to determine the amount of storage.

The safety valves *S S* are attached to each pressure tank, and should an overpressure occur, the safeties will discharge the surplus water back into the storage tank; thus the system becomes almost automatic in its mode of operation.



General Arrangement of System

Electrical Department

Primer of Electricity

By Cecil P. Poole

Dynamo Armature Cores

The diagrams of armature windings given in previous lessons served to illustrate the principles of arranging and connecting the conductors in armature windings. They were not intended to show the way in which the conductors are placed, mechanically, on the armature core.

In all early dynamos, the armature cores were made up of smooth disks or rings, as in Figs. 60 and 61, and the wire was wound on the surface of the core, as indicated by the heavy black lines, except that there were many turns of wire in each coil and the entire surface of the core was covered with wires. Consequently, cross-sections of these cores would look about like Figs. 62 and 63.

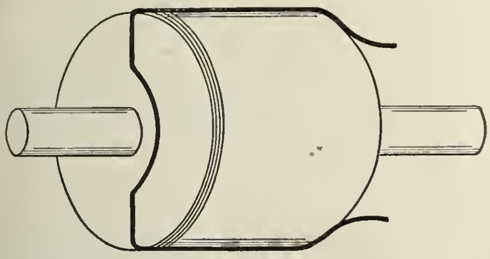


Fig. 60

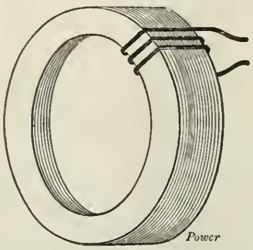


Fig. 61

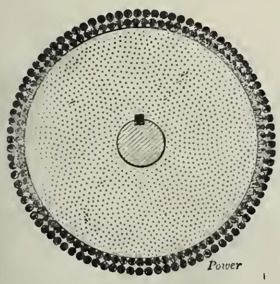


Fig. 62

The ring-shaped core was abandoned a good many years ago partly because a much greater quantity of wire is required to generate the same electromotive force than is required on a drum armature, but chiefly because of the great difficulty of putting the wire on the core. Fig. 61 shows that the wire must be threaded through the interior of the ring as many times as there are convolutions in the winding. There are other disadvantages in the ring winding, but the two mentioned were ample to put this type of winding out of use. The all-surface or drum type is the only one made today, and for this reason only that type will be referred to throughout the remainder of these lessons.

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Torque

A serious disadvantage of the surface-wound drum armature is the tendency of the winding to slip bodily around the core. This is due to the magnetic pull on the conductors. When a wire is moved across a magnetic field to induce an electromotive force in it, a definite force must be applied to it in order to move it through the flux, if the ends are connected together so that a current can flow. If this were not so, electrical energy could be generated with practically no application of mechanical energy, which would violate one of the fundamental and absolutely rigid laws of physics.

Now, if an effort is required to move a wire through a magnetic field in developing electrical energy, then the magnetism must pull on the wire, and that is exactly what happens. Therefore, when an armature is revolved between magnet poles, if the circuit connected to the winding is closed, there will be a backward mechanical drag on each wire which is directly proportional to the current in the wire and the flux in the magnetic field. The approximate relation between the different quantities is

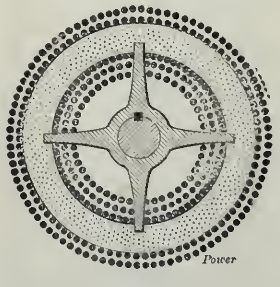


Fig. 63

$$\frac{B\,l\,i}{11{,}302{,}360} = Lb.$$

in which B represents the magnetic density of the field in lines per square inch of field cross-section; l is the length of that part of the wire which actually passes through the field; i = the current in the wire, and $Lb.$ = the pull on the wire in pounds.

The force that a complete winding has to withstand is much greater than you would be likely to think it. For example, if an armature had 300 conductors around the surface, each wire 10 inches long and carrying 100 amperes, and if the armature revolved in a field of 50,000 magnetic lines per square inch, the average pull on each wire would be

$$\frac{50{,}000 \times 10 \times 100}{11{,}302{,}360} = 4.42$$

pounds, so that the pull on the whole winding would be

$$4.42 \times 300 = 1326$$

pounds. It is easy to understand, therefore, that there would be great difficulty in holding the winding on the core of a large machine.

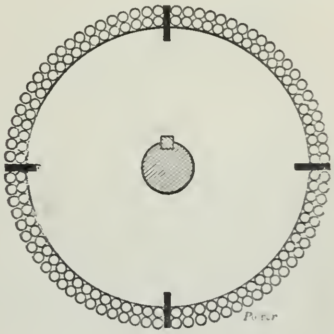


Fig. 64

To make sure that the windings of large armatures would not be dragged back around the cores, it became necessary to cut slots in the cores and set driving strips of hard wood or other strong insulating material in these slots, as represented in Fig. 64; then the winding *had* to go with the core. The winding was bound to the core surface by hard brass wires wound over strips of insulating material, as shown in Fig. 65.

Although driving strips, as shown in Fig. 64, will prevent slippage of the winding, the surface-wound armature has the disadvantage of compelling the use of a very long airgap between the polefaces and the armature core. If the winding is ½ inch thick on the surface of the core, the airgap must be slightly more than ½ inch long to prevent the polefaces from scraping the wires. This entails the use of an excessive amount of magnetizing force to drive the flux across the airgap, and a large magnetizing force requires either a large magnetizing current or a large quantity of wire in the winding which produces the magnetic flux. This will be discussed in the next lesson, when the source of the field magnetism will be taken up.

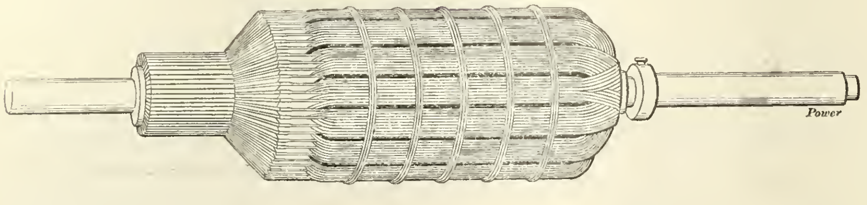


FIG. 65

THE SLOTTED CORE

Because of the disadvantages mentioned, it is the universal custom now to put the armature winding in slots cut in the core much like those in which the driving strips were set on surface-wound armatures. Fig. 66 illustrates the commonest form of slotted armature core; it represents a cross-section through the core, showing the cut-off ends of the wires in the slots.

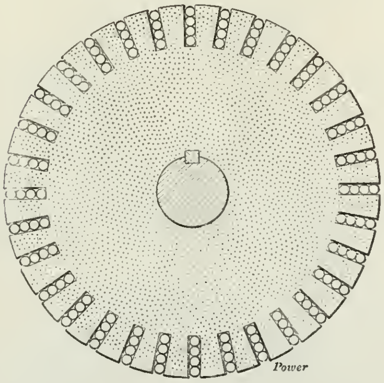


FIG. 66

This arrangement permits the airgap between the polefaces and the ends of the armature-core teeth to be reduced to the small distance necessary for mechanical clearance. It also protects the winding from accidental mechanical injury and relieves it of a great deal—nearly all—of the magnetic pull, which is almost entirely taken by the armature teeth.

It entails the slight disadvantage that the magnetic flux passing from a poleface to the armature is not smoothly distributed over the poleface, but is divided into as many concentrated "bunches" as there are armature teeth under a poleface, as represented roughly in Fig. 67.

It might seem that when the wires are sunk in slots, and the magnetic flux passes from the poleface to the armature core through the teeth, the wires will not cut the magnetic lines and will therefore not generate any electromotive force. This is, of course, not true; if it were, machines would not be built that way.

It is, furthermore, a fact that the wires generate exactly the same e.m.f. in the slots that they would on the surface of a smooth core with the same magnetic flux passing between the magnet poles and the armature core. That is, if the flux passing through the teeth to the core body of the slotted armature is the same as that passing across the airgap of a smooth-core machine, each wire in the slots will generate the same e.m.f. as each wire on the surface of the smooth-core armature, if the speeds are the same in both cases.

The reason for this is that the wires in the slots cut across the magnetic flux in the teeth, because as the armature revolves each "bunch" of flux shifts from tooth to tooth progressively. This shifting and flux cutting may be explained somewhat crudely as follows:

In Fig. 67 the group *F*, of magnetic lines, is represented as passing from the poleface edge to the armature tooth *m*, between the slots *f* and *g*. As the armature rotates, this particular "bunch" of flux shifts back to the next tooth *n*, between the slots *e* and *f*, and when that tooth reaches the position previously occupied by the tooth *m* (see Fig. 68), all of the lines that previously passed to that tooth will pass to the tooth *n*, as represented in the sketch.

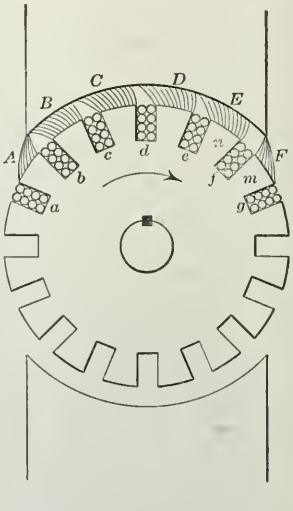


FIG. 67

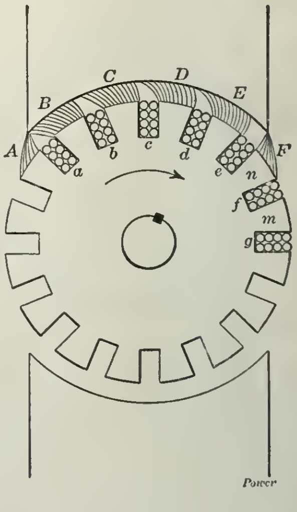


FIG. 68

In shifting from the tooth *m* to the tooth *n*, the flux *F* necessarily cuts across the wires in the slot *f* and induces an electromotive force in them. (It makes no difference whether one considers the flux as cutting across the wires or the wires as cutting across the flux. It is relative displacement of the one with respect to the other that generates the electromotive force.)

The shifting process is carried on also by the "bunches" of magnetic flux in the other core teeth. When the armature rotates exactly the distance of one slot pitch—that is, the distance from center to center of two slots next to each other—from the position shown in Fig. 67 to that of Fig. 68, the wires in the slot *a* will cut the flux *A*; those in *b* will cut the flux *B*, and so on, in all of the slots within the reach of the magnetic flux under the poleface.*

As dynamos are usually built, practically all of the magnetic flux that passes through the body of the armature core is carried from the airgap to the core through the teeth. Some of it does pass down through the coil slots but the proportion is so small that it is usually ignored.

A single glance at Fig. 66 will show that the magnetic path through the teeth is necessarily much smaller in cross-section than that through the core body. Consequently, the magnetic density in the teeth is much higher than that in the core. For this reason it is common practice, in estimating the hysteresis and eddy-current losses of an armature,† to consider the teeth and the body of the core separately.

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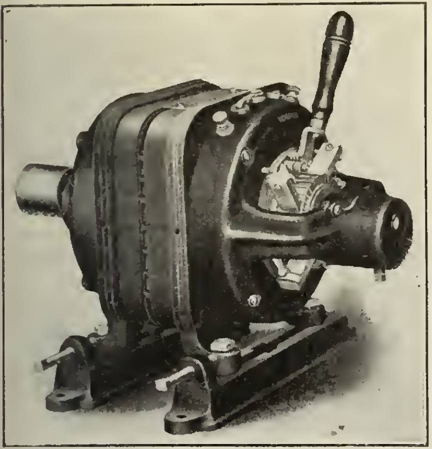
*This explanation is not scientifically accurate because it is incomplete. The actions are not quite as simple as described but the results are the same as though they were and it is impossible to give a complete exposition simply.

†See pages 714, April 19, and 804, May 3, 1910.

The Kimble Alternating Current Motor

There are few small alternating-current motors that operate satisfactorily with variable speed of wide range, or even as well as the smallest variable-speed direct-current motor. The Kimble alternating-current motor is one of the exceptions to the rule, since it operates over a wide range of speeds. In construction, the single-phase motors, which are built in sizes from ⅛ to 1½ horsepower, are four-pole machines, very similar to the ordinary series-wound direct-current motor, except that they are built with laminated field magnets. They operate at any rate of speed within a wide range without the use of either external resistors or induction coils—in fact, without any form of outside controlling device. With the exception of the ⅛ and ⅙ horsepower sizes all of the motors are reversible.



The Kimble Motor

The entire control of the motor, including starting, adjusting the speed, reversing and stopping, is vested in a single lever which merely shifts the brushes around the commutator and actuates a switch in the supply connection. When the lever is in the "off" position no current passes through the motor.

The armature is wound with wire made of a composition which is of higher resistance than copper, the object being to prevent the flow of excessive current in those coils short-circuited by the commutator brushes. The winding is said to be so proportioned that the current in the short-circuited coils cannot exceed full-load current under any possible operating conditions.

There are no "steps" in the speed gradation, because the position of the brushes on the commutator determines the speed; therefore, any rate of speed from zero to maximum may be had and the gradations are imperceptible without the use of a speed indicator. In operation the brushes always remain in contact with the commutator, the operation of the machine being almost exactly the same as that of an ordinary direct-current series-wound motor.

It is claimed that these motors start with full-load torque and that the starting current is no greater than the full-load running current. It is also stated that the motors show an unusually high power factor and good efficiency.

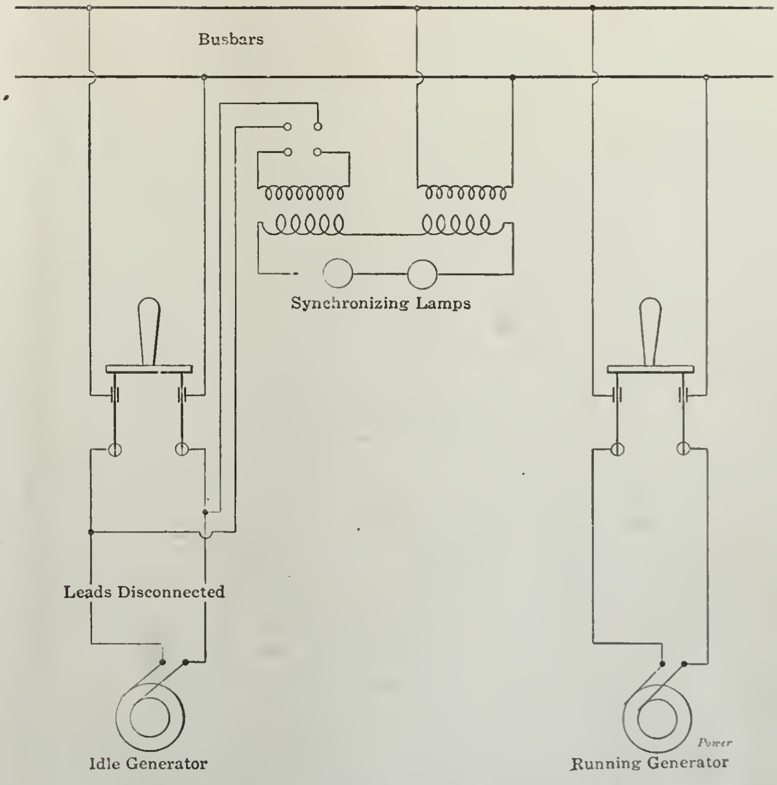
In the polyphase machines, which are built in sizes up to 7½ horsepower, the polar area is large, making possible a much lower no-load current than in the single-phase type. The polyphase motors are not designed for high starting torque, but rather for low slip; the starting torque is ample for most purposes, however. Tests of the 1-horsepower size are said to have shown 83 per cent. efficiency and 81 per cent. power factor at full load.

The tendency in the design of all the Kimble motors is toward short lengths and large diameters, which, of course, permits the use of short, stiff shafts and large bearings set close together. These machines are built by the Kimble Electric Company, Chicago.

LETTERS

Testing Synchronizing Lamps

In a certain electric-light plant there are two alternators operated in parallel. The night engineer quit suddenly and an engineer was employed who had before operated alternators in parallel. During his first shift when the load demanded the second generator, he started it and thought he had it synchronized, but as the switch went in there were "doings." He then sent for an electrician who knew that the synchronizing lamps could be connected either for light or dark synchronizing, and he spent about an hour tracing out connections to find out which way the synchronizing transformers were connected.



Connections for Testing Synchronizing Lamps

In a case of this kind all that is necessary is to raise the brushes or disconnect the leads at the terminal board of one generator and throw in its switch; then run the other generator and plug in the synchronizing lamps on the disconnected leads of the dead generator. The lamps will then indicate which is right, light or dark. The diagram illustrates the principle of the method for a single-phase plant; it applies just as well to polyphase work.

F. W. Reed.

Havre, Mont.

It is estimated that the Panama canal when finished will have cost the United States nearly $500,000,000. Yet, of our annual exports and imports amounting in value to about $4,000,000,000, less than 8 per cent. are carried in American ships. Is it not about time that we awoke to the need of rehabilitating our merchant marine?

There are water-power possibilities within a radius of 45 miles of Ottawa, Ont., which it is estimated amount to about 900,000 horsepower.

Gas Power Department

The Necessity of Thoroughly Cleansing Producer Gas

By L. L. Brewster

It has been my privilege to visit a considerable number of gas-power plants in the past few years and in almost every one I have discovered conditions which lead me to believe that altogether too little attention is given to the cleansing of the gas before delivery to the engines, more particularly in plants using producer gas.

The negligence in some cases seems to be in the builders of the producer equipment; in others, the plant attendants are at fault, and in many cases the fault is on both sides. It would seem, in view of the disastrous experience of some of the earlier plants, that both makers and operators would be thoroughly alive to the importance of clean gas; but recent observations do not show that this is so.

Almost everybody familiar with the history of the gas engine in this country will remember the troubles due to dirty gas which were experienced in the plant of the Lackawanna Steel Company at Buffalo, N. Y. Cleansing apparatus was not then developed to the efficient condition of today, but evidence that the lesson of that experience was well learned by steel makers is afforded by the elaborate and highly efficient cleansing plant at the Gary plant of the United States Steel Corporation.

The description of this plant and the results obtained, by H. J. Freyn, in a paper presented at the spring meeting of the American Society of Mechanical Engineers, should make a deep impression on any reader. In his words, "Not so many years ago it was thought in good faith that gas engines could operate on dirty gas, and it required years of costly experimenting to develop the art of gas purification to its present perfection, after the far-reaching importance of the problem was at last understood."

The overall efficiency of the wet scrubbing and secondary cleaning plants at Gary was 99.5 per cent., which reduced the dust content per cubic foot to 0.00583 grain and yet this amount meant that 3.3 tons of dust entered the engines during the year 1909.

Accurate figures showing the results of gas cleansing in producer plants are difficult to obtain, principally because accurate determinations are not regularly made and kept, but the effect on the engines speaks for itself in many plants I have seen.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Some Results of Inadequate Cleaning

I recall very well the experience at a plant near New York in which is a 300-horsepower double-acting tandem engine using gas from an anthracite suction producer. After only two months' run the handhole plates on the fuel duct leading to the inlet valves were removed and dust by the handful was brushed out. So much dust had entered and evidently settled on top of the piston rod that the wear was perceptible. Large particles of dust and much fine dust had so worked into the stuffing boxes as to fill all the grooves and put the packing rings out of business. This caused the box to leak and soon the inside of the bed forward of the crank-end cylinder was smeared with a dirty mixture of dust, oil and water. When the boxes were removed it was found necessary to drive some of the packing rings out of their grooves with cold chisels, so tightly had they been wedged about with a mixture of dust and lubricating oil.

Now, it will probably be said that the scrubbing part of this plant was entirely inadequate, but it was fully up to the then best practice both as to capacity and arrangement. The producer had an internal diameter of 6 feet 6 inches; the wet scrubber was 4 feet 6 inches in diameter and 20 feet high; the dry scrubber was built with two compartments, one above the other, and each was 8 feet in diameter and 4 feet 3 inches high.

In my opinion the spray pipes in the wet scrubber were not well designed. They were simply a pair of pipes placed at right angles and perforated for their entire length with small holes. The distribution of water in the scrubber was probably faulty.

I visited this plant recently and found many evidences that dirty gas has been flowing to the engine in the intervening three years. A set of piston rods have been replaced and the old ones show wear right up to the limit of the packing-ring travel. The surface of the worn part has the appearance of having been lapped, it is so smooth.

I believe this excessive wear was due largely to the presence of fine dust, which with the lubricating oil had the effect noted.

I found also that the suction at the fuel duct in the engine had increased from 4 to 20 inches, water pressure, and on examination found that the elbows in the gas pipe were packed so full of dust that there was only a small opening next the inner radius for the gas to pass through.

In another plant, also near New York, there are two anthracite suction producers and two three-cylinder vertical engines, controlled by governors which throttle the mixture. It is necessary to remove the throttle valve of each engine every second day and clean it to prevent it from sticking and becoming inoperative.

The producer equipment in this plant is that of a well known maker and the water pressure at the scrubbers is 30 pounds per square inch. In fact, the conditions are those asked for by the makers and the attendance is of the best. Yet these engines are expected to run on gas of the dirty character indicated, and do, but only by frequent cleaning.

In another plant which I see occassionally, a 150-horsepower horizontal double-acting engine is supplied with gas from an anthracite suction producer of equal rating. The throttle valve first installed with this engine was a ported sleeve rotating about a vertical axis. It was found soon after starting the plant that it was necessary to remove and clean this valve every day in order to prevent it from sticking. The makers of this engine have found that the piston type of throttle valve is simply put out of business by dirty gas in so many plants that they have replaced this type with one having balanced double disks which come down on two seats and have no sliding surfaces. It does not seem fair that builders of engines should be compelled to furnish engines capable of running on dirty gas, but the makers just mentioned actually abandoned a well designed mixing and governing valve because it was almost always required to handle large quantities of dust in operation.

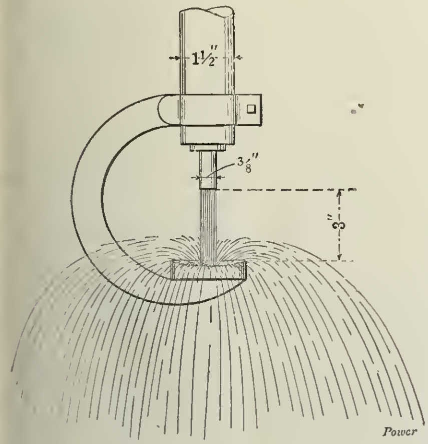
I might mention many other plants in which dirty gas is a constant source of trouble and cost, but enough has been said, probably, to show that such conditions exist, and I believe they exist in a majority of the producer plants now in operation.

Rotary Scrubbers Necessary

It is a really difficult matter to thoroughly cleanse producer gas. A large part of the dust is taken out in the wet and dry scrubbers, especially if the latter is partly filled with oily waste, but a certain amount of fine flocculent dust seems to pass through both scrubbers and find its way to the engine. This is true more often when a white-ash coal is used in the producer.

Mechanical scrubbers seem to be the only ones that will remove the dust to a sufficient degree and these are seldom installed in the producer plant. It is the rotary washers that are the salvation of the Gary plant, although a most efficient "static" cleaning plant is in use.

The addition of some form of mechanical scrubber would increase the first cost of a producer plant and the cost of operation, and thereby make the selling of a plant more difficult in competition with the simpler form of plant now generally offered. It would therefore be much better if, by common consent, all builders of producer plants would provide adequate scrubbing apparatus, for in the last analysis the faulty operation must come back to them.



Suggestion for a Scrubber Spray

The theory of gas cleansing is well understood, but seems to be poorly applied. It is, briefly, that if each particle of dust be wet it will become sufficiently heavy to deposit on some surface whence it can be washed off instead of being carried along in the stream of gas. The natural process then would be either to divide the gas into a large number of small streams and pass it through a tortuous passage whose sides are wet, as in the common wet scrubber, or to pass the gas through an extremely fine water spray, or a combination of both.

A Simple Sprayer

In the ordinary wet scrubber, the water distribution is not uniform or the supply is insufficient, or both. In many cases spray nozzles become clogged, with the result that the water runs down over part of the coke only and uncleaned gas passes through the other part.

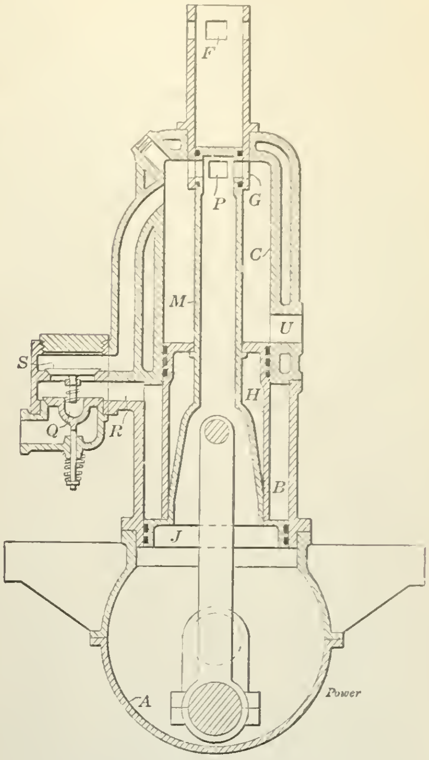
There are many forms of spray nozzle, but one which has the advantages of not becoming clogged, of distributing the water well and being easily made is shown in the accompanying sketch.

The spray pipe is reduced by a bushing to a nozzle of about 3/8 inch diameter, which should be made of the correct shape to give a clean, solid stream of water if the drop is to be considerable, but this usually is not necessary. The stream is delivered into a shallow saucer which is supported from the delivery pipe above in any convenient manner; the bracket shown is simple and easily made.

Pradeau's Two-Stroke Engine

The accompanying engraving illustrates Carlo W. Pradeau's idea of a scavenging two-stroke-cycle gas engine. The general arrangement is fairly well presented by the drawing. It includes a closed crank chamber, for the purpose of compressing air to be utilized for scavenging and a tubular extension of the piston, which works in a corresponding extension of the cylinder; this latter extension is provided with ports which communicate at one end with the explosion chamber and at the other end with the atmosphere.

The action of the engine is as follows: In the position shown in the drawing the annular chamber *B* in the lower end of the cylinder is full of explosive mixture which has been drawn in during the downstroke through the port *R* and valve *Q*. When the piston rises, the explosive mixture contained within the annular space *B* is forced out through the passage *R* and past the valve *S*, the spring controlling which is sufficiently powerful to prevent the valve from opening until the exhaust passage *U* is covered by the piston *H*, and the ports *G* and *P* are no longer in alinement. The explosive mixture is therefore forced into the top of the cylinder. The ports *P*, at the end of the upstroke, coincide with the holes *F*, allowing free ingress of air down the extension *M* to the crank chamber. The volume occupied by air at atmospheric pressure is that of the interior of the tubular extension *M*, that portion of the interior of the piston in communication with the extension, the interior of the piston extension *J*, and the crank chamber *A*. Upon the explosion the piston descends, and the air contained in the spaces just mentioned is compressed. When the top of the piston *H* passes the edge of the exhaust port *U*, the exhaust gases commence to escape through this port, and just after such opening the ports *P* in the extension *M* communicate with the ports *G*, allowing the compressed air to rush from the crank case into the cylinder *C* and drive out the burned gases. This occurs when the piston is in the position here shown. While the piston was descending, a fresh charge of mixture was being drawn in through the valve *Q* by the suction of the piston *J*, and the cycle of events is repeated.



Pradeau Two-stroke Engine

We are indebted to *The Mechanical Engineer*, Manchester, Eng., for the illustration and the foregoing particulars.

Transatlantic Steamers with Diesel Engines

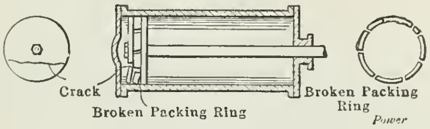
Emil L. Boas, general manager and resident director at New York of the Hamburg-American Line, has just returned from Europe full of confidence in the two freight ships which that line is building and equipping with internal-combustion engines for service between New York and Hamburg. The vessels are of 8000 tons displacement each and each will have two 1500-horsepower engines, giving them a speed of about 12½ knots. This will be the first time that this form of propulsion has been applied to vessels of such size. The engines are being built by the Augsburg & Nurnburg Company and are of the Diesel type, using the heavier grades of fuel oil. The oil can be carried in the double bottoms of the ships, doing away with coal bunkers and boiler rooms. There is thus effected great saving in space and weight, and the vessel is made more efficient as a cargo carrier. The engine-room force will be materially reduced. There will be no stokers and few oilers as the bearings will be lubricated automatically, the oil being filtered and used continuously. It is expected that the vessels will be ready in 1912, and if successful, will be followed by a passenger ship, propelled by the same method.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

A Peculiar Piston Break

Early this spring I took a position as operator on an Austin excavator. The engine was of the double-crank type with cylinders 8x12 inches in size. A pinion on the crank shaft acted directly on the masterwheels of two drums on the machine. The soil was very hard, and the builder's expert had been with the machine for eight months because he could not secure an operator who could handle it successfully on account of the hard soil. The operator who came from the shops with him used the engine very roughly. The piston rod screwed into the crosshead and was secured by a lock nut on the rod against the crosshead. This nut would come loose and the piston rod would unscrew from the crosshead. Time was never taken to adjust the piston rod properly in the crosshead, or to secure marks by which the proper adjustment could be insured without removing the cylinder head, but a good or bad guess was made and the lock nut tightened. If the engine turned over without the piston striking either end of the cylinder, all was considered well enough and the engine was run in this manner.



Sectional View Showing Break in Piston

When I took charge the expert explained that none of the former operators were "there with the goods," that he was not there for his health, etc., and that if I did not in a few days show my ability to run the machine, I would be requested to leave the job.

I made up my mind to give the machine a thorough inspection so as to be sure that everything was in good order, before I tried to make any showing. To my surprise, however, the expert stopped me and said that the machine was in good order, and to get a move on and start the machine. I objected, saying that any good engineer always acquainted himself with his machine before he attempted any operating. He insisted that he was somewhat of a mechanic himself and that he had the running of things and did not care to take any objections from me. I was to take his word for the condition of the machine. That, of course, took the responsibility from me. I knew where my levers and the throttle were, and I started.

After a short run the piston unscrewed from the crosshead again, and when I proposed getting the proper clearance and a few marks to guide myself by should the piston unscrew again, the expert told me to "just tighten her up and let her go at that." I did. The piston rod held for a week or so, and things moved nicely. All of a sudden, the engine gave a hard jolt and nearly stopped. I shut the steam off immediately. When I tried to turn the engine over by hand I found that it would not pass the center. I took the cylinder head off and found that the piston was broken as shown in the figure. The packing ring was broken into a large number of pieces of from ½ to 3 inches in length. The rod had unscrewed from the crosshead enough to allow the packing groove in the piston to pass well over the counterbore of the cylinder. One of the small pieces of the broken packing ring dropped into the counterbored part of the cylinder and, of course, something had to go when the piston started forward. The back wall of the piston gave way as shown in the figure.

Louis T. Watry.

Pueblo, Colo.

Practical Packing Record

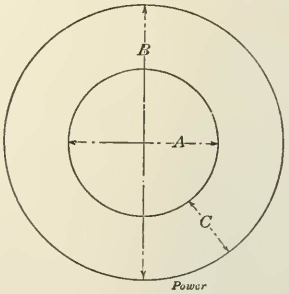
I have found the accompanying form of record very useful in operation, as well as convenient in ordering packing for the engine room.

PACKING RECORD

	Packing for	A	B.	C.	D.	Remarks.
No. 1	Corliss engines, etc.					
No. 2	Feed pump					

A is the diameter of the rod.
B is the diameter of the box
C is the size of the packing.
D is the depth of the box.

It is tacked on the inside of the packing cupboard door and the cupboard is divided into compartments corresponding to the headings on the record. Thus much time is saved by not having to measure the packing and glands each time any new packing is installed.



Key to Measurements

Edward T. Binns.

Philadelphia, Penn.

A Steam Heating Kink

Most engineers have experienced trouble in heating a certain part of the building when the steam would not circulate where it was required. When such trouble is encountered it is well to trace out the return pipe when the system is in use. It will probably be found that the pipe becomes cold at a certain point. This indicates an air pocket.

With a breast drill and an 8-inch bit bore a small hole at the top part of the pipe. This allows the air to escape and the return water goes on its course. In some cases it may be advisable to screw a ¼-inch petcock in the pipe so that it can be closed when the air has escaped.

William Watt.

Lambton Mills, Ont.

AVERAGE OF THE ANALYSES OF SEVERAL SAMPLES OF FUEL
COMPARED WITH THE ANALYSIS OF A COMPOSITE SAMPLE MADE UP OF THESE SAMPLES.

Fuel	Fuel as Fired															Dry Fuel.		
	Fixed Carbon Per cent.		Volatile Per Cent.		Moisture Per cent.		Moisture in Composite differing from Average		Ash Per cent.		Sulphur Per cent.		B. T. U.			B. T. U.		
	Average	Composite Sample	Average	Composite Sample	Average	Composite Sample	Per cent. of Total	Per cent. of Average Moisture	Average	Composite Sample	Average	Composite Sample	Average	Composite Sample	Difference	Average	Composite Sample	Difference
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Anthracite	78.55	77.68	5.77	7.07	3.93	3.47	0.46	11.65	11.75	11.78	1.17	1.61	12,682	12,754	+ 72	13,201	13,212	+ 11
Pocahontas	73.33	73.43	19.49	19.50	2.01	1.90	0.11	5.47	5.17	5.17	0.98	1.00	14,753	14,754	+ 1	15,056	15,040	— 16
Byproduct coke	80.68	81.74	3.34	2.74	5.92	5.73	0.19	3.21	10.06	9.79	1.15	1.41	12,049	12,071	+ 22	12,807	12,805	— 2
Solvay process coke	85.80	85.76	2.14	2.45	1.60	1.51	0.09	5.63	10.46	10.28	0.65	0.65	12,504	12,490	— 14	12,707	12,681	—126
Western bituminous No. 1	42.52	42.12	39.48	40.48	9.80	8.93	0.87	8.88	8.19	8.47	3.82	3.84	11,675	11,781	+106	12,943	12,936	— 7
Western bituminous No. 2	42.53	43.74	41.38	41.49	11.14	10.09	1.05	9.42	4.96	4.68	2.39	2.37	11,990	12,139	+149	13,493	12,356	—137
Western bituminous No. 3	40.59	40.63	37.58	37.55	11.02	11.07	0.05	0.45	10.81	10.75	3.00	3.00	11,115	11,109	— 6	12,492	12,492	0
Western bituminous No. 4	41.15	41.51	37.36	38.01	10.10	8.14	1.96	19.41	11.39	12.34	2.92	3.00	11,155	11,209	+ 54	12,422	12,202	—220
Western bituminous No. 5	40.36	40.35	39.53	39.57	11.18	11.14	0.04	0.36	8.93	8.94	2.21	2.48	11,548	11,383	—165	13,002	12,810	—192
Western bituminous No. 6	40.85	41.45	36.67	36.62	10.98	10.86	0.12	1.09	11.51	11.07	3.75	3.71	10,943	10,998	+ 55	12,293	12,338	+ 45
Western bituminous No. 7	40.84	40.56	37.56	37.67	10.24	10.01	0.23	2.25	11.36	11.76	3.19	3.56	11,005	11,033	+ 28	12,260	12,260	0
Western bituminous No. 8	40.30	40.36	35.42	35.31	11.43	11.30	0.13	1.14	12.85	13.03	5.93	6.66	10,443	10,405	— 38	11,791	11,731	— 60
Western bituminous No. 9	40.06	40.48	40.22	40.37	7.94	7.65	0.29	3.65	11.78	11.50	4.58	4.55	11,331	11,412	+ 81	12,308	12,357	+ 49
Western bituminous No. 10	48.11	48.72	33.31	33.33	7.60	7.44	0.16	2.11	10.98	10.26	1.40	1.17	11,783	11,784	+ 1	12,752	12,731	— 21
Western bituminous No. 11	48.07	48.42	36.27	36.28	6.55	6.66	0.11	1.68	7.27	8.64	2.09	2.11	12,274	12,233	— 41	13,134	13,106	— 28
Western bituminous No. 12	50.85	51.00	34.93	35.02	6.57	6.20	0.37	5.63	7.65	7.78	2.00	2.22	12,642	12,602	— 40	13,531	13,435	— 96

Comparison of Coal Analyses

I recently had occasion to run a series of boiler tests in all of which the same kind of coal was used and the question arose as to the method of preparing a sample of the coal for the chemist. The choice lay between three methods: To follow the usual custom of taking a sample during each test, have it analyzed and use the results in working up the test; to average these analyses and use the resulting values in computing all the tests; or to take the usual samples, and after air-drying, mix them in the proportion of the weights of coal fired during the respective tests, then have this composite sample analyzed and use the results in computing all of the tests.

To obtain some idea of the results that might be expected from these last two methods, a study was made of analyses already available, and the results are shown in the accompanying table. This gives analyses of sixteen different fuels, and the average results are taken from six to twelve different analyses. A composite sample had already been made and analyzed for each of these sixteen fuels, and the two values for each item of the analysis are given in the table.

The analysis of the separate samples in several instances varied considerably, probably due to the manner of taking the sample rather than a difference in the coal. This may, in a measure, account for some of the uneven results obtained in working up the tests, for when they were computed by using the average values the results were very uniform.

It will be noticed from the table that with the exception of two cases the samples had lost some moisture before the composite sample was made, although this difference was small in most cases. The greatest variation in the fixed carbon among the bituminous coals was about 1¼ per cent., although in most cases it was less than ½ per cent.

The B.t.u. are given both for "fuel as fired" and for "dry fuel," and a study of the differences as shown by columns 15 and 18 is of considerable interest. In the case of fuel as fired, Western bituminous coal No. 2 showed 149 B.t.u. more per pound from the composite sample, while coal No. 5 had 165 B.t.u. less. The average of all the positive differences is 57 B.t.u. and of the negative differences 51 B.t.u. When considered on a dry fuel basis, two of the coals showed the same heat value. three a very slight increase in the composite sample and in eleven cases the heat value was less according to the composite sample. This difference reached as high as 220 B.t.u. with an average of 82 B.t.u. for the eleven samples. In the case of coals Nos. 2 and 4 this difference changed from positive to negative, a total of almost 300 B.t.u.

In view of these irregularities it is decidedly questionable as to whether it would be advisable to use the values obtained from the composite sample in computing a series of tests.

FRANK L. BUSEY.

Urbana, Ill.

Causes of Failures

It is not an unusual occurrence for an engine or a boiler to break down through causes totally unknown to the engineer in charge, and it is only by knowing how engines have failed, by carefully noting symptoms of a possible failure, and by applying the remedy before the breakdown occurs, that a man can qualify as a competent engineer. The skilled engineer's eye is trained and is quick to detect the slightest indication of trouble; also his ears are tuned to the hum of the machinery and should any unusual noise, however slight, break in upon the regular pulsations of the engine, he will at once notice it and will be able to satisfy himself as to the cause. This skill is due to mental training in conjunction with the practical experience gained through manual labor, which unfortunately is none too often appreciated by employers.

On the other hand, the unskilled engineer will run an engine month after month without inspection or overhauling, until finally a shaft breaks in consequence of a bearing being in need of adjustment, or a disregarded leaky rivet hole allows corrosion to set in; or he will overload the safety valves of the boilers by moving the weight so as to make the boiler do more work at the risk of life and property. It is a fact that there are men in charge of engines who know practically nothing beyond how to shovel coal; yet they are not to blame for getting into the engine room, changing the shovel for the starting lever.

One cannot view with complacency the startling number of boiler explosions that occur annually in the United States, especially when it is remembered that a large part of these are avoidable, some being due to culpable negligence. The primary cause of most boiler explosions may be traced to corrosion of the plates, either as a uniform wasting of the surface, or as rapid local decline, rendering the plates too weak to withstand the ordinary working pressures. The idea of high pressure is only a relative one, and can have reference only to the comparative strength of the boiler upon which the pressure is exerted; one boiler may be as safe when carrying 200 pounds pressure as another carrying 50 pounds. Of all causes of failure, there is none so deceptive and so certain to reduce the strength of the plates as corrosion, both internal and external, especially the latter, when the sweating of the plates underneath, in conjunction with the soot and moisture, is not suspected. This dangerous condition usually does not attract the attention of the unskilled engineer, hence it does its work unarrested until ultimately the inevitable happens.

I have been engaged for over 25 years in the construction and operation of engines and boilers, and have had many opportunities of observing the causes of failure and breakdowns under the care of unskilled engineers; hence it may be of interest to cite a few cases.

In one case an engineer broke the beam of an engine into halves by allowing the injection water in the condenser to enter the cylinder; the water, being incompressible, resisted the down stroke of the piston, and the momentum of the flywheel resisted the upward tendency of the water, consequently the weakest point between the crank shaft and the condenser, which proved to be the center of the beam, parted and came down with a crash. This accident was caused soon after starting by not regulating the supply of water to the condenser so that the air pump could discharge it into the hotwell.

In another case a beam was broken by allowing blocks of wood to lie about the engine room until some were accidentally knocked into the flywheel pit, where they caught between the rim of the wheel and the masonry, forming a brake which acted against the steam in the cylinders.

An engineer allowed the lid of an oil box to fit air tight, consequently the siphon would not act properly. He retrimmed it and still the bearing continued to become hot. As his efforts were of no avail the journal and the brasses were badly cut, because the mill had to be kept running.

In another case feed water was taken from a brook and some chips were drawn through the suction pipe and into the pump, getting under the valve and disabling it. As the boiler became short of water the engine had to be stopped, and the top check valve being up, the second check could not be examined, so steam had to be blown from the boiler and the works stopped, all because of the absence of a strainer in front of the intake, and by the want of a "head" that could conceive of the chips floating in the brook.

A boiler was washed out and filled with water again and the next morning the fire was started as usual. When the man in charge began to think it time for steam to appear, he perceived something smoking on top of the boiler and upon investigation found it to be an old bag. Shortly afterward he found the flue to be red hot. It was found that the water had passed out through the blowoff cock which had not been closed. All blowoff cocks should be carefully watched, because they may fail at any time, especially when least expected, through ordinary wear and tear.

Enough has been said to establish the fact that a large percentage of failures and breakdowns are avoidable, if competent and careful engineers are in charge. When a man's mind possesses no scientific principles for his guidance, every act is invested with doubt, and any accident which may be caused through ignorance is generally surrounded with mystery.

JOHN GREEN.

Seattle, Wash.

Graphic Method of Determining Rope Sag

In the article under the above heading in the August 16 issue of POWER, the question of the sag to be allowed in rope drives is touched upon, but only an approximate rule for determining this is given. Since writing the article it has occurred to me that it would be well to supplement it with an excellent graphic method.

In getting out details of a rope drive it is necessary, for structural reasons, to anticipate fairly closely the amount to which the slack side will sag. It may, for one reason or another, be necessary to place a girder or pipe in such a position that it passes at right angles to and between the two sides of the drive, and where the lower side is the driving side there is some risk of the upper ropes coming in contact with the girder when sagging under load. Further, when the top is the driving side the floor must be kept sufficiently low to accommodate the sag of the slack side. It is not an easy matter to say offhand what the sag will be with ropes running under normal conditions, and for this reason the following graphic method will be found very useful.

The sketch illustrates the method of procedure both when the sag is at the top and at the bottom. Considering the first case, which is shown in the upper diagram, the drive is laid out to scale and two arcs are described, having for their centers the centers of the pulleys and for their radii the distance between the pulley centers and the inside periphery of the opposite pulley at the center line. These arcs intersect at the top of the diagram, and the next step is to draw a line radially from the centers of each pulley toward this point of intersection. These lines touch the peripheries of the pulleys at certain points as *A* and *B*, and if a third arc be described, having for its center the intersection of the first two arcs and for its radius the distance between this point and either *A* or *B*, it will show correctly the maximum amount of sag to be allowed for. This may be scaled in feet and inches by measuring the distance between the point of extreme sag and a line running at a tangent to the two pulleys, as shown.

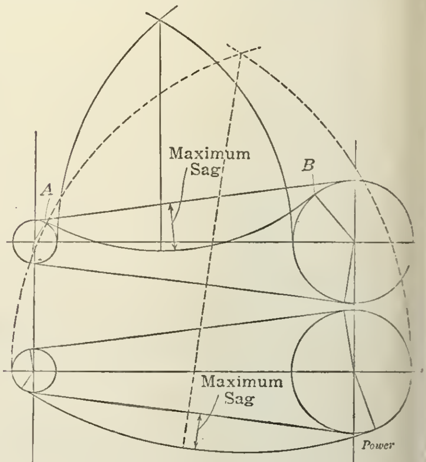


DIAGRAM SHOWING SAG

In the second case, where the slack side is at the bottom the procedure is similar to the foregoing except that the radii of the first two arcs, indicated in the lower figure by the broken lines are the distances between the pulley centers and the outside periphery of the opposite pulley at the center line.

In practice, when the sag exceeds the amount arrived at by this method it is time the ropes were taken up. In the article previously referred to, the allowable sag is given as from 8 to 10 per cent. of the distance between the pulley centers. The result of the graphic method gives a larger sag which must therefore be taken as the maximum.

FRANCIS H. DAVIES.

London, England.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

The Massachusetts License Law

As an engineer in the State of Massachusetts, I am naturally interested in all discussions dealing with its license law, and have read lately of a desire on the part of some men to make the law more strict and also to make it harder for engineers by contesting their applications for license.

In these times of close competition, when men must do more, know more and earn more than ever before, I think we should move slowly in trying to hinder a man when he is trying to better himself.

When a man is known to be unsafe, a user of intoxicants or notoriously careless, it is right that he should not be allowed to handle steam machinery. There are, however, many men, honest and conscientious in purpose, who do not have the nerve to match wits with an examiner, yet would make good and faithful workers in any plant.

I know, and I have no doubt that there are others also who know, of cases where such men have not been given a fair show, when being examined for license, and have become well nigh discouraged.

The license law was passed in order that we might have engineers and firemen who would be safe to intrust with the operation of steam boilers and machinery so that the public would be protected. When this end has been attained, it is of no consequence to the inspectors or the public how far the engineers may go into the finer points of the business, such as economy and efficiency of operation. These depend upon the men themselves and just as far as they develop their talents will they attain greater success. But I do not know that they will make any safer men than those who are not as smart mentally. If the action of the law were independent of the vagaries of human nature, we should have no complaint about one man passing the examination easier than another, when both had the same ability. But the inspectors are but men subject to the same changes of nature.

The examiners have the law on their side, and to appeal to the board would be still worse for the engineer, for three men can disconcert an applicant much more easily than one. For the examiners I will say that they are often imposed upon, and that there are some who are perfectly fair and just.

In conclusion, considering all these things and the fact that the present law is a good one and is enforced as well or better than the average, when a man has had the required experience and satisfied the examiners of his fitness for a license, he should not be hampered by any man or group of men in getting his license. As we have a department of State officials for the purpose of deciding who shall receive a license and who shall not, why not leave the question to them?

G. H. KIMBALL.

East Dedham, Mass.

Installing Globe Valves

Much has been written about the proper way in which to install globe valves. I would like to add my experienc to the rest. I received my training in the engine room of a lake steamboat, where safety is the first consideration, and have worked with pressures of from 40 up to 275 pounds. Globe valves are invariably installed with the pressure on the under side of the disk. I have never known a single valve to fail by the stripping of the threads on the stem. I have taken valves apart that had been in constant service ten months in the year for 20 years with the pressure on the under side of the disk and have seen no signs of wearing or stripping of the threads. In fact, the only trouble I have ever had was with valves where the pressure was above the disk.

As E. F. Tracy remarks, in the August 30 issue, more harm is done with the monkey wrench than by wear. I never use a wrench nor allow one to be used to close a valve. If the handwheel will not close it tight it is time to unship and grind the seat or replace the disk.

I know of valves that have been in service under 275 pounds pressure for nearly ten years in which it would be impossible to detect any wear of the threads or stem. The valves are to all appearances as good as ever.

I know of an engine which has been in operation for over 25 years, working under 90 pounds pressure. The pressure is on the under side of the throttle valve. This valve has been opened and closed an average of once a day all that time, yet the valve is steam tight and has no lost motion, showing that the threads are not worn a particle.

So far as globe valves in feed-water lines are concerned, there is room for argument. I have seen valves placed both ways with satisfactory results. But, for myself, I invariably place them with the under side toward the boiler. I never have had any trouble. I claim that a globe valve has no business on a feed-water line anyway; I prefer a good gate valve which will give a straightway opening from the pump to the boiler.

While I do not like globe valves on a blowoff line, I have never had any trouble with them excepting that due to rapid wear of the seat and disk. They are more easily clogged with scale than the gate valve or the regular blowoff cock.

In conclusion, I wish to say that I think with proper care and usage, a globe valve designed for the pressure to be taken care of, will cause very little trouble when installed with the pressure under the seat.

A. A. BLANCHARD.

Oak Harbor, O.

Colors for Station Piping

It was up to me recently to scheme up a color system for a power plant. After looking over the various articles in POWER, I made up the following, which I think has some "system" to it.

All high-pressure or supply lines are painted a solid color; that is, the fittings are the same as the pipe.

The various colors and their meanings are shown by the following table:

COLOR SCHEME FOR STATION PIPING.

		COLOR OF FITTINGS.		
Contents of Pipe.	General Color.	Supply or High Pressure.	Exhaust. Low-Pressure or Return.	Drips. Drain. Etc.
Steam..	white	white	red	brown
Water...	slate	slate	red	brown
Oil.....	dark blue	dark blue	red	
Air.....	light blue	light blue	red	brown
Fire service...	vermilion	vermilion		
Hydraulic pressure...	black	black	red	
Ammonia..	yellow	yellow	red	brown
Brine...	green	green	red	brown

As a further aid in distinguishing what the pipes contain, I had stencils cut with arrows and numbers to indicate the normal pressure and the direction of the flow. The arrows were put on every pipe where the direction of flow was not obvious from the position or location of the pipe. The pressure figures were necessary because of the fact that three different pressures of steam and two of compressed air were carried.

LOUIS TONG.

New York City.

Leaky Floats

In the August 16 issue, E. Granfield asks for information regarding leaky floats.

To my knowledge, there is no successful method of repairing a metal float other than by soldering it.

I suppose the float Mr. Granfield has reference to is a round one. If the float is made of copper, take some muriatic acid, 5 cent's worth will be sufficient, in a drinking glass. Fill the glass one-quarter full and put in some zinc, in pieces small enough to be covered by the acid.

The liquid will boil as soon as the zinc is put in. When the foam rises to the top of the glass, ignite it with a match and allow it to burn off. Do this in the evening and let the glass stand uncovered over night. In the morning the liquid will be as clear as crystal. The solder should be composed of two parts lead and one part good tin.

When ready to solder, have a small brush handy, light some charcoal in a soldering pot and put the soldering iron on to heat. Punch a hole in the float to let the water out. In 99 out of a 100 cases the float leaks at the seam. As it is hard to tell just where the leak is, it is best to melt the ball apart. While the solder is in a melting condition, wipe the joint clean with a rag. Scrape and tin all black spots. Remember, both edges of the joint must be perfectly tinned to insure a tight job. If everything is tinned nicely and no lumps of solder remain on either of the edges, put the halves together again. In putting them together take a strip of tin or copper and force it into the upper part of the seam and pull it around, forcing one part into the other. This done, heat the iron well and with a fine file rub two sides of the point until they are bright; dip the point into the acid for two seconds and then apply it to the solder which will melt immediately and tin the iron. Swab the float all around the seam. The soldering may then be commenced. Do not put too much solder on. Go all around the seam and solder smoothly. Put a small drop of solder here and there on the seam so as to prevent the float from coming apart. When the iron does not melt the solder any more, heat it again, do not try to smear the joint tight, the solder must run freely; otherwise a leak will surely develop in a short time.

After the joint is soldered smooth all around and everything appears to be tight, go over the joint again, this will make it more solid. Solder the socket on and then close up the hole which was punched to let the water out.

Before putting the float back in the tank try it. Push the float into a bucket of water so that the ball is wholly immersed for a while. If time permits, tie the float down over night. If there is no water in the float in the morning, it is safe to assume that the float is perfectly tight.

Remember, a hole only as small as the end of a pin will cause trouble. If all the rules are followed, however, the ball will be tight and as good as or better than a new one. I have repaired several floats. The last one that I repaired has been in use for the last three years. An experienced hand can do the job in about two hours.

WILLIAM L. KEIL.

Philadelphia, Penn.

Chimney Design and Wind Pressure

In the August 16 issue appears an article on chimney design by H. R. Austin. Perhaps the following information may be of service to him and others who read POWER. I believe that Mr. Austin is correct in his statements.

The relation between the mean pressure of the wind against an exposed surface and the corresponding wind velocity has never been worked out with entire satisfaction. Nevertheless, better results can be obtained by using such facts as are known than by attempting to measure the wind pressure directly.

TABLE OF WIND PRESSURES.

Indicated Velocity in Miles per Hour.		Tenths									
		0	1	2	3	4	5	6	7	8	9
		Wind pressure in pounds per square foot.									
	0						0.104	0.144	0.190	0.243	0.303
	10	0.369	0.433	0.511	0.586	0.666	0.762	0.853	0.949	1.05	1.16
	20	1.27	1.38	1.50	1.63	1.76	1.90	2.04	2.19	2.34	2.48
	30	2.64	2.81	2.98	3.14	3.32	3.50	3.67	3.87	4.04	4.24
	40	4.44	4.64	4.84	5.07	5.27	5.51	5.72	5.93	6.18	6.40
	50	6.66	6.89	7.12	7.40	7.64	7.88	8.14	8.43	8.69	8.95
	60	9.22	9.49	9.76	10.1	10.4	10.6	10.9	11.2	11.6	11.9
	70	12.2	12.5	12.8	13.1	13.5	13.8	14.1	14.4	14.8	15.1
	80	15.5	15.8	16.2	16.5	16.9	17.3	17.6	18.0	18.4	18.8
	0	19.2									

Experiments were made by the Weather Bureau at Washington to determine the relation between wind velocity and pressure for certain limited conditions by exposing squarely against the wind plates having from 4 to 9 square feet of surface. The deflection produced by the pressure in springs of known strength was continuously recorded by a pencil upon a revolving cylinder of paper. At the same time and on the same sheet of paper was electrically recorded the velocity of the wind as indicated by an anemometer exposed near the pressure plate. The accompanying table gives the results of these experiments.

H. WILKINSON.

Hartford, Conn.

The Selection of an Engine

The articles that have appeared recently under this heading have been very interesting, and it will be still more interesting if the discussion results in some definite information on the subject.

Apparently, Mr. Myers' proof that high piston speeds mean greater economy, misled his critic, and it should be taken that more steam is passed through the cylinder by increasing the number of strokes, the cutoff in the cylinder being kept constant. It would, perhaps, have been more conclusive, and a greater help, had authorities been quoted for the statement that, with two engines of the same class and design, working with the same mean effective pressure and under the same conditions, the low-speed engine would be more economical than the high-speed one. That this is correct is well known, and, no doubt, authorities could be given for it. I would be glad to know of other authorities besides Mr. Willans who have investigated this. Mr. Myers' experience is directly opposed to this, and it would be a great help toward a better knowledge of this subject if he would kindly give the exact data upon which his statements are based. There are very few engineers who are in a position to establish the truth of many of the present-day assertions, and this probably accounts for the fact that, after more than a hundred years' study of the steam engine, there are still some mysteries about this ancient prime mover.

As far as I can ascertain, Mr. Myers is the first to state that the cylinder condensation is directly proportional to the speed of rotation of the engine. Up to the present in this country it has only been suggested that the speed of rotation might affect the cylinder condensation, and there is definite information that a moderate alteration in the speed does not affect the economy of an engine.

I am able to give the following example from actual practice to support this statement, and it is possible to obtain such an example from more than one of the leading engine builders in this country.

In answer to an inquiry for an engine to work under certain conditions, a 1 and 25½ by 30-inch engine to run at 165 revolutions per minute was offered at a guaranteed steam consumption of 11.7 pounds per brake horsepower under the specified conditions. This quotation was followed up by a representative calling upon the firm inquiring for the engine. He was informed that unless the steam consumption were improved his engine would be out of the running. As the result of this information the quotation was amended by offering a 17½ and 30 by 36-inch engine to run at 100 revolutions per minute and a guaranteed steam consumption of 10.8 pounds per brake horsepower under the same conditions as before. Now, these two engines were of the same class and of exactly similar design in every respect; the 30-inch stroke engine was designed for a maximum piston speed of 825 feet per minute and the 36-inch stroke engine was designed for a maximum piston speed of 840 feet per minute. Although the latter engine was quoted at nearly 30 per cent. below the speed for which it was designed, yet the steam consumption was quoted the same per brake horsepower as it would be if the engine were run at its maximum speed under the same conditions and with the same mean effective pressure.

It is interesting to record that the amended quotation was successful in spite of the additional cost of the engine, and this tends to show that the question of economy versus capital cost is considered.

This example has no uncertain quantities in it, and the engine builders guaranteed the steam consumption under the heavy penalty of £500 per pound of steam in excess of this guarantee. It shows that it is not correct to assume that the cylinder condensation is in proportion to the speed, and it also shows that assumptions of the probable effects of certain parts of the steam-swept surface of the cylinder are valueless.

To make this discussion of any practical value, concrete facts only are required, and it would be a great advantage to readers of POWER if, instead of mere statements of experience, the actual data upon which the statements are based be given. I would point out that Mr. Myers is mistaken in claiming economy for the high-speed engine on account of its reduced clearance, for it is just on this point that the low-speed engine scores.

It would be of great assistance to engine designers if some definite information were available of the exact effect, in pounds of steam per brake horsepower, of additional clearance.

The superior economy of the low-speed engine is now most conclusively proved by our workshops, for, now that there is a due regard paid to the cost of coal, the high-speed engine maker is obliged to make the exhaust-steam turbine as well as the standard engine, and the low-speed engine maker is reaping the benefit of being able to supply an engine which easily competes with the overall economy of the high-speed engine and exhaust-turbine combination.

That the high-speed engine has had a great run in this country is not disputed, and in a way it was a natural consequence of cheap coal and a large export trade; the price of coal, however, is now receiving its proper consideration all over the world and the result is that engine builders are now selling both at home and abroad heavier, and more costly, but more economical engines.

JAMES CANNELL.

Stanford le Hope, Eng.

Trouble with Barometric Condenser

I was interested in the description of the trouble that Mr. Elston is having with his Bulkley condenser. A few years ago I worked in a plant which used Bulkley condensers and the same kind of trouble was experienced. The chief engineer was under the impression that it was caused by the tail pipes of all of the engines discharging into a common pipe which discharged into a hotwell, only one hotwell being used. From the sketch shown it is evident that Mr. Elston is having the same trouble with the condenser discharging into an independent hotwell.

I am of the opinion that the trouble is caused by producing a vacuum too rapidly; the action being as follows:

Before the vacuum is produced, the water in the tail pipe is at the same level as the water in the hotwell. When a vacuum is made very quickly, the water in the hotwell rushes up into the tail pipe and the kinetic energy is sufficient to carry it beyond the static barometric level and discharge it into the exhaust pipe. This trouble can probably be eliminated by cutting off the bottom of the tail pipe a little so that only enough water can rush into the tail pipe to fill it about two-thirds full, the remainder being obtained from the supply pipe.

CHARLES R. SUGG.

Wilmington, N. C.

Safety of Boilers

The editorial in the August 16 issue entitled, "Safety of Boilers," contains this statement:

"Overworking a boiler as regards the quantity of steam generated is just as dangerous as overpressure and possible more so, because of the insidious manner in which defects rendering a boiler unsafe are produced by this means."

Considering the number of disastrous explosions which have occurred lately, this warning to users of multitubular boilers should not be forgotten. I have been able to prove the assertion to be correct and probably an account of my observations may be interesting to other readers.

I am chief engineer of an ice factory. We have a 20- and a 60-ton ice machine. The boiler plant originally consisted of two 7x30-foot Lancashire boilers; two 5x15-foot multitubular boilers, and one 5x18-foot Cornish boiler, a miscellaneous collection, but that is another story. We used to run the 60-ton machine at 40 revolutions per minute, making 42 tons of ice per 24 hours with one Lancashire boiler and one multitubular, changing after three weeks' run to the other Lancashire and multitubular. These ran the machine without trouble due to want of steam. But when we worked a Lancashire and the Cornish together, we had trouble in keeping steam.

The earthquake of 1907 threw down the brick chimney, wrecking the boiler roof over the Lancashire boilers and the steam piping. As the boilers were old, I advised remodeling the plant and installing four 6x18-foot multitubular boilers which would occupy very little more floor space than the two Lancashire boilers although they would give much more power.

While the boilers were being imported I put up temporary steam piping to the two 5x15-foot multitubulars and the 5x18-foot Cornish boiler and ran any two at a time, making 30 tons of ice. When I ran the two multitubulars together, I got all the steam required without forcing but when one was shut off for cleaning and the Cornish boiler connected, I had to force the multitubular. (A Cornish boiler cannot be forced with natural draft because the furnace is too small). After doing this for about four months, I noticed that the girth seam over the furnace was leaking. I had it calked and tested. It did not leak when steam was up and the boiler working but only after it had been cut out and allowed several days to cool. This leaking seam bothered me for I could not find the cause. These multitubular boilers had been run by me for the previous ten years and were in good condition.

All this while I was pulling out the old Lancashire boilers and preparing a place for the new 6x18-foot multitubulars. Two came and were erected but, before they could be worked, I had to pull down one of the 5x15-foot multitubulars. We then had the two new boilers, one 5x15-foot multitubular and the Cornish boiler. I started to make 42 tons of ice as before the earthquake, using one of the new boilers and the old multitubular boiler for three weeks, then changing to the other new boiler and the Cornish. In about three months the new boiler which worked in connection with the Cornish began to leak in the girth seam over the furnace. I had it calked and tested. After running for three weeks more it leaked again on cooling; while running it did not leak. This was a condition that could not be allowed to continue, for calking was of no permanent value. The cause must be discovered and removed. The boilers were new and why should one be troubled in this manner and not the other? The answer came, you are forcing the boiler too much. To prove it, cut out the Cornish boiler, pull it out to make room for the other two 6x18-foot multitubulars and run the three boilers any two at a time. I did so and the leak stopped and has stayed so for the last eighteen months.

From this experience I have drawn the conclusion that the multitubular boiler is an element of danger when forced to generate steam beyond a certain capacity. Supplying more steam than can safely be generated is the short cut to a patched girth seam. When a boiler is forced, this seam expands so much that in contracting the rivets do not contract in the same proportion.

E. Browstorph.

Kingston, Jamaica, B. W. I.

An Emergency Decision

I read with interest the letter by Mr. Bloss under the above caption in the August 23 number. The question may be asked, how long can a boiler be run with any degree of safety when the water is out of sight in the glass? With a vertical water-tube boiler the conditions are more favorable than with other types, as there is less liability of burning the tubes. The tops of the tubes where they expand into the steam drum or header are vital spots in water-tube boilers even if they are reinforced. This is especially true if the bottoms of the drums are exposed to the direct action of the fire. The fusible plug would probably melt with 6 inches of water in the drum and prevent the pressure from rising and causing a serious explosion, but the tube ends are apt to be contracted and cause leakage and the tube sheet or header is in danger of becoming weakened and giving way later under working steam pressure.

Mr. Bloss evidently depended on the fusible plug for safety. He states that everything went well for several days. It seems to me that in this length of time a feed pump could have been connected up temporarily to the boiler. If a pump was not on the premises at the time, it would appear that Mr. Bloss had overreached his mark in trying to make good by endangering life and property in leaving a fireman in the plant at night with improper equipment to operate the boiler. I believe that he should not have started the boiler until the pump was received.

I left a watchman who had a fireman's license to fire some horizontal water-tube boilers one night. On entering the plant the next morning I found him very nervous and the work rather neglected. "Those boilers don't hold much water, do they?" he remarked. During four hours of the night the fires were covered and the damper closed as enough steam could be furnished with the fires in this condition. He had taken a nap during this period. When he awoke the water was out of the glass. The feed water was cold, but could have been heated by allowing the steam to blow through the engine. Instead of doing this he had pumped in the cold water, and as nothing had let go, he concluded that all was well. I opened the back connection door and the hood doors in front and found that a large number of tube caps had been started and were leaking badly. The boiler had to be run for three days and nights in this condition before it could be shut down and the caps reground.

In another case, three nipples connecting the header and the mud drum let go and put the boiler entirely out of service. Had the fire in this case been bright and working, these tubes would have been badly burned.

In still another case, both the fireman and the night engineer went to sleep. When the fireman awoke he found that he could not keep water in one of the boilers, which were of the horizontal tubular type. Investigation showed the fusible plug had melted and the top tubes were leaking badly. A new row of tubes had to be put in at a cost of $300, as the burned ones could not be rolled tight. The boiler was made perfectly tight but the tube sheet was weakened by being overheated.

R. A. Cultra.

Boston, Mass.

In the August 23 issue there appeared a letter by Mr. Bloss on an emergency decision. In my opinion the decision was foolhardy.

There might possibly be occasion for such proceedings, providing the engineer had been around the boiler immediately before the water passed out of sight, or knew the time that had elapsed since the alarm began to blow; if he knew that the columns and gages were working properly, and knew the actual distance from the lower gage cock or glass to the danger line, besides having some idea as to how fast the water would be lowered due to the load. I might add that he should be sure the blowoff valve was all right. Notwithstanding all this, it would not be good engineering but might be tolerated in extreme cases. But as Mr. Bloss says, he was about a mile from the plant when he heard the alarm, and he had no assurance of how long it had been blowing.

Going that distance, getting the fires started, and raising steam about 30 pounds, required considerable time and he had no guarantee that the injector would work immediately after obtaining the required pressure. I think the majority of engineers will agree with me when I say that it was foolhardiness, and such practice should be severely condemned.

Joseph Stewart.

Hamilton, O.

Operating Engineer versus Consulting Engineer

I notice that in his reply to Warren H. Miller, Thomas B. Whitted, in the August 30 issue, says that power plants designed by those whose sole experience lies along operating lines are likely to be of rather inefficient type. This may be true in most cases but, on the other hand, power plants designed by those whose only training is a college education are likely to be conglomerations of hair-splitting theoretical monstrosities entirely unfit for practical results.

What the power-plant owner is looking for is results; what the practical operating engineer is looking for is results. Besides, Mr. Whitted must remember that there are very few operating engineers today whose education lies solely along operating lines. It makes but little difference where a man gets a thing as long as he has it. The time has passed when the operating engineer is content with knowing how to shovel coal and squirt oil.

Of course, consulting engineers are necessary adjuncts in the engineering world; so also are operating engineers. We must not jump at the conclusion that an operating engineer is incapable of understanding the theoretical part of steam engineering. The sooner this idea is laid aside and the consulting engineer comes to the conclusion that he is not the whole thing and the operating engineer conceives the fact that he can learn some things worth knowing from the consulting engineer and the consulting engineer realizes the fact that his dignity does not suffer by considering a suggestion offered by the operating engineer, the sooner will the plant owner secure better results.

To show the inconsistency of some consulting engineers I will relate an incident which came to my notice recently. When a certain consulting engineer was figuring on a plant I asked his opinion regarding centrifugal pumps for boiler-feed purposes. He remarked that they were only a fad. But I noticed that when the specifications came out centrifugal boiler-feed pumps were specified. From this it would seem that this particular consulting engineer figures on fads for his clients.

I also asked him about his opinion of boiler-feed regulators. He said that he would not recommend such things as he would not have a man in a boiler room who could not look after the water properly without such things. This man did not consider the economy of such appliances at all but condemned them merely because they took some of the work off the boiler-room attendant. I know from practical experience that good boiler-feed regulators will pay for themselves several times in one year.

The sooner steam-plant owners learn that their regular engineer has as great an interest in their plant as anyone and is best qualified to pass on contemplated improvements, the better off they will be. While he may not be qualified to design a plant, he is best qualified to judge whether the apparatus designed by the consulting engineer is the proper thing for his plant or not.

When the consulting engineer and the steam-plant owner acquire sense enough not to feel that their authority has been invaded or their dignity lowered by calling in the operating engineer, results which will be more nearly ideal will surely be secured.

H. R. Rockwell.

Mt. Vernon, Ill.

Rope Transmission Problem

Replying to Mr. Kirlin's question in the August 30 issue regarding the "Rope Transmission Problem," I believe that one sheave being $\frac{1}{4}$ inch smaller in diameter than the other will impair the efficiency. Assume that the two sheaves are not keyed to the shaft but can revolve around it, each sheave independent of the other.

Let A denote the sheave whose diameter is 9 feet, and let B denote the sheave whose diameter is 8 feet 11$\frac{3}{4}$ inches.

The revolutions of the driven sheave equals the revolutions of the driver multiplied by the diameter of the driver and divided by the diameter of the driven sheave.

Assume that the driving sheave, which is 14 feet in diameter, makes 1000 revolutions per minute; then, the revolutions of

$$A = 1000 \times \frac{14}{9} = 1555.5'$$

The distance that the rope will travel per minute is

$$1555.5 \times 3.1416 \times 9 = 43{,}980.46 \text{ feet.}$$

For the same number of revolutions of B the rope will travel

$$\frac{1555.5 \times 107.75 \times 3.1416}{12} = 43{,}878.06 \text{ feet.}$$

Under the conditions cited by Mr. Kirlin, for every 1000 revolutions of the engine pulley there is a rope slippage of 102.4 feet on either A or B. Now, proceed to determine on which sheave the slippage occurs. The horsepower transmitted to the sheave is

$$\text{Horsepower} = \frac{2\pi D \times RPM\,(T - F)}{3 \times 550}$$

where,

T = Tension on driving side of the rope, in pounds;

F = Centrifugal force of the rope in pounds;

D = Diameter of the sheave.

From this formula it will be noted that the larger D is, the greater will be the horsepower transmitted. Therefore, it follows that the rope will slip on the sheave which has the least horsepower transmitted to it, which in this case is B, the smaller sheave. This means that the ropes which travel over this sheave will wear faster than the others.

G. F. Atwater.

South Norwalk, Conn.

Soot and Ashes on the Tubes

Referring to William Westerfield's article in the August 23 issue of Power, I take exception to some of the impressions which it left with me.

In the plant where I am employed there are three 250-horsepower horizontal water-tube boilers, two of which are in one setting, the other one being in a setting by itself. The distance between the double and the single setting is about 36 inches. Oil is used for fuel. The oil has a specific gravity of 14 degrees Baumé and contains an average of 8 per cent. basic sediment and moisture. The plant is operated continuously. Each boiler is cut out and cleaned once every 30 days. This is done with a mechanical cleaner. The cleaner is run through the tubes from the front end, the space required for this purpose being about 8 feet. The soot is blown from the tubes by steam pressure applied through the front sheet, at the side of the setting and through the rear. It is quite true, however, that this alone will not keep the tubes clean and that some other work must be done on them from time to time.

As to Mr. Westerfield's "assumed" and "actual formation of soot on the tubes," he claims that Fig. 1 in his article is an extreme case to be found only "in some Arkansas sawmills." While our plant here can hardly be classed as an "Arkansas sawmill," I am forced to confess that the boiler tubes are in that condition and we do not consider it as an extreme either. In fact, close observations in practical experience lead me to class Mr. Westerfield's Fig. 1 as representing a more common condition than his Fig. 3, which he designates as representing actual conditions.

If there were nothing to contend with but the soot, Mr. Westerfield would be right in speaking of Fig. 1 as representing an assumed formation, but so long as ash is one of the products of combustion it also will have to be considered. The ash which concerns us is that which is in the form of dust. The construction of the horizontal water-tube boiler is such that the hot gases are baffled by the tubes being set staggered. This gives a condition which produces a dust trap on the top of each tube where the gas is not in rapid motion.

B. F. Hartley.

Tipton, Cal.

Engineers' Wages

In the September 13 number I find a very interesting piece of literature on "Engineers' Wages," the author of which is certainly not bashful about letting us know what rot an engineer (?) can dish out when writing about something beyond his accustomed horizon.

His daring to distinguish the engineer from the "stopper and starter" is equaled only by his innocent childishness when telling about the good, satisfied engineers working for $45 per month. I can imagine his wrath at finding a lot of progressive and aggressive "stopper and starters," who, by the way, probably would not submit to his teachings, working for $85 per month, and kicking at that.

His remarks about "unions" show an utter ignorance of the fundamental principles of self-preservation as well as a good-size affliction of what is generally termed "swelled head." Intelligent people are not opposed to unions. Our forefathers fought for the Union; they knew by dear experience that divided they would fall. The lawyers have their bar associations to uphold their standing and, incidentally, prices. The medical associations are formed for the same purposes, also the civil and mechanical engineering societies and similar organizations. Carpenters, masons, machinists, etc., bettered their financial and social conditions through unionism. Why should the operating engineer be the only "fool," to try to better his social standing single handed? The National Association of Stationary Engineers is a living proof of the fallacy of standing on your own merits, and allowing your employer, some say exploiter, to be the jury of one to fix the dollar value of these merits.

At present, there are a lot of good engineers (not the $45 kind) hard at work to form and organize an uptodate association, "The Institute of Operating Engineers," which, as far as I can help, will be a union for education, elevation and protection.

I am proud to be a charter member of this really progressive institution and hope that all readers of Power will step nearer and investigate, and help to build up a powerful organization for *our own* benefit, an organization that will enable us to claim and secure by superior education and stick-together-activity the professional and social recognition the uptodate stationary engineer has deserved for so long.

H. F. Heyrodt.

Bridgeport, Conn.

Inquiries of General Interest

Selection of a Steam Boiler

What are the three prime requisites to be considered in the selection of a steam boiler?

S. R. B.

There are many things to be considered in the selection of a steam boiler, among the most important of which are safety, durability and the economy of operation.

Vacuum

What is meant by vacuum?

M. V. B.

By definition, vacuum is empty space. In steam-engine practice the term refers to space in which the steam is or has been condensed to a greater or less degree, and is never really vacuum.

Pressure in Steam Boilers

Is the pressure equal in all parts of a steam boiler?

P. S. B.

No, the pressure on the bottom of the boiler is greater than that on the top by the weight of the water. The pressure of the steam is exerted equally in all directions on the water as well as on the shell, but the weight of the water is supported by the bottom of the boiler only, and in addition to the pressure of the steam.

Foaming Boilers

What is foaming and what will prevent it?

F. I. B.

Foaming in steam boilers is a violent frothy ebullition which carries water in the form of foam to the steam space and the main steam pipe. It can be prevented by using clean water in a clean boiler.

Object and Size of Safety Valves

What is the object of a safety valve and how is the right size for any boiler found?

O. S. S.

The object of a safety valve is to prevent the pressure in a boiler from rising higher than some predetermined point, by opening at this pressure and allowing the excess of steam to escape to the atmosphere. Its capacity to relieve the boiler is calculated from the amount of water that may be evaporated by the boiler. It is customary to allow ⅓ square inch of safety-valve area for each square foot of grate area when the valve is "spring loaded" and ½ square inch of valve area per square foot of grate when the valve is of the ball and lever type.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Air and Moisture

I wish to pass a current of air over a steam coil for the purpose of drying it. How hot will the air have to be to be thoroughly dried?

A. A. M.

Heating the air will not dry it at all, but, on the contrary, will increase its capacity for absorbing moisture. Air can be thoroughly dried by freezing.

Comparative Evaporation

How may the capacity of a boiler taking feed water at 70 degrees temperature be compared with one which is being fed with water at 180 degrees?

B. E. C.

By reducing both to the standard of equivalent evaporation. Subtract the number of heat units in a pound of feed water at the given temperature from the number of heat units in a pound of steam at the temperature at which it is made and divide the difference by 970.4. The quotient multiplied by the number of pounds of water actually evaporated will give the equivalent evaporation from and at 212 degrees, that is, the weight of water which would have been evaporated from feed water at 212 degrees into steam at atmospheric pressure. By doing this with both boilers their respective work may be compared.

Multiported Valve Lead

In setting the Allen valve, should it be given the same lead as the plain slide valve?

M. V. L.

Yes, but owing to the fact that the Allen valve is double ported it should have but one-half the apparent lead or port opening as a plain slide valve, as the lead is really double what it appears to be, because the steam is admitted to the cylinder past two valve edges instead of one.

Compound Engine Cylinder Ratio

In a noncondensing compound engine the high-pressure cylinder is 18 inches in diameter. What should be the diameter of the low-pressure cylinder, if 150 pounds steam pressure is used?

E. C. R.

For an equal distribution of work between the cylinders the number of expansions in each should be equal. Expanding to five pounds pressure the number of expansions will be 8.25. The cylinder ratio should be the square root of the number of expansions

$$\sqrt{8.25} = 2.87$$

Therefore the area of the low-pressure cylinder should be 2.87 times that of the high

$$2.87 \times 254.47 = 730.32$$

square inches.

To find the diameter of a circle from its area, use the formula

$$D = 2\sqrt{\frac{a}{\pi}}$$

Substituting, the formula reads

$$\mathit{Diameter} = 2\sqrt{\frac{730.32}{3.14}} = 30.5$$

inches, the proper diameter for the low-pressure cylinder.

Does the Crosshead Stop?

Does the crosshead of an engine stop at the end of the stroke, and, if so, how can it be proved?

C. H. S.

During one stroke the crosshead moves in one direction and during the next it moves in the opposite direction. If it does not stop there will be an interval between the changes in the direction of its movement in which it must travel in two directions at the same time, which is absurd.

Temperature of Gage Spring

Why is the pipe leading from the steam gage to the boiler always cold even when there is 100 pounds pressure in the boiler?

T. G. S.

The pipe leading to the gage is always put up with a bend in it which serves as a pocket in which water collects from condensation the first time the boiler is fired. It always stays in the pipe and prevents live steam from coming in contact with the gage spring and causing it to register incorrectly as it would if exposed to the temperature of the steam in the boiler.

Running Engine Single Acting

Can I run a Corliss engine with one exhaust valve removed and the steam valve blocked shut while a new stem is being fitted to the exhaust valve?

J. I. D.

Yes, if the flywheel is heavy enough to carry the engine through one-half revolution and over the center with full load.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T McKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for September, 1910, *was* 134,000.

October 4.............. 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

The Cost of Power

It has been so frequently repeated that an engineer should know more about the art and science of steam engineering than any other one subject, that another repetition of the statement seems almost, if not quite, out of place. But the rate at which the central station is placing contracts for furnishing current where it is evident that the local plant has a manifest advantage when all of the factors are considered, shows that the engineer is handicapped by his inability to show the proprietor that central-station service is not the cheapest available.

In New York City the engineer is crowded harder than elsewhere in the country by the competition of the central station. But here too he has the best opportunity to show that he can "make good" if he so desires.

It is not necessary to say that the man in direct charge of an isolated plant should know just what can be accomplished in that plant. It is self-evident. But there are hundreds of plants in the city where the engineer has no more idea of the actual cost of the service he furnishes than he has of the plan of salvation. But the central-station solicitor has. He knows how much coal is burned and the amount of light and power furnished. He knows the payroll and the cost of extra work. He has all the data at his finger tips which he uses with a glibness of tongue and the convincing manner of one who knows, and knows that he knows.

Talking in a logical way about things with which he is familiar, and reasoning directly from the coal bunker and water meter to the switchboard and thence to the proprietor's pocket is not the average engineer's long suit, and in competition with the trained central-station man he appears at a disadvantage. But if he knows what he is doing and can show in plain English and correct figures the coal cost, the labor cost, interest and depreciation charges, water rates and repairs there will be little danger that the central station will ever take over his plant, for he will generally be able to show that he can produce electric energy at a lower price per kilowatt-hour than any central station, however well equipped, will furnish it.

If any New York engineer feels that the task of proving what can be done with his plant is beyond the limits of his ability, the general committee of the subordinate associations of the National Association of Stationary Engineers of Manhattan and the Bronx, is at his service.

This committee has a complete line of apparatus suitable for power-plant testing, and is prepared at any and all times to make an exhaustive test of any plant operated by a member of the association.

If at any time it is intimated that heat, light or power may be bought for less than it costs to make it at home, the engineer is, if he does not already know, in a position to have determined for him the exact cost and the product of every department under his supervision. From the results of tests made by the committee tabulated statements can be made which may be submitted for comparison with the proposals of the central-station solicitor. The committee is at the service of any member of the National Association of Stationary Engineers. If an engineer who is not a member of the organization needs the service, the proper course for him to pursue is obvious.

Newspaper Accounts

The following extracts from the account of an explosion in a power house in Cambridge, as described by a Boston paper, illustrate how accidents are often misrepresented by the daily press and confirm the statement that anything will do to fill up the news columns.

The report commenced by saying that a boiler exploded, seriously injuring one man and endangering the lives of six others, also badly wrecking the building. A little further it said that the boiler was the largest of four in the plant. Now, after giving the impression that it was a boiler which exploded, the next few lines contained the statement that "one section of the engine shot upward, tearing a hole in the roof." One of the officials, whose business is to issue such reports, is quoted as saying, "the explosion was confined to a receiving engine having a capacity of 2700 kilowatts." And, by way of further explanation, the account says that "the engine which exploded is used to receive the high-tension steam where it is transformed into low tension. It is thought that the capacity of the engine was overtaxed."

Now, for a large number of people this account is sufficient, but for those who would be interested in knowing the real facts of the case, it is extremely inadequate and confusing. There is no doubt that something was overtaxed, but it might as well be the memory or the imagination of the reporter as anything else, for, while the reader is thinking about a boiler explosion, he is suddenly switched over to the consideration of an engine breakdown, and a receiving engine at that, which might be run by electricity as it is measured in kilowatts. One would also be led to believe that it was a new form of transformer for steam which is measured by tension instead of pressure.

What actually happened was this: The receiver between the high- and low-pressure cylinders of a compound engine burst, and caused all the trouble. If there is anything about a receiver that refers to boilers, kilowatts or transformers, it would be interesting to know it. If a newspaper would send one of the firemen in its own steam plant to write up such accidents they would in most cases give a better account than the one described.

A Head for the Steam Plant

Operating a power plant twenty-four hours a day, week in and week out, is much different from operating the same size of plant but ten hours a day. Not only is the wear and tear on the engines, boilers and auxiliaries greater, but the fact that two, and in some cases, three sets of men are employed adds greatly to the difficulty in maintaining smooth operation. Every steam plant requires a head—someone who is responsible for all that takes place, whether he is present or not.

One of the difficulties encountered in power plants is that the men in one watch are likely to shift the responsibility for some mishap to the men of the other watch, if it is possible. Take, for instance, the case of keying up an engine. It is a simple matter, but considerable trouble has been caused by improper keying which has not been reported to the relief engineer.

In one case, an engineer decided that the main pillow block of an engine required taking up and at the end of the run proceeded to carry out the work, but he neglected to notify his relief of what had been done. When the relief engineer started the engine, the bearing began to heat badly, but it was noticed and relieved in time to prevent the damage to the bearing. If the engineer had not been watchful the engine would doubtless have been disabled, and all because the relief engineer had not been notified of the adjustment.

There should be one man to look after such work as keying the engine, and that man should, of course, know how to do it. Furthermore, when work of this nature has been done the force coming on duty should be notified. Even the oiler should be told to watch such-and-such a bearing more closely than usual in order to prevent heating in case the box or bearing has been too finely adjusted.

In moderately small plants running on a twenty-four hour basis trouble is frequently caused by either the day or night engineer adjusting the engine to his liking. Under all circumstances the day engineer is the one to make all adjustments. The night engineer should not touch the engine, unless it is something that is absolutely necessary in order to keep it running.

If the night engineer thinks he knows more about an engine than the day man, he should keep his thoughts in the background, and wait until he gets a day run before he tries to put his knowledge into practice. The day man is really the chief and should be so considered by all employed in the plant.

The work in the boiler room also demands a head. Because a certain routine has been carried on for years does not mean that it will be continued. A change in the position of a fireman may cause trouble galore. In a certain plant the head fireman resigned and the next best man was put in charge. One of the ten boilers was washed out each week and this fireman had assisted in the work for years.

In promoting him the chief engineer assumed that the washing of boilers would go on as formerly and, therefore, gave the matter no personal attention. For some reason the new head fireman did not have a boiler washed for four weeks, so that one boiler was run for fourteen weeks without being washed out. At the end of that time it bagged so badly that the fireman's negligence was discovered by the chief engineer.

These are only a few of the many instances that might be mentioned in showing that the power plant requires a chief who will keep in touch with the routine work and see that everything is properly done.

Technical Education

The feeling of contempt so often evinced by the practical man toward the young technical graduate is largely due to too much being expected of the latter through a misconception of the aims and limitations of the technical school. Within the past ten or fifteen years, engineering methods have undergone a vast change, and their scope and magnitude have increased to such an extent that the technical school, in order to keep pace, has been forced to revise its curriculum. The fact is now appreciated that there exists a close relation between the technical school and the engineering profession, but that in order to succeed, the former must base its work upon a thorough understanding of this relation. This is now being accomplished through a study of industrial conditions, with courses arranged to fit these conditions.

Formerly too much emphasis was placed upon the manual features of the curriculum, and attempts were made to accomplish in the laboratory what could be done only in the factory. The technical schools do not attempt to teach trades. and are now breaking away from the manual-training idea, giving only suc shopwork as will enable the student t become familiar with modern methods o production. This seems a wise expedient as the student could not become an expert mechanic even within the maximu amount of time that was formerly allotte to shopwork, and this excess time was a the expense of other subjects essentia to modern engineering. Further, it i questionable whether one man out of te would have occasion later in life to mak use of his handiness with tools, and suc men can acquire this skill in shops outside of the school. Many of the larg manufacturing and operating companie are recognizing this fact and have established apprenticeship courses to supplement the course in the technical schoo With an engineering foundation to buil upon, they supply the necessary practical training to make the men of greate use in their particular field.

Another feature that the engineerin schools are breaking away from is specialization. This is largely due to tw causes: First, the average student doe not know what particular line of eng neering (this does not apply to the tw great divisions, civil and mechanical e gineering) he will follow, and as oppo tunity is the deciding factor in such case he is apt to go into an entirely differe branch from that which he studied, which case much of the time devoted the details of a specialty is wasted. Se ond, technical schools cannot give cours in special branches that would be equiv lent to the knowledge gained by sever years' practical experience in tho branches, but they can give a broad trai ing in the fundamental principles of e gineering, embodying a familiarity wi other branches, which will enable the i dividual to make the most of his futu experience—the training which no amou of experience in a special branch cou supply.

It is a fallacy to suppose that a su cessful technical education depends up book knowledge. Textbooks are only means toward an end, and their conte are forgotten soon after the class roo is forsaken for the business or constru tion office. The aim of the technic school is to give a training that will e able the student to systematically atta and solve problems as they arise, to pr mote a faculty for securing informati along any line when needed, and to co ceive, organize and direct extensive en neering projects, as well as industrial e terprises.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Watts Feed Water Regulator

This feed water regulating system will operate with pressures varying from 5 to 200 pounds, and depends upon the variation of water level, and the steam pressure in the boiler for its action.

It consists of a copper ball which is secured to an L-shaped casting in which two passages have been provided, one for water and one for steam, as shown in Fig. 1.

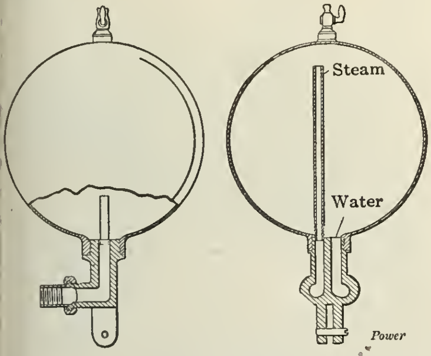


FIG. 1. SECTIONAL VIEWS OF BALL

This copper ball and L connection are attached to the lever of the feed water regulating valve which is placed in the feed pipe line leading to the boiler, as shown in Fig. 2. The ball is also connected to two pipes running to the water column. The pipe leading from the passage connecting with the small vertical pipe in the ball is connected at a point in the water column at which the water level is to be maintained. The water pipe is connected to the water column at a lower point.

The arrangement of the ball and lever is shown in Fig. 2. The regulator may be placed at any distance from the boiler by using longer connecting pipes. The copper ball must always be placed above the water line in the boiler.

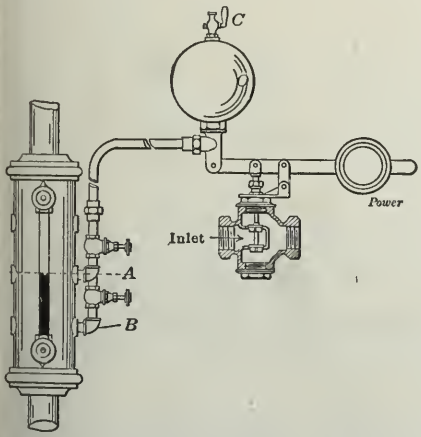


FIG. 2. ARRANGEMENT OF COPPER BALL, VALVE AND PIPING

A sectional view of the balanced regulating valve that controls the feed-water supply is shown in Fig. 3. If conditions are such that the weight and ball lever cannot be attached to the regulating valve, connection can be made by a chain, as shown in Fig. 4.

The general arrangement of the regulator when attached to a boiler is shown in Fig. 5. The pipe *A* is ⅜ inch in diameter and is connected to the water column or to the boiler at the desired point of water level. The pipe *B* is ⅜ inch in diameter and is connected to the water space of the water column or any point below the water line. A ⅜-inch valve is placed in each of the pipes *A* and *B*, near the connection on the water column, so as to cut out the copper ball from the water pressure if necessary. The valve *M* is placed in the feed pipe above the water line of the boiler and the weight *G* is placed on the lever so that it will overbalance the ball *O* when it is empty. The air valve *C* allows air to escape when firing up a cold boiler.

If the water in the boiler or water column is below the pipe *A*, or the water line, the ball *O* will remain empty; also the valve *M* will be opened, because the weight *G* is heavier than the copper ball.

When sufficient water has passed into the boiler to cover the entrance to the pipe *A*, the water will rush into the ball

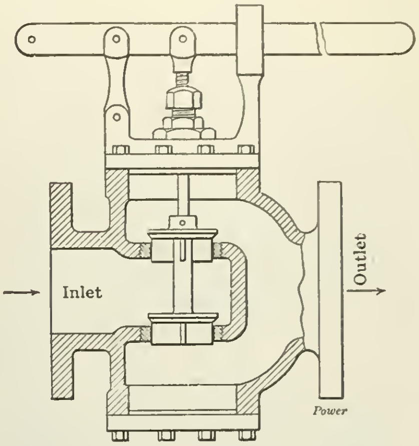


FIG. 3. BALANCED REGULATING VALVE

O, and when a sufficient quantity of water has entered the ball, it overbalances the weight *G* and at the same time closes the valve *M*. The valve M will remain closed until the water in the boiler or water column falls below the pipe *A*, when the water will immediately be discharged into the boiler and steam will take its place. The ball *O* will then be overbalanced by the weight *G* and the ball in rising will open the valve *M* and admit more water to the boiler. The valve *M* will remain open until the boiler is again filled up to the point indicated by *A*, when the operation will be repeated and continued indefinitely. The pipes *A*

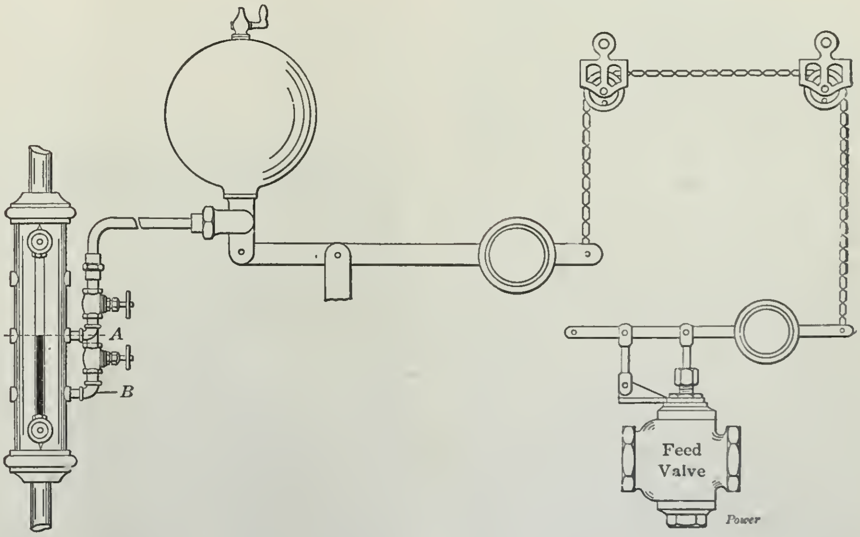


FIG. 4. REGULATOR CONNECTED WITH FEED VALVE BY CHAIN

and *B* are 4 feet long and, being of small size, are flexible.

The water pressure in the feed pipe, which builds up as soon as the valve is closed, is taken care of by a pump regulator valve shown in section at the left of Fig. 5.

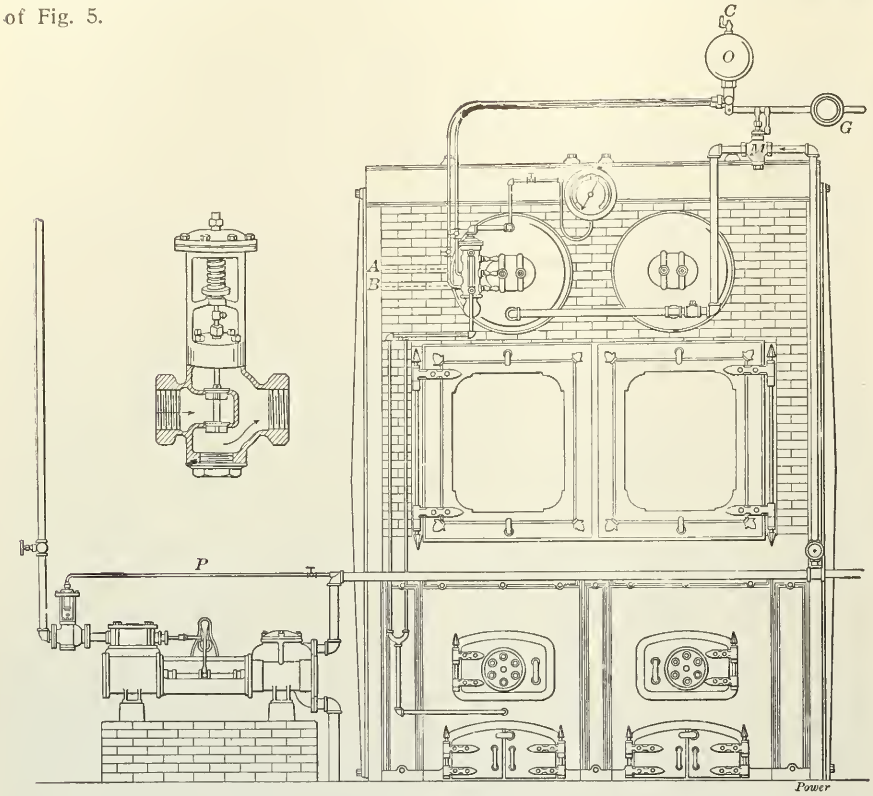


FIG. 5. REGULATOR CONNECTED TO A WATER TUBE BOILER

This valve is balanced and is operated by the water pressure in the pipe, which is exerted against a diaphragm placed above a coiled spring. When the water pressure on the diaphragm overcomes the spring, the balanced valve in the steam pipe line feeding the pump is closed. As soon as the water pressure is reduced, the spring acts and lifts the valve disk from its seat.

The system is automatic in action and requires practically no attention. This regulator is made by the Watts Regulator Company, 250 Lowell street, Lawrence, Mass.

Fugo Grease Remover

This is a liquid preparation made to remove burnt oil or grease from the metallic parts of an engine. It is applied with a small paint brush or a small cotton dauber. After it has been spread over the grease evenly and a few seconds allowed for action to take place, the hardest grease will disappear, it is claimed. After the surface appears bright it must be wiped with a piece of waste.

The fluid acts quickly and ceases to act as soon as the grease is removed, it is said.

This fluid is made by the Fugo Manufacturing Works, Collegeville, Penn.

Osborne High Pressure Valves and Fittings

These specialties are designed to take care of high-pressure work and are suitable for ammonia, steam, either saturated or superheated, air, oil, gas or hydraulic service. The valves are made in both globe and angle varieties and, it is claimed, contain a feature of construction whereby cutting or wire drawing of the seat is practically eliminated.

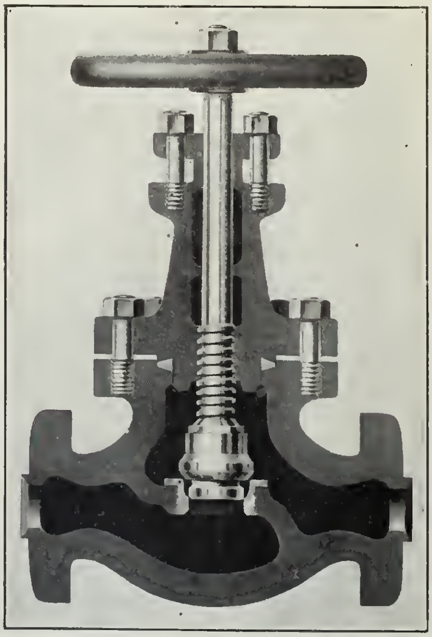


FIG. 1. OSBORNE GLOBE VALVE

Fig. 1 shows a globe valve in section. The disk of the valve carries an extension which closely fits the machined inside surface of the seat. With the valve partly open, the disk extension throttles the flow of steam, transferring the cutting from the seat to the cutting edge provided for this purpose. This prolongs the life of the valve and eliminates wear on the disk, it is claimed.

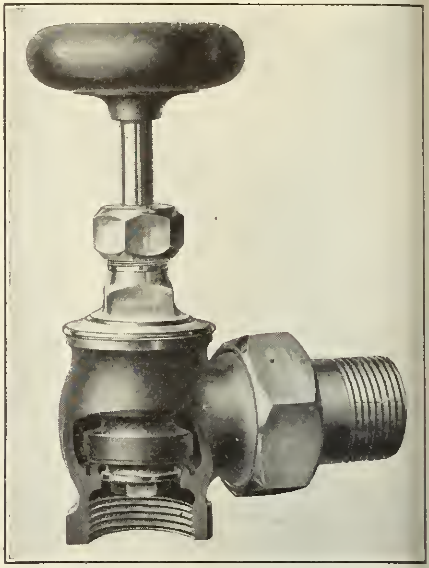


FIG. 2. RADIATOR VALVE

No considerable flow can take place when either opening or closing the valve, until the disk and seat are far removed from each other, thereby eliminating wire drawing at the place where it would affect the tightness of the valve.

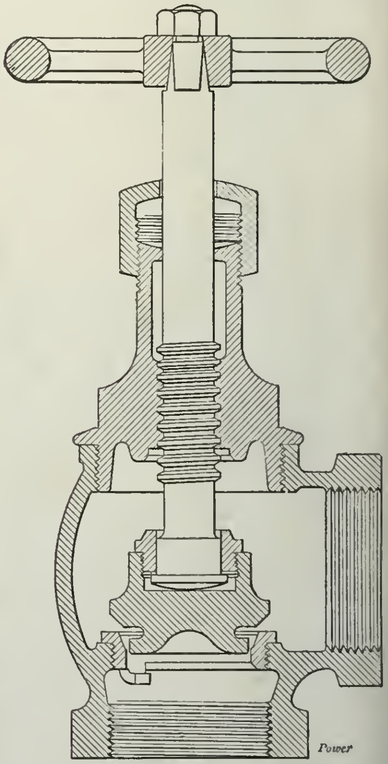


FIG. 3. ANGLE VALVE

Fig. 2 shows an application of the same principle to a radiator valve, the disk extension fitting closely to the inside surface of the valve seat until the valve itself is opened sufficiently so that wire drawing cannot occur.

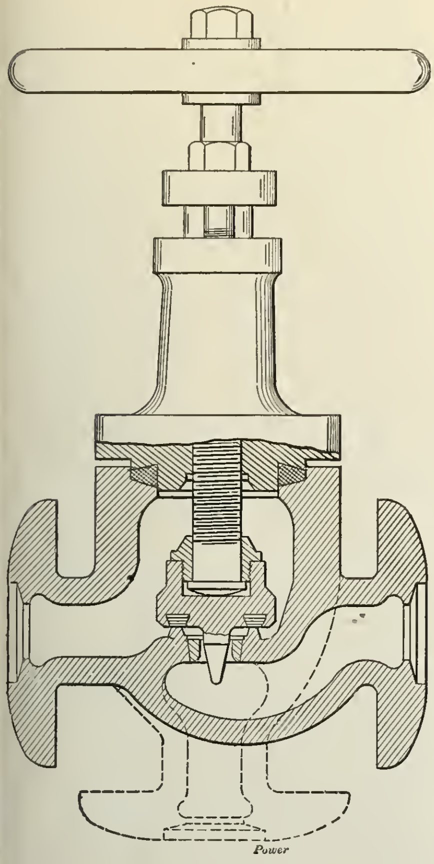


Fig. 4. Expansion Valve

Fig. 3 is a sectional view of an angle valve having the same design of seat and valve disk. A modification of this valve is built for expansion service on refrigerating lines. A section of one of these is shown in Fig. 4. The extension in this case extends through the expansion ring and when the valve disk is raised the pressure across the seat is balanced, transferring the cutting to the swivel extension and expansion ring. These two parts are hardened steel, thus reducing the cutting to a minimum.

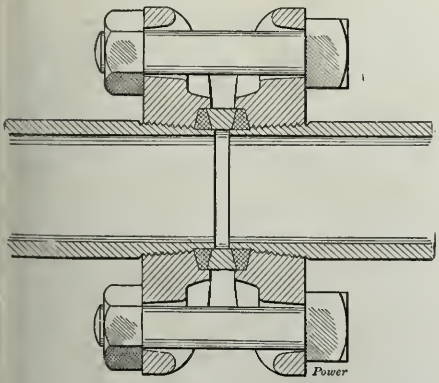
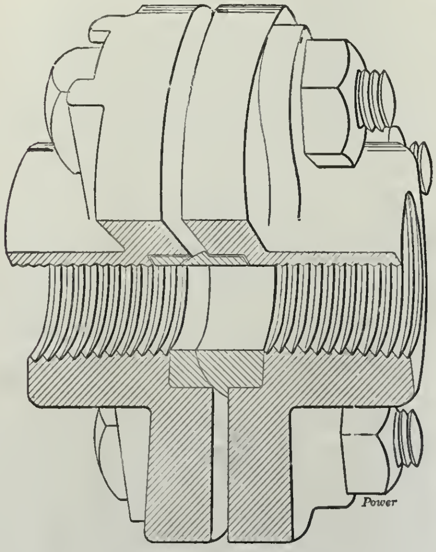


Fig. 5. Standard Flange Coupling

A standard flange coupling is shown in Fig. 5. In making this joint a pipe is threaded a considerable distance from the end and a flange screwed on loosely. The gasket, made of fiber or other composition suitable for the service, fits in a recess in the flange and fits closely around the pipe. A metallic ring is placed between the flanges, covering the junction between the ends of the pipe, making an unbroken metallic surface on the interior. This construction entirely incloses the gaskets, two of which are used on each joint, making it impossible for them to blow out, and enables the joint to be made repeatedly with the same gaskets, it is claimed. As no pressure is placed on the threaded connections, no soldering or sweating is necessary. Further, no welding or facing of the pipe ends is required as they do not touch when the joint is made up. Owing to the construction of the flange it is claimed that absolute alinement is not necessary and that the joint will stand a large amount of vibration and expansion. These valves and fittings are made by the Osborne High Pressure Joint and Valve Company, "The Rookery," Chicago, Ill.

The Dart Flange Union

This flange union consists of two threaded flanges which are screwed onto pipe lengths. One flange is fitted with a female seat, the other with a male seat; both are made of bronze, ground true on the bearing surface. The seats are held together by suitable bolts, as shown in the illustration.



Dart Flange Union

This kind of flange coupling permits of connecting pipe lengths when not in line. This flange union is made by the E. M. Dart Manufacturing Company, Providence, R. I.

Pyramid Double Acting Pump

The Goulds Manufacturing Company, Seneca Falls, N. Y., have recently placed on the market the "Pyramid" double-acting piston pump.

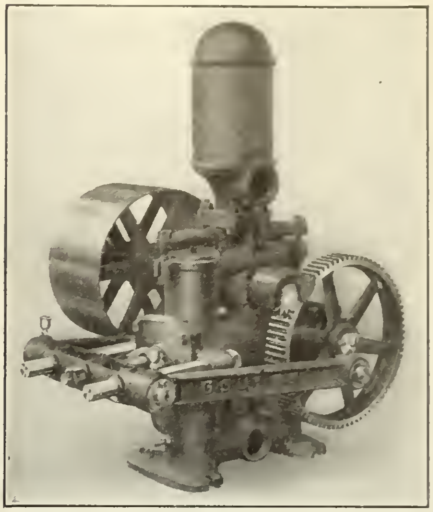


Fig. 1. Pyramid Double-acting Pump

The base, cylinder, bearings and one cylinder head are cast integral, thus giving stiff and rigid construction. One cylinder head is cast on the cylinder. As the bearings are cast as a part of the cylinder, the gearing is kept in line.

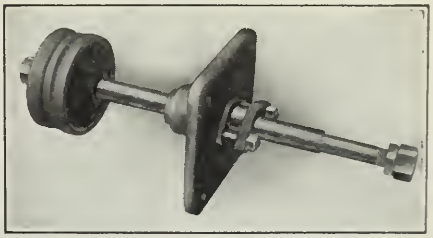


Fig. 2. Head and Piston

The cyilnder is brass lined, and is provided with large suction valves, placed just below the bore of the cylinder; these valves are easily accessible through two handholes, the covers of which can be removed by backing off one bolt.

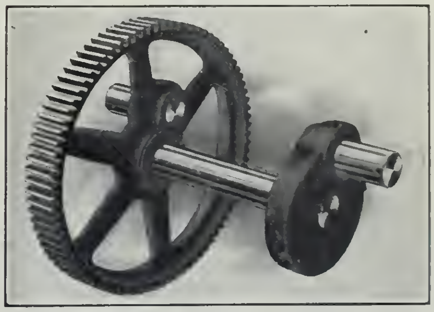


Fig. 3. Pinion Shaft

The discharge valves are placed above the cylinder and are reached through handholes located on both sides of the air chamber. An exterior view of the pump is shown in Fig. 1.

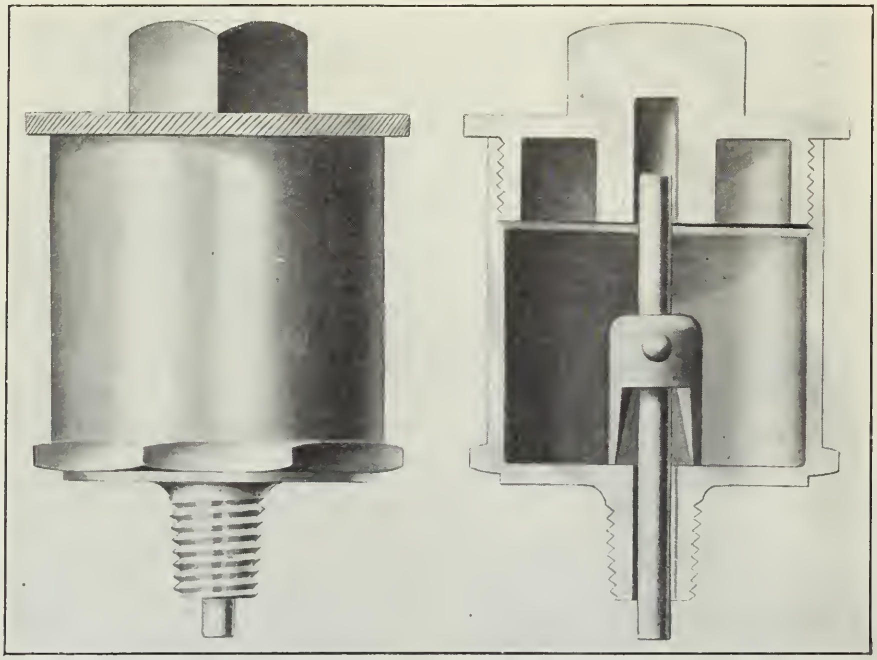
Two steel rods act as a guide for the crosshead. The stuffing box on the cylinder head is provided with a bolted bronze gland. Fig. 2 shows the design of the head and piston.

The power end of the pump consists of a pinion shaft, Fig. 3, supported on a long, double bearing, placed in the center of the pump directly under the air chamber. No outboard bearing is required. Both gear and pinion are made of cast iron, and the teeth are cut from the solid metal. The pump is geared 5 to 1, and is suitable for work up to 75 pounds pressure.

The Jackson Crank Pin Grease Cup

Two views of this grease cup are shown in the accompanying illustration. As shown, by the sectional view, the cup is fitted with a plunger which works up and down, carrying enough grease to lubricate the crank-pin brasses.

This cup can be attached to the strap of any crank rod. The stop shown in the sectional view prevents the plunger rod from dropping into the oil hole in the crank pin, if such hole exists. The cup requires no regulation and does not feed except when the engine is running. It is made by Richard D. Jackson, 96 Warren street, New York City.



Jackson Crank-pin Grease Cup

Within 175 miles of Boston, declared an expert before the Massachusetts gas and electric light commission, water capable of yielding 300,000 horsepower is going to waste. One does not have to go into the West to apply conservation.

Meeting of the Central States Water Works Association

The Central States Water Works Association held its fourteenth annual meeting at the Claypool hotel, Indianapolis, Ind., on September 20 to 22. Although the attendance was not large, some interesting papers were read, among which there was one on the subject of rewarding fireman for efficient service, by F. W. McNamee, of Wabash, Ind. In the plant with which he is connected he said that the question of combustion had been investigated and it was found that the results they were getting were bad. Analysis of the flue gases developed the fact that the percentage of carbon dioxide was running from 4 to 5, and sometimes 6 per cent. After studying the problem for some time, a scheme was worked out of paying the fireman on a bonus basis. They are now paid on a flat rate per hour, plus a bonus depending on the percentage of carbon dioxide in the flue gases. Each boiler is supplied with a five-gallon glass bottle, the top being connected through brass pipe to the breeching, just under the stack damper. This bottle has in the bottom of it a brass valve containing a calibrated plug so that in twelve hours it will just empty itself, the water dripping out. The bottle is filled with water at the beginning of a run and on top of the water is placed a small amount of kerosene to prevent solution of carbon dioxide by the water. Each fireman, when he goes on duty, puts on as many bottles as there are furnaces for him to run and starts them to dripping, with the result that at the end of the run there is an average sample of the gas produced for each furnace under the fireman's charge.

When the fireman finishes his run he takes off the bottles, labels them, putting on his name and date, and the next day the chief engineer analyzes the gas and the fireman is given credit for the carbon dioxide found. The bonuses run from 15 cents a day to 70 cents. If the fireman shows an average percentage of five, or less, he is taken off the job, because he would be an expensive man if he worked for nothing.

If the analysis shows six per cent. of CO_2, he is paid 15 cents a day extra, and so on up to 70 cents, according to the following table:

Per Cent.	Cents.
6	15
6½	20
7	25
7½	30
8	35
8½	40
9	45
10	50
11	55
12	60
13	65
14	70

It has been found impractical to try to go beyond 14 per cent. because losses in other directions more than offset the increased gain through higher CO_2. Experience shows that with this kind of an incentive the fireman will pay a great deal better attention to his business, and instead of firing a few scoops of coal into the middle of the grates he will fire often, and fire thin, keeping his grates thoroughly covered.

The consumption of coal at the plant where this plan has been enforced is from 12 to 16 tons a day, and the saving, after paying the bonuses of the fireman, has averaged from $6 to $8 a day. The institution of this system has proved to be the most profitable investment made at this plant during the past year.

In a discussion of the steam-turbine centrifugal pump, C. H. Hurd brought out the conditions under which this type of pump would prove the best investment. His paper read in part as follows:

In considering the economical advantages of the various types of engines or pumps, it is necessary to take into account not only the first cost or interest upon invested capital, but also the depreciation and maintenance as well as the operating expenses. Unless all of these items are considered it is not possible to come to a proper conclusion as to which would be the most economical type.

In comparing the reciprocating and turbine type of pumps it is also necessary to consider not only the cost of the units themselves, but also the cost of foundations, buildings and boiler plant, as it will readily be seen that the total cost of delivering water will include both the fixed charges and operating expense of the entire plant. It is obvious that a high fixed charge may mean a low cost of operation, or, on the other hand, a smaller investment in pumping equipment and a correspondingly lower duty may mean a higher boiler-room expense and additional boiler capacity.

In making the comparisons, a plant has been selected with an average pumpage of 10,000,000 gallons for 24 hours, which is perhaps slightly larger than the average water-works plant in the United States. In designing an equipment to take care of the service for the average city it would be considered poor judgment to make the pumping capacity less than twice the average consumption, while perhaps it would be better to divide the installation into three or four units, particularly if the plant is to operate against a direct-pressure system. For the purpose of this paper we assume that the city has a reservoir of such capacity that two 10,000,000-gallon pumps would be sufficient.

As to the cost of such a plant, it is understood that the figures can only be approximate, and that they will vary in both cases according to the conditions and demands. In assuming the operating conditions, we will consider the price of coal at $2.50 per ton, fired into the boilers, and the head pumped against to be 200 feet. The cost of such a pumping plant, using vertical triple-expansion pumping engines, for twenty million total capacity would be about $180,000. With interest and depreciation at 9 per cent., the fixed charge would amount to $16,200 per year, or $4.33 per million gallons pumped. It would be fair to assume that the centrifugal pumping plant, to do like service, including boilers as above, would not cost more than $90,000, making a fixed charge per million gallons of $2.16.

In the case of the triple-expansion engines under most favorable conditions, the station duty would not exceed one hundred and fifty million foot-pounds per one thousand pounds of steam. With the centrifugals it would be fair to assume a duty of eighty million foot-pounds per one thousand pounds of steam; in both cases the water evaporated to be six pounds per pound of coal. One hundred and fifty million foot-pounds duty is equivalent to 13.2 pounds of steam per water horsepower-hour; eighty million duty is equivalent to 24.75 pounds per water horsepower-hour. By using 13.2 pounds per water horsepower-hour and a total head of 200 feet, the steam required is 11,232 pounds per million gallons pumped, and with 6 pounds evaporation, 1872 pounds of coal will be used, which, at $2.50 a ton, would mean that the cost of fuel is $2.34 per million gallons pumped. Also for the equipment using 24.75 pounds per water horsepower-hour, 20,800 pounds of steam are required per million gallons pumped, and with the same evaporation, 3460 pounds of coal would be used at a cost of $4.33 per million gallons.

Assuming labor, depreciation and maintenance to be the same in both cases, this gives a difference of $8100 per year in the fixed charges, in favor of the centrifugal plant, and approximately $7300 per year difference in operation, in favor of the triple-expansion installation, or a net balance of about $800 per year in favor of the less expensive plant. With an average pumpage of ten million gallons per day, this would represent a saving of $0.21 per million gallons pumped.

These figures are not intended to be accurate in all the various details, but are fairly approximate for the conditions assumed. The writer, in making estimates on cost of plants of this character, has come to the following conclusions: For large water-works plants, where the pumps will be in almost constant service, the high-duty pumping engine is the better investment. This is particularly true in districts having a high cost of fuel. On the other hand, for smaller plants and plants where the machinery is liable to be out of service more than half the time, and for auxiliary equipment, the turbine-driven centrifugal will show a marked economy over the larger and more expensive unit.

Election of officers resulted as follows: Alba L. Holmes, Grand Rapids, Mich., president; William J. Scroggins, Wheeling, W. Va., vice-president; William Allen Veach, Newark, O., secretary, and A. W. Irman, Massillon, O., treasurer.

Denver Convention of Steam Engineers

The convention of the International Union of Steam Engineers, held at Denver, Colo., during the week beginning September 12, was a success in every way. During the several business sessions of the convention many matters of importance conducive to the welfare of the organization were disposed of. The many features of entertainment which had been arranged by a competent committee were thoroughly enjoyed by the delegates and guests.

The election of officers resulted as follows: Matthew Comerford, president, St. Paul, Minn.; M. Snellings, first vice-president, Washington, D. C.; T. J. Roberts, second vice-president, Oakland, Cal.; Michael Kelley, third vice-president, Cornwall-on-the-Hudson, N. Y.; Michael Murphy, fourth vice-president, New York City, N. Y.; P. J. Finley, fifth vice-president, Pittsburg, Penn.; James J. Hannahan, secretary-treasurer, Chicago, Ill.; S. J. Stevenson, chairman of board of trustees, Chicago, Ill.

It was decided to hold the next convention at St. Paul, Minn., in September, 1911.

Belgian Exposition

In the August 30 number we published an article on the Belgian Exposition at Brussels, and in the last paragraph gave credence to a report prevalent at the time that the exposition had been practically destroyed by fire. We have since found that the early reports were greatly exaggerated. Only the Belgian section of food products, etc., and a part of the English section, all told less than a tenth part of the exposition, were destroyed. The conflagration caused but a few days' inconvenience and did not seriously interfere with the exposition.

OBITUARY

Frederick M. Wheeler, inventor of the Wheeler surface condenser, died at his summer residence at Westhampton, L. I., on September 16, of heart disease. He was born in Brooklyn in 1848 and was a descendant of an old New England family.

Mr. Wheeler was educated at the Summit Academy and the Brooklyn Polytechnic Institute and later studied mechanical engineering for four years under Henry J. Davison, of New York City, one of the leading mechanical engineers of that time. He afterward took up hydraulic and marine engineering as a specialty, and for over thirty-four years was associated with the George F. Blake Manufacturing Company. He was a director and secretary of the company, and later a director of the International Steam Pump Company, which corporation absorbed the Blake company, the Worthington and other hydraulic works. Mr. Wheeler organized the Wheeler Condenser and Engineering Company, whose works are located at Carteret, N. J., and was also officially connected with the Ludlow Valve Manufacturing Company. He was one of the charter members of the American Society of Mechanical Engineers and the Society of Naval Architects and Marine Engineers, besides being a member of the American Society of Naval Engineers and the Engineers' Club. He served as a member of the advisory council of the Engineering Congress at the Columbian exposition.

Mr. Wheeler was prominent in social circles in Montclair, his permanent home, and was the founder of the Montclair Equestrian Club, which subsequently became the Montclair and Essex County Country Club. He was also one of the founders of the Outlook Club, one of the early members of the Montclair Club, and for many years one of its board of directors. While assisting in the various public improvements from time to time, the subject which interested him most was that of public parks for Montclair, which subject he agitated for many years until it became a reality.

He is survived by a widow, two daughters and two sons.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

ELASTIC FLUID TURBINE. William James Albert London, Manchester, England. assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 969,891.

WAVE MOTOR. William L. Raht, San Diego, Cal. 969,903.

HYDRAULIC POWER SYSTEM. Thomas A. Macdonald, Clifton, N. J., assignor of sixteen and one-third one-hundredths to Isaiah E. Zimmerman, Paterson, N. J., sixteen and one-third one-hundredths to William H. Castles, Kingsland, N. J., and sixteen and one-third one-hundredths to George Lentz, North Arlington, N. J. 969,967.

TURBINE. Paul F. Nydegger, Elizabeth, N. J., assignor to the Singer Manufacturing Company, a Corporation of New Jersey. 969,974.

WAVE MOTOR. Nathaniel O. Harmon, Los Angeles, Cal. 970,048.

INTERNAL COMBUSTION ENGINE. Simon Lake, Bridgeport, Conn. 970,063.

STEAM TURBINE. Herman Peters, Hamburg, Germany. 970,106.

TURBINE. Willis G. Dodd, of San Francisco, Cal. 970,193.

GAS TURBINE. Erik August Forsberg and Birger Ljungstrom, Stockholm, Sweden. 970,204.

INTERNAL COMBUSTION ENGINE. Leon Auguste Martha, Paris, France. 970,251.

FLUID ACTUATED MOTOR. William Snee, West Elizabeth, Penn. assignor to American Power and Manufacturing Company, Pittsburg, Penn., a Corporation of Delaware. 970,404.

BOILERS, FURNACES AND GAS PRODUCERS

LIQUID FUEL BURNER. Bradford R. Phillips, Cement, Tex. 969,978.

GAS PRODUCER. William B. Chapman, New York, N. Y., assignor to Walter B. Hopping, Trustee, New York, N. Y. 970,017.

OIL BURNER. Mollie J. Wright and William L. Wright, Dunlap, Kan. 970,156.

OIL BURNER. William H. Bradley, Half Moon Bay, Cal. 970,300.

LIQUID FUEL BURNER. Jacob P. Hollenbeck, Frederick, Okla., assignor of one-half to John W. Fondren, Frederick, Okla. 970,335.

OIL OR GAS BURNER. Antone W. Andorfer, Shinglehouse, Penn. 970,363.

GAS BURNER. Ferdinand Mummelthey, Coraopolis, Penn., assignor to Graham Nut Company, Pittsburg, Penn., a Corporation of Pennsylvania. 970,382.

FURNACE. Maurice Blanchard, Wardner, Idaho. 970,417.

POWER PLANT AUXILIARIES AND APPLIANCES

GAGE COCK. John Rowland Brown, Mansfield, O., assignor to Ohio Brass Company, Mansfield, O., a Corporation of New Jersey. 969,770.

STOP VALVE. Thomas Pollock, Folkestone, England, assignor to the Dover Engineering Works, Limited, Dover, England. 969,803.

GAS PRODUCER REGULATING VALVE. Henry I. Lea, Pittsburg, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 969,884.

PRIMING CHECK VALVE. Joseph A. H. Provost, Van Anda, Texada Island, British Columbia, Canada. 969,902.

CARBURETER. Frederick John Cox, Kentish Town, London, England. 969,941.

CARBURETING APPARATUS. Frederick Ostermayer, Elizabeth, N. J. 970,102.

VALVE SPRING REMOVER. John Turner, Colorado Springs, Colo. 970,141.

FLOAT CONTROLLED VALVE. Winfield S. Bellows, Steelton, Penn. 970,167.

AUTOMATIC FUEL FEED MECHANISM FOR FURNACES. George Dinkel, Jersey City, N. J. 970,191.

METHOD OF COVERING PIPES. Eli A. Kellam, New York, N. Y. 970,235.

STOP AND THROUGH-WAY VALVE. Frederic William Leevers, Walthamstow, England. 970,245.

PUMP VALVE. George E. Mack, Detroit, Mich. 970,249.

FLUID PRESSURE VALVE REGULATOR. Thomas M. Wilkins, East Randolph, N. Y., assignor of five-sixteenths to W. E. Zierden, Johnsonburg, Penn., one-fourth to John G. Whitmore, Ridgway, Penn., and seven-sixteenths to Ziermore Regulator Company, Johnsonburg, Penn., a Corporation of Pennsylvania. 970,284.

LUBRICATING VALVE. Thomas Julian Barbre. Denver, Colo. 970,292.

FURNACE DOOR. George O'Neill, Moncton, New Brunswick, Canada. 970,347.

RELIEF OR SAFETY VALVE. William L. Osborne, Chicago. Ill. 970,348.

VALVE. Matthew H. Owen, Chicago, Ill. 970,387.

WATER LEVEL REGULATOR. Noiberto Guy Copley, Fostoria, O., assignor to the "S. C." Regulator Company, Fostoria, O., a Corporation of Ohio. 970,421.

CARBURETER FOR INTERNAL COMBUSTION ENGINES. Fitzwilliam Richard Davis, Kew Gardens, England. 970,429.

AUTOMATIC FUEL FEED MECHANISM FOR FURNACES. George Dinkel, Jersey City, N. J. 970,432.

PUMP. George J. Dourte, Longmont, Colo. 970,435.

OIL FILTER. John Francis Deems, New York, N. Y. 966,173.

WATER-GAGE GLASS. Nicholas Zucke, Sparks, Nev. 966,389.

REGULATING VALVE FOR DASHPOTS. Christian Dorn, Philadelphia, Penn., assignor to James F. Marshall, Philadelphia, Penn. 966,320.

WATER-GAGE ATTACHMENT. Zachariah C. Ferris, Caldor, Cal. 966,323.

AUTOMATIC FEED REGULATING AND MIXING DEVICE FOR CRUDE-OIL BURNERS. Joseph B. Willings, Maricopa, Cal. 966,373.

CARBURETER. Evert P. Brooks, Cincinnati, Ohio, assignor by direct and mesne assignments, of one-third to Charles A. Gibson, and one-third to John Remmers, Jr., Cincinnati, Ohio. 966,381.

VALVE. Robert W. Elder, Finch, W. Va. 966,390.

ELECTRICAL INVENTIONS AND APPLICATIONS

ALTERNATING CURRENT MAGNET. Arthur Simon, Milwaukee. Wis., assignor to the Cutler-Hammer Manufacturing Company, Milwaukee, Wis., a Corporation of Wisconsin. 969,890.

PORTABLE ELECTRIC HEATER. Frank L. Dyer, Montclair, N. J. 969,849.

ELECTROLYTIC APPARATUS. William Thum, Hammond, Ind. 969,921.

PROCESS OF ELECTRICAL SEPARATION. Henry Azor Wentworth, Lynn, Mass., assignor to Huff Electrostatic Separator Company, Boston, Mass., a Corporation of Maine. 970,001.

ELECTRIC CUT OUT. Thomas E. Murray, New York, N. Y. 970,088.

ELECTRICAL PROTECTIVE APPARATUS. Jules A. Birsfield, Rochester, N. Y., assignor to Charles A. Brown and Lynn A. Williams, Chicago, Ill., a Copartnership. 970,169.

ELECTROMAGNETIC DEVICE. William S. Burnett, Milwaukee, Wis., assignor to Morse Code Signal Company, Milwaukee, Wis., a Corporation of Wisconsin. 970,180.

PROTECTED OR ARMOR CLAD ELECTRIC SWITCH FUSE. Herbert Henry Berry and William J. Markham, London, England. 970,296.

STORAGE BATTERY. Rufus N. Chamberlain, Depew, N. Y., assignor to Gould Storage Battery Company, New York, N. Y. 970,420.

POWER-PLANT TOOLS

CUTTER FOR PIPE AND THE LIKE. John R. Hamilton, Yonkers, N. Y., assignor to Ventura Alarm Company, Dover, N. J., a Corporation of New Jersey. 969,858.

WRENCH. Alix Fresko and Bela W. Fresko, Amsterdam, O. 970,209.

RATCHET WRENCH. Ambus M. Fulkerson, Sac and Fox Agency, Okla. 970,320.

WRENCH. Uriah C. Greggs, Terre Haute, Ind. 970,371.

DRILL. Charles Seals, Lovilla, Iowa, assignor of one-half to William Livix, Lovilla, Iowa. 970,401.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

Vol. 32 NEW YORK, OCTOBER 11, 1910 No. 41

IT has been solemnly stated that once a week is too frequent for the visitations of POWER. Fifteen or twenty years ago there might have been some ground upon which to base this assertion. However, the remarkable progress which has been made during the last decade or two in all branches of the science of power has served to remove all grounds for objections to a weekly power paper.

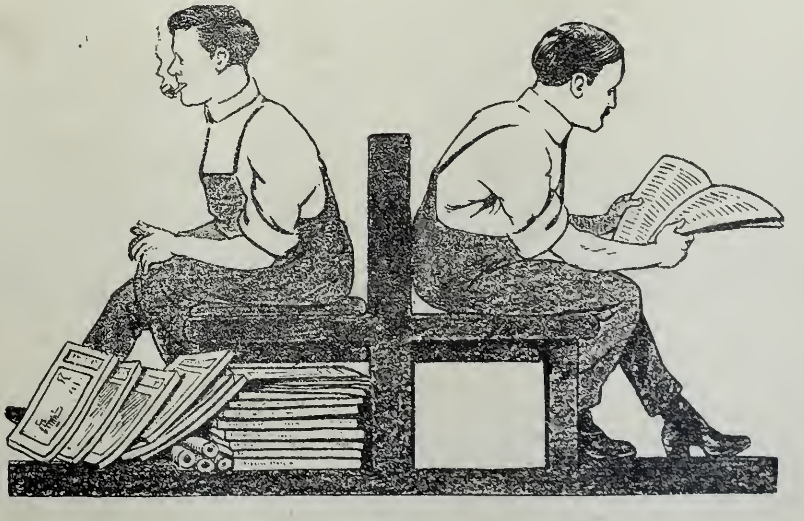
* * * * * *

When journalism was in its swaddling clothes the proper frequency of publication was about once a month. The editions usually consisted of about four pages, say, eight by fourteen inches in size. The size and the interval exactly fitted the times. News could only be gathered by stage coaches. But even if it could have been accumulated by more expeditious means it would have taken so long to set it in type and print it that the news would still be stale. Furthermore, transportation being difficult, people did not move often or far; hence, except in the cases of a few national events, such as a war, news items had very little other than local value.

Imagine, please, a newspaper of today which contained only four pages a little larger than a postage stamp in size and which appeared only once a month. Such a paper would be a farce.

Cheap and comfortable transportation coupled with cheap means of quick communication has served to diffuse the population and amalgamate its interests, so that many otherwise purely local events are of widespread interest and have real news value. Great progress in the mechanical sciences has made the speed of publication keep pace with the demands of the times. As a result we have great daily newspapers which issue anywhere from four to ten editions a day.

* *



A very similar development has taken place in the power field.

You can easily figure out how long a power paper would have lasted in the time of Watt. About as long as a snowball in a furnace. There was no practical field for such a paper then, and, consequently, no demand. Even a paper which came out only once a year would have been superfluous.

* * * * * *

About 30 years ago, when electricity was still in its inception, a paper devoted to steam engineering was brought out to meet an embyro demand for news and information relative to the science of power generation. It was issued once a month. The average edition contained about ten reading pages. There was no need for more for the field was not large, judged by the standard of today, and practically only steam and water power were used.

* * * * * *

But, today, with the vast application of power in a great variety of forms, the field is enormous and progress is rapid. Of necessity the engineer of today must know far more than the engineer of those days. He must keep pace with the times.

To do this the one best way available is to follow the current literature which deals with the latest achievements, tells of what is being done and how.

The progressive man devours all the information he can get and, like Oliver Twist, wants more. To him the weekly consignment is most welcome.

The weekly idea is good for the less ambitious, too. As it comes in homeopathic doses he is likely to retain some each week and in the end have more than if his "medicine" were administered in such large monthly potions as to discourage him completely. In other words, he is less liable to get mental indigestion.

Equipment of the Blackstone Hotel

One of the latest architectural additions to Chicago's downtown district is the Blackstone hotel, situated at Michigan avenue and Hubbard court. This structure is twenty-three stories high, and unusual attention has been paid to the exterior decorations, with the interior appointments in keeping with the impression gained from the outside.

The ground space is unusually restricted for a building of such hight, necessitating much ingenuity in the layout of the mechanical plant, which is of the most complete character. The foundation measures only 80 feet on Michigan avenue by 173 feet on Hubbard court, the building resting on thirty-four caissons, each 11 feet in diameter, filled with concrete and sunk 100 feet to bed rock. These caissons are tied together by interlocking concrete-covered steel beams, forming the foundation into one solid supporting structure, but making the problem of suitable foundations for the machinery a very difficult one. Owing to the restricted space available, all the equipment has been placed in one room, the boilers, engines, pumps and other auxiliary machinery having been placed as close together as possible and still allow room for convenient operation. The ground-floor plan, shown in Fig. 1, indicates to some extent the compactness of the installation.

The coal, which is delivered in wagons at the Hubbard court side of the building, is discharged into a bin which is located under the sidewalk, and which has a capacity of 175 tons. Facing this coal bin are the boilers. four in number, each of the Stirling type and rated at

By Osborn Monnett

> *Owing to the unusually small area covered by this hotel in proportion to its hight, special attention had to be given to the layout of the power plant in order to insure an equipment which, although complete in every respect, would require as little floor space as possible. Elaborate systems of refrigeration and ventilation have been installed and a complete record is kept of operating details.*

250 horsepower. They are set separately with a building column between each setting. Each boiler is equipped with a McKenzie chain-grate stoker, driven by an individual 1½-horsepower motor. Fig. 2 is a front view of the boilers, showing the stokers. Coal from the storage bin is fed to a one-ton Howe traveling, weighing hopper, which distributes it to the stokers as required. The uptakes from the boilers pass over the engine cylinders to the steel stack which is located in the northwest corner of the building. The stack is oval in section, and extends to a hight of 328 feet above the grates.

Williams feed-water regulators are fitted to the boilers and the feed lines, which are of extra-heavy pipe, are entirely in duplicate up to the point where the feed enters the rear upper drum. There are two 12 and 7½ by 12-inch Marsh boiler-feed pumps and, in addition, it is possible to feed the boilers by means of four 12x16x9¼x16-inch tandem - compound American pumps which are ordinarily used to supply water for general purposes. The feed water passes through a Venturi meter before entering the boilers, the indicating and recording gage being located in the engineer's office.

The ashes are removed to a storag bin, located under the alley at the back o the hotel, by means of a pan conveyer from there they are elevated to wagon and carted away.

The hydraulic-elevator equipment con sists of two 12x20x11x20-inch Ameri can tandem-compound pumps, deliverin water into a compression tank at 12 pounds pressure. This pressure sup plies the motive power for one sidewal hoist with a lift of 20 feet, one kitche elevator, with a lift of 66½ feet, and on ball-room elevator with a 48-foot lift, a operating at a speed of 200 feet per mi ute. A general view of these pumps shown in Fig. 3.

PIPING

Leading from each boiler there is 6-inch and a 4-inch steam header, eac fitted with two stop valves and a non turn valve, all in series. The headers looped together and cross-valved to m any emergency that might arise. T 12-inch main header extends along center of the engine room just above

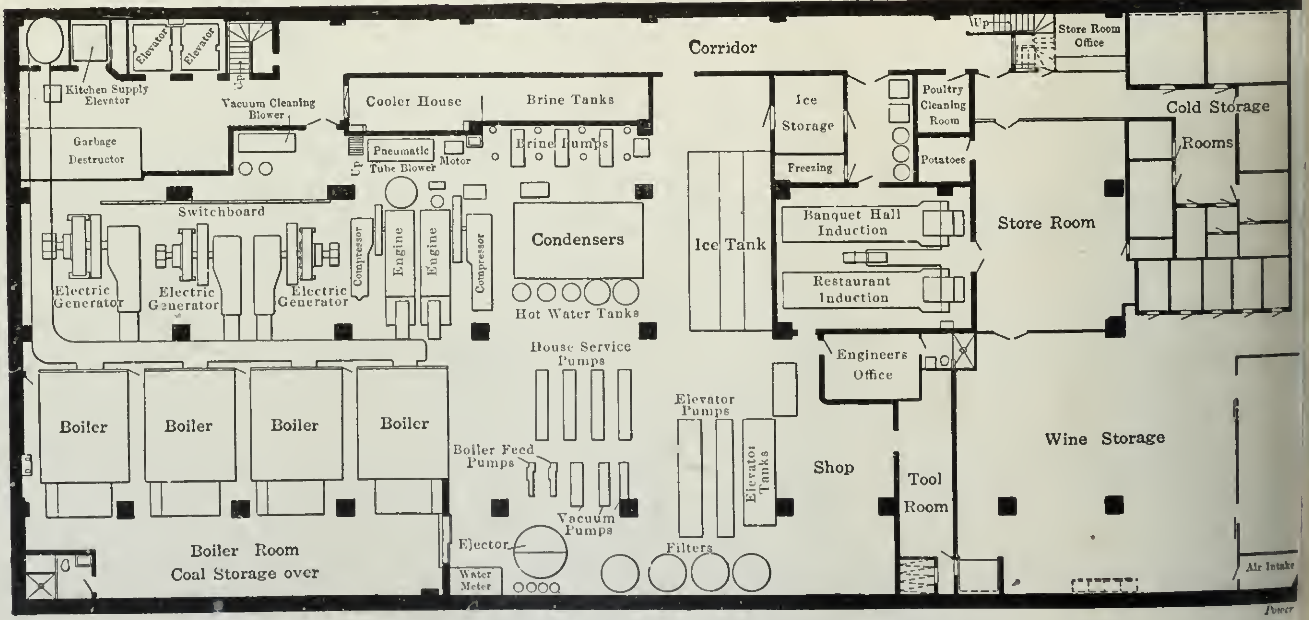


FIG. 1. PLAN OF POWER PLANT IN BASEMENT OF BLACKSTONE HOTEL

ngines and the branches from each boiler drop into this. From the top of this eader the leads rise to the various engines, and a separator is fitted above each hrottle valve. All high-pressure piping s extra-heavy with welded flanges, and Nelson valves are used on all high-pres-ure steam lines. Morehead tilting traps ke care of the condensation on both e high- and the low-pressure piping.

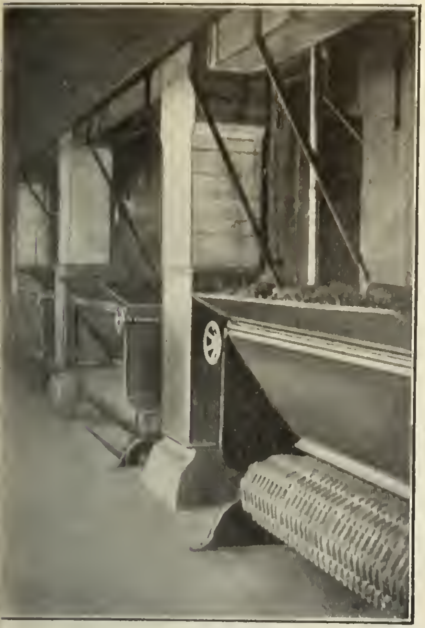


FIG. 2. BOILER ROOM

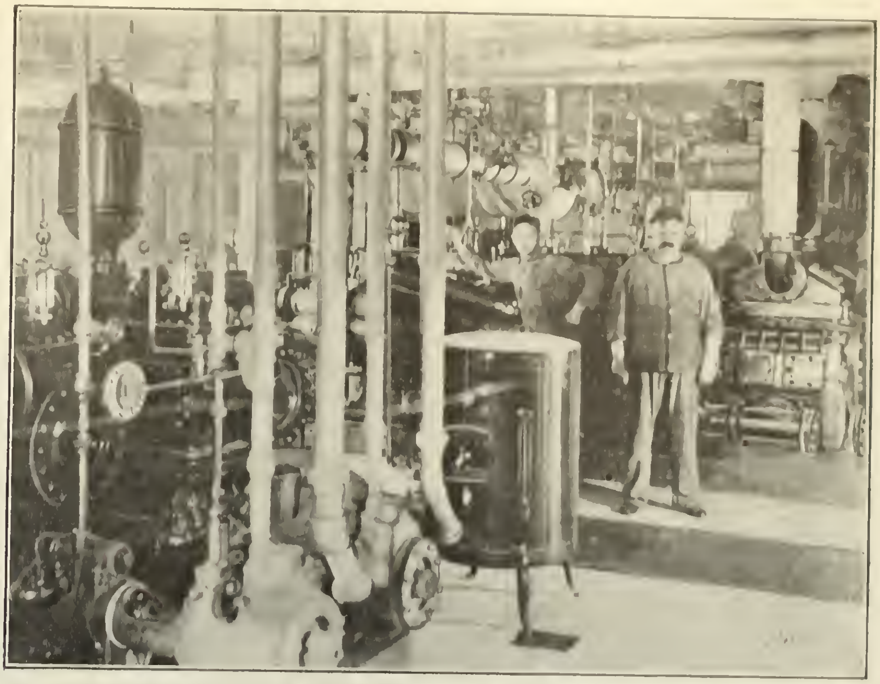


FIG. 3. PUMPING EQUIPMENT OF BLACKSTONE PLANT

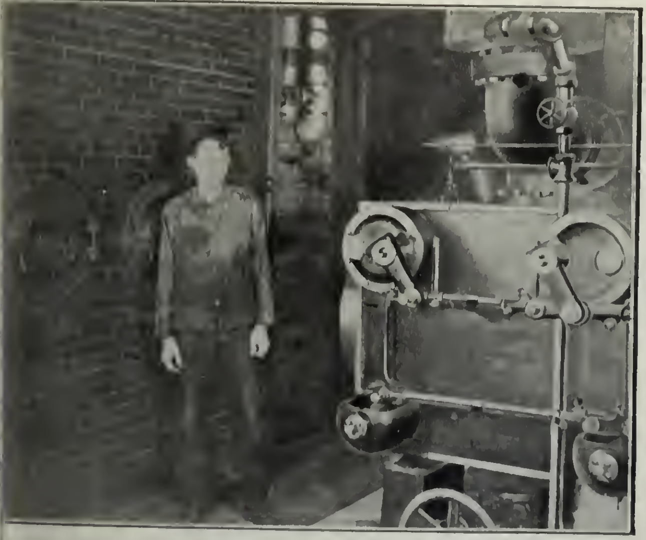


FIG. 4. SHOWING CLEARANCE SPACE BETWEEN ENGINES AND BOILERS

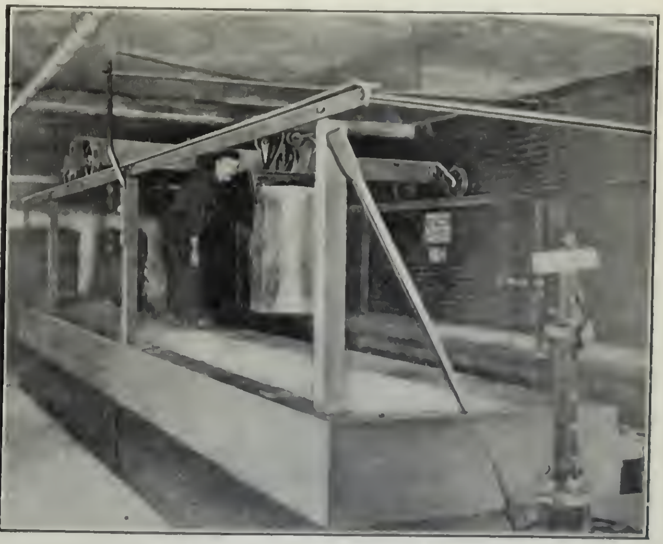


FIG. 5. HARVESTING PLATE ICE MADE BY THE JEWELL SYSTEM

ENGINES AND GENERATORS

The engines are arranged in a row di-ctly behind the boilers and as close to em as is possible with allowable room r taking out the pistons. Fig. 4 gives comparative idea of the distance be-een the cylinder heads and the backs the boilers. There is one 22x27-inch Ball four-valve engine connected to a 300-kilowatt generator and two 18x27-inch Ball four-valve engines, each connected to a 200-kilowatt generator. Each engine is fitted with an individual gravity oiling system, the pump for which is operated from an extension arm on the crank pin of the engine. Richardson-Phœnix force-feed pumps supply oil to the cylinders, the oil being introduced at two different points over the steam valves.

The generators are all of the Western Electric Company type and deliver current on the three-wire system at 220 volts for the power circuits and 110 volts for the lighting circuits. The switchboard consists of thirteen panels of Tennessee marble and is 34 feet long. The current is measured on the power panels as it comes from the generators and is also measured at the individual power and lighting panels. The feeders are protected by double-pole circuit-breakers and there are no switches on the board, the circuit-breakers serving this purpose. The power is used to run four passenger and two service elevators of the Otis type, laundry and kitchen motors, motors driving the ventilating fans, the machine shop, and the centrifugal pumps; besides this there are a number of motors for various purposes, making from 75 to 80 motors installed. There are about 9000 lamps in use, and the average electrical load for the month of August amounted to 3530 kilowatt-hours per day.

Exhaust Heating System

A 16-inch exhaust main extends under the floor to a point near the stack, from which point all of the exhaust steam for heating the building is controlled. A 16-inch exhaust riser extends to the roof and terminates in a Swartwout exhaust head; a 16-inch back-pressure valve is also located in this line and the heating is accomplished on the up- and down-flow system. A 12-inch heating main carries the steam to the upper floors where it divides into two 8-inch branches from which the steam flows down to the radiators in the various rooms, supplying in this way a total of 35,000 square feet of direct radiation. The returns are handled by the vacuum-return system for which purpose three Marsh pumps are installed in the basement and maintain a vacuum of about 7 inches. A live-steam connection is supplied to aid the heating system in case of a scarcity of exhaust steam. By this means the steam passes first through a Collins reducing valve which reduces the pressure from 150 to 50 pounds and then passes through a 12-inch Locke hydrostatically operated reducing valve, which delivers the steam at one pound pressure. Live steam is also stepped down through Collins reducing valves to 80 pounds for use in the laundry and to 30 pounds for use in the kitchen. Connected to the exhaust system is one Wainwright feed-water heater for the boilers and three Baragwanath heaters for the house service, the latter being controlled by a thermostat which maintains a temperature of about 150 degrees Fahrenheit.

Refrigerating Equipment

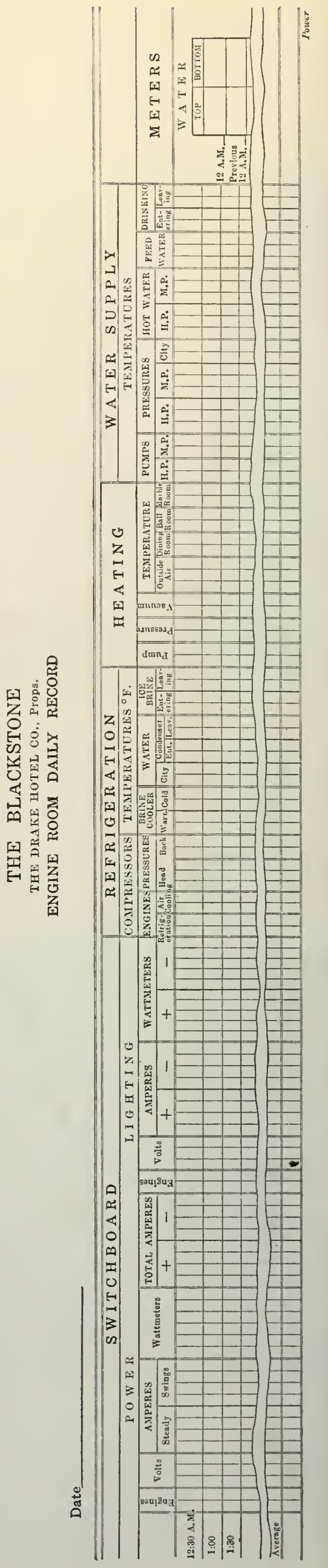
Among the important features of the equipment are two Kroeschell carbon-dioxide ice machines, one an 80-ton machine used for cooling the air supply to the various public rooms, and the other of 50 tons capacity, used for making ice and cooling various ice boxes throughout the building. The compressors are of the regular CO_2 type, operating in conjunction with a double-pipe condenser and are driven by Murray engines. In the former service air is taken at a point 60 feet above the street and is conducted to the basement where it is passed through coils which heat it to 45 degrees in the winter, and thence through a water spray which removes any impurities. In the summer time the air is then passed through the cooling coils and finally through the eliminator, which removes all excess moisture, thence to the various public rooms in which a predetermined temperature is maintained. On one of the hottest days during the past summer, with the outside air at 106 degrees, the public rooms of the hotel were maintained at a temperature of 74 degrees without trouble.

The 50-ton ice machine cools an elaborate system of ice boxes in the kitchen, service pantries and basement. In addition to this service, eight tons of ice is made daily on the Jewell system, that is, making plate ice in cans. This is one of the first installations of its kind and is worthy of some attention. With this system it is feasible to make clear ice without distilling or reboiling the water, and freeze 400-pound cakes within 24 hours with brine at about zero degrees.

The cans are provided with a jacket around which the brine circulates, and in the bottom of each can there is a non-freezing zone consisting of a shallow rectangular pocket in which is introduced a jet of refrigerated air. This keeps the water agitated at this point and any impurities are deposited here. When the cakes are frozen the jet of air, which is supplied at very low pressure, is automatically shut off by the ice. In harvesting, the supply of brine, which is handled by a 6-inch motor-driven centrifugal pump, is shut off from the brine cooler and circulates through a coil in a tank containing the water for the next freezing; this cools the water and warms the brine, thereby thawing the ice from the cans. The new supply of water is fed to the ice cans through a float-regulating valve so that when the ice is loosened in the can it floats and the can is automatically filled with fresh water after the cake has been removed. A compressed-air hoist handles the ice and delivers it to the store room, as shown in Fig. 5. This system of precooling the fresh water and using the heat given up by the water to thaw the ice from the cans, utilizes heat which is ordinarily wasted and is said to about balance the loss occasioned by the introduction of refrigerated air into the bottoms of the cans.

Water System

The building is supplied by two 6-inch mains in which Worthington turbine meters are installed. All water for the building passes through International quartz filters before being used and then passes to the feed pumps, house pumps, etc. In addition to passing through these filters, the water used for drinking and freezing is again filtered in International disk filters. Water for house use is supplied on a high- and medium-pressure system. The high-pressure system consists of a tank on the roof supplied direct by one of the house pumps under a static pressure of approximately 135 pounds; from this tank the water is distributed to the different floors of the building down to and including the seventh floor. Hot water for the high-pressure system is supplied from this tank and passes through one of the Baragwanath heaters. The medium-pressure system is supplied by one of the house pumps at 90 pounds pressure, the pump delivering into an expansion tank located on the line. This system supplies both hot and



THE BLACKSTONE
THE DRAKE HOTEL CO., Props.
ENGINE ROOM DAILY RECORD

Date

	SWITCHBOARD													REFRIGERATION											HEATING							WATER SUPPLY									
	POWER							LIGHTING						COMPRESSORS				TEMPERATURES °F.																			TEMPERATURES				
			AMPERES			TOTAL AMPERES				AMPERES		WATTMETERS		ENGINES		PRESSURES		BRINE COOLER		WATER	Condenser		ICE BRINE					TEMPERATURE				PUMPS		PRESSURES			HOT WATER		FEED	DRINKING	
	Engines	Volts	Steady	Swings	Wattmeters	+	−	Engines	Volts	+	−	+	−	Refrigeration	Air Cooling	Head	Back	Warm	Cold	City	Ent.	Leav.	Entering	Leaving	Pump	Pressure	Vacuum	Outside Air	Dining Room	Ball Room	Marble Room	H.P.	M.P.	H.P.	M.P.	City	H.P.	M.P.	WATER	Entering	Leaving
12:30 A.M.																																									
1:00																																									
1:30																																									
Average																																									

METERS

	WATER	
	TOP	BOTTOM
12 A.M.		
Previous 12 A.M.		

Power

Chief Engineer

Fig. 6. Daily Record Sheet Upon Which Readings Are Entered Every Half Hour

cold water from the ground floor up to and including the sixth floors. For the basement floors city-water pressure is relied upon.

System of Records

A complete record is kept of all operating details, the daily report blank of which is shown in Fig. 6. The daily totals are then posted on a monthly report sheet which gives a complete record of the month's run in such shape that a clear idea may be had of the character of the service in every department.

In every large office building or hotel the manner of handling complaints is of great importance. Frequently it is necessary for the protection of the engineering department that a proper record of these complaints be kept and it is also a satisfaction to the management to know, just what has been done and who is responsible. In the case of the Blackstone hotel there has been worked out a very satisfactory scheme by which complaints are handled. Complaints are received at the engineers' office by telephone, verbally, by means of the telautograph, or by written messages through the pneumatic-tube service. When a complaint is received the name of the person making the complaint is entered on the sheet with a complete description of the difficulty and the name of the workman to whom the repairs are delegated. This sheet is then stamped with the chief engineer's name and the time received, and the other half of the sheet is delivered to the workman with a description of the job to be done, and whom to see in regard to it. All the material used on the job is entered and the time noted when the work is completed. This is then signed by the workman and delivered to the chief engineer's office. The two portions of the sheet bear the same serial number and are filed together, constituting a complete record of the transaction.

Stopping a Troublesome Pound

By R. O. Warren

A dull, heavy pound occurring once in every dozen revolutions puzzled the engineer for a number of days. He was at his wits' end when a friend stepped in and by mere chance located the trouble.

I had finished a long, hot, twelve-hour run, such as befalls the lot of most central-station operating engineers, and was on my way home, when I heard a call coming from the direction of the King Cycle Manufacturing Company.

I stopped and spied George Hicky swinging his arms like a windmill as he beckoned me to come over to his engine room.

"Did you ever have a pound, a pound that was out of step with everything about the engine?" he asked, before I had a chance to speak.

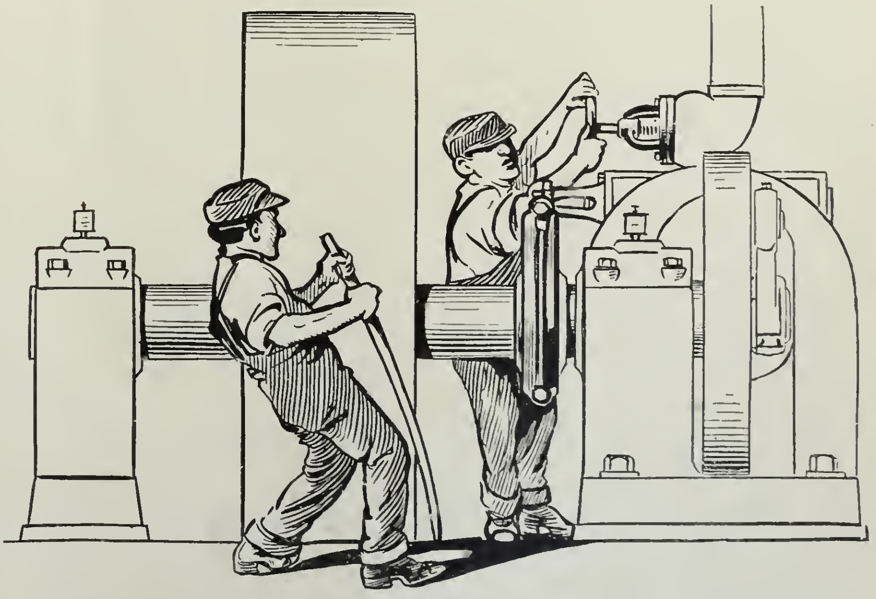
"Come on in," he exclaimed, as he grabbed me by the coat sleeve. "I'm all befuddled and about as near crazy as I ever will be. I've had a pound in my engine for about a week and can't locate it. I'm ——"

"Now go slow," said I, "and let's take things easy. Have a cigar, and just rest yourself in your old soft-seated chair. That cigar will soothe your nerves." When we had gotten our cigars going, he began again.

"It's the dundest thing I've ever run up against. I had an idea that I could find a pound in an engine about as quick as the next one, but this one has got me stumped. My night engineer is as much puzzled as I, and both of us have been hunting that pound, day and night, for a week. Listen—did you hear that?"

I did; there was no mistaking the dull, heavy thud that occurred at regular intervals, but at no particular point of the engine stroke. In fact, the engine would make about a dozen revolutions before the pound would be repeated. This made the matter all the more puzzling and might have baffled my friend for a few days more if the cause had not been discovered by chance.

We had left our chairs and I was standing facing the cylinder on the flywheel side of the engine. Fortunately, I happened to look at the flywheel when it appeared to move sideways, accompanied by the thud that had been causing so much worry. Keeping my eyes fixed on the flywheel, I waited until the thud was repeated, which was accompanied by a side movement of the flywheel.



As the Engine Slowed Down, the Fireman Utilized His Strength on a Scantling

A close examination disclosed the fact that the eccentric and distance collar that was placed on the shaft between the eccentric and the main pillow block had worked over on the shaft about ¼ inch. Every so often the shaft would work into its proper place with the crank against the pillow block. The shaft would run in this position for a time and would then slide through the bearing until the collar and eccentric brought it up with a thud.

When the engine was shut down for oiling ready for the night run, a piece of scantling was brought to bear on the rim of the flywheel, with one end in the wheel pit. While George shut down, the oiler bore against the outer end of the scantling and forced the shaft over in the bearings so that the crank came against the pillow block. The eccentric was then moved over on the shaft and secured in that position. When the engine was started up the pound had disappeared and so far as I know has never returned.

As I started out of the engine room George remarked, "It don't take much to keep a man guessing, does it?"

The Zeuner Valve Diagram

By W. J. Harden

A convenient and simple means of studying the relative positions of the valve and main crank during one revolution is offered by the Zeuner diagram. This is familiar to many, but for the benefit of those who have never used it, or those whose conception of it is hazy, a simple explanation is presented.

The Zeuner valve diagram, by means of which the valve displacement can be found for any position of the main pin, is a simplification of longer processes to obtain the same result, and its construction can best be studied by showing how this is done. Assuming an engine whose valve has no lap, as in Fig. 1, it is evident that $A'E$ and $A''E'$ represent the distance the center of the valve has moved from its mid position when the crank pin is at C' and C'' respectively. With the pin at C, the valve will be on the center of its seat. While the valve displacement for any position of the pin could be found by drawing the main crank and the crank representing the eccentric at right angles to each other, and then drawing a perpendicular to the vertical line AB, or AB produced, yet a simpler method of obtaining the same result is shown in Fig. 2. The valve displacements for the positions C' and C'' of the pin will be $A'E$ and $A''E'$. From A draw AF and AF' perpendicular to $C'B$ and $C''B$. Considering the two triangles AFB and BEA', the angle FAB is equal to the angle ABA', as AF is parallel to $A'B$. Also the angle AFB is equal to the angle BEA', as both are right angles. The side AB is equal to the side $A'B$, both being radii of the same circle. The two triangles are therefore equal and the side FB is equal to the side $A'E$. But $A'E$ represents the valve displacement when the crank pin is at C'; therefore FB also represents the valve displacement when the crank pin is at C'. It is then evident that the valve displacement for any position of the pin between C and D can be found by dropping a perpendicular from A to the line representing the main crank, while for the return stroke of the piston the corresponding valve displacements can be found by drawing the perpendiculars from the point A'.

The diagram has then been simplified to the extent of eliminating the line representing the valve crank, and the perpendicular from the extremity of this line to the vertical line AB; in other words, the valve displacement can now be found by dropping a perpendicular to the main-crank position and noting the intercept between the center and the point at which the perpendicular falls. However, the diagram can be still further simplified, and it will be shown next how the perpendicular to the main-crank position can also be eliminated.

Fig. 3 represents the main pin in three positions, C, C' and C'', and the corresponding valve displacements, FB, $F'B$ and $F''B$ have been measured off on the main crank by the perpendiculars AF, AF' and AF''. The three right-angled triangles, AFB, $AF'B$ and $AF''B$ are all described upon the same base AB. It can be proved by geometry that when a series of right-angled triangles are constructed upon a common base, their vertices all lie upon the circumference of a circle, the diameter of which is formed by the base line of the triangle. If a circle be described with such a diameter, the part of the line representing the main crank, which is intercepted between the center of the axle and the circumference of the circle, will then be the valve displacement for that position of the main pin. This circle, known as the valve circle, then renders unnecessary the drawing of a perpendicular to ascertain the valve displacement. The lower valve circle described upon the position of the eccentric radius $A'B$ with the pin at G gives the valve displacement for the return stroke of the piston. This completes the construction of the Zeuner valve diagram, considering an engine having a valve without lap.

Fig. 4 shows the diagram applied to a valve with lap, but with no lead, HON being the angle of advance of the eccentric. When this diagram is applied to a valve having lap, the angle of advance must be laid off negatively if a clockwise rotation of the main crank is to be considered, and the valve circles described, using as diameters the valve crank MO or MO produced to M'. If a negative angular advance is not assumed, that is, if NO or NO produced to N' is taken as the diameter of the valve circle, a negative rotation of the crank will have to be assumed. In this case, in order to find out how far the valve has moved from mid-position when the main crank has moved through a certain angle from one center, an imaginary crank must be assumed rotating through the same angle in the opposite direction from the other center. In order to avoid this a negative angular advance is assumed, although the diagram is sometimes found with the circles described on the positions the valve crank would assume in actual practice with the engine on either center, a negative rotation of the crank then being necessary. A negative rotation of the main crank assumes that the eccentric is fixed independent of the crank shaft, the whole engine revolving around the center of the axle, the direction of motion being such as already stated. Having described the valve circles, using MO and OM' as diameters, next, from the center O describe a circle with a radius equal to the lap. With the pin at A, the valve displacement is equal to OR, but as OR is equal to the lap, the valve displacement will be equal to the lap, or the valve will be at admission. With the main pin at B, the valve displacement is equal to

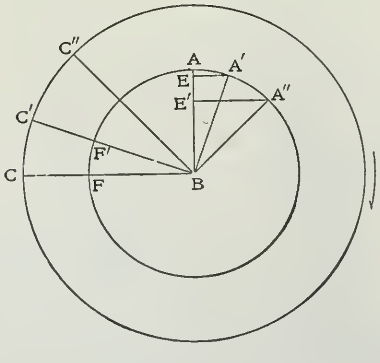


FIG. 1

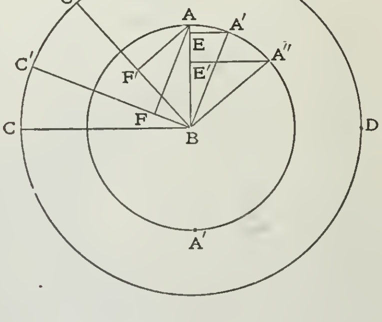


FIG. 2

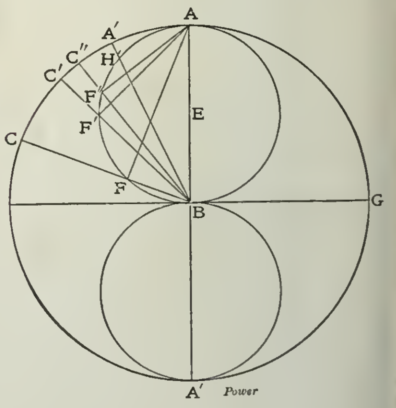


FIG. 3

M O, and the port opening equal to *M M″*, this position of the pin giving the greatest port opening. At *C* the displacement of the valve is again equal to the steam lap *O C′*, and the valve is at cutoff. At *D* the displacement is zero, no part of the main crank being found in either valve circle, the pin now being in mid-position and release about to occur at the left port and compression at the right port. With the pin at *A* the valve displacement is equal to the lap and the right-hand steam port will be on the point of taking steam. The left port will be open to the exhaust, an amount equal to the steam lap at this time. The lower valve circle shows the events for the return stroke of the piston. At *M′* the right-hand steam port is wide open, cutoff occurring with the pin at *E*, and release and compression with the pin at *F*.

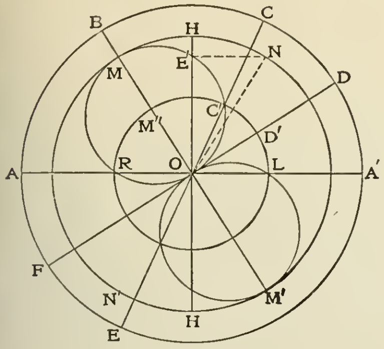


FIG. 4

Fig. 5 shows the diagram applied to a valve having lead. With the pin at *A″* the valve displacement is equal to the lap *O C* and the valve is on the point of opening the port. With the main pin at *A*, the valve displacement is equal to *L O*, and subtracting the lap *R O* leaves a port opening *L R* which is the extent of the lead. Cutoff occurs at *E;* if no lead existed this event would occur at *F*. Release and compression take place at *G*, and if no lead existed these would occur with the pin at *H*. With the pin at *A′* the port opening would be equal to the amount of lead *K I*. Considering the return stroke of the piston, the greatest port opening would occur with the pin at *M′*.

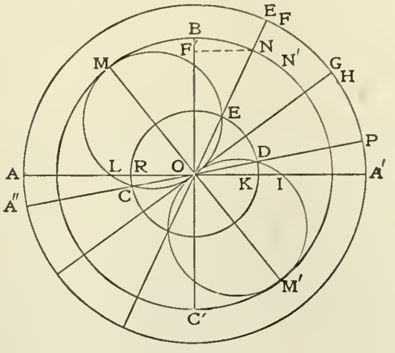


FIG. 5

Fig. 6 shows the diagram applied to a valve which has lead and exhaust lap, the inner circle being described with a radius equal to the exhaust lap. With the pin at *F*, compression is about to occur at the right-hand steam port. The valve will be in mid-position when the main pin is at *G* and release will occur at the left-hand steam port when the main pin is at *H*. With the pin at *F* and *H* the valve displacement is equal to *O F′* and *O H′;* that is, it is equal to the exhaust lap. Considering the inner circle as being described with a radius equal to the negative exhaust lap with the main pin at *F*, the valve displacement is equal to the negative exhaust lap and release will be about to occur at the left steam port, and compression at the right port with the main pin at *H*. Cutoff occurs at the left port with the pin at *E*. The valve events for the return stroke are shown in the lower part of the diagram.

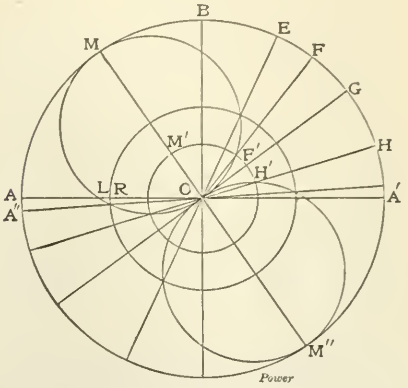


FIG. 6

Leakage through a Piston Valve

By George Mitchell

> *A significant fact, although one often lost sight of, is the enormous proportions assumed by the leakage past the piston valve of a steam engine; it reaching, in small engines, as high as 22 per cent. of the total steam consumption. When one stops to consider that this represents energy lost, its importance will at once be comprehended. The method herein described was devised by the author to measure this leakage under actual running conditions.*

Obviously, valve leakage occurs because the valve does not fit perfectly into the cage in which it runs; that is to say, the difference in the diameters of the valve and cage is sufficiently great to allow a sensible amount of steam to escape between them. Though it might be possible to construct a valve with a fit so perfect that it would run with a small amount of friction and yet allow a negligible quantity of steam to escape through it, such is practically never the case; because if a valve of this type were made as above stated, it would probably be in such a condition only when the pressure of the steam was such that the expansions of the valve and cage were just of the right amount to produce the desired fit. Therefore, it is reasonable to suppose that some leakage would occur through any valve of the piston type, and it was the object of this experiment to determine as nearly as possible the amount of this leakage.

The subject of valve leakage is very important because through the existence of any leak between the steam chest and exhaust or between the cylinder and exhaust, or through the escape of steam in any way through the valve without doing the work which it is capable of doing, a certain amount of energy is lost, and the efficiency of the engine is reduced in direct proportion to the amount of this loss.

Many schemes have been devised for measuring the amount of leakage, but in general they are crude and for many reasons the results cannot be regarded as accurate. It is not necessary to enumerate the defects of these various methods; it will suffice to say that the most important defect common to all of them is that the leakage is not measured under normal running conditions, and erroneous assumptions are made regarding the leakage being the same when the engine is stationary as when running at normal load and normal speed. In deciding on a method for determining the leakage, the rule was strictly adhered to, that the leakage should be measured under running conditions that were as nearly normal and with as few assumptions as possible.

The tests were made at the mechanical laboratory of the University of Pennsylvania on a 6x6-inch simple engine equipped with a centrally balanced centrifugal governor and with a piston valve without rings, operating in removable bushings.

It was decided after due deliberation that it would be impossible to make the tests by using the standard valve supplied with the engine, because its length was such as to preclude all possibility of separating the leakage from the steam in the cylinder, and on account of the exhaust passage through its interior, it would not be possible to separate the leakage from the exhaust. Further, if the standard valve had been used the leakage could have been determined only through one linear distance, that being the length of the valve face.

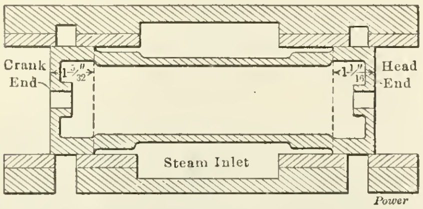


Fig. 1. Standard Valve in Mid-position

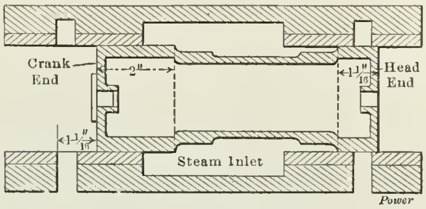


Fig. 2. Special Valve in Mid-position

The method to be used for determining the leakage was finally decided upon as follows: That a special valve should be made by the makers of the standard valve so that as far as its fit was concerned, it should be similar to the original valve. The head end of the valve was to be the same as the standard head end, but the crank end was to be made much longer so as to permit of being cut down to various lengths, through which the leakage could be measured. The total length of the valve was to be made much shorter than the standard length, this being done to permit the crank end to run only on the bushing on the inner side of the steam ports. It is now evident that if the engine were operated single acting by means of the head end, and a pipe were connected to the crank end of the cylinder, that when the engine was in action, all the steam coming through this pipe would be, if the exhaust passage of the valve were closed, the result of leakage through the valve and through the piston; and if the piston leakage were subsequently determined and subtracted from the total leakage, the result would be the valve leakage.

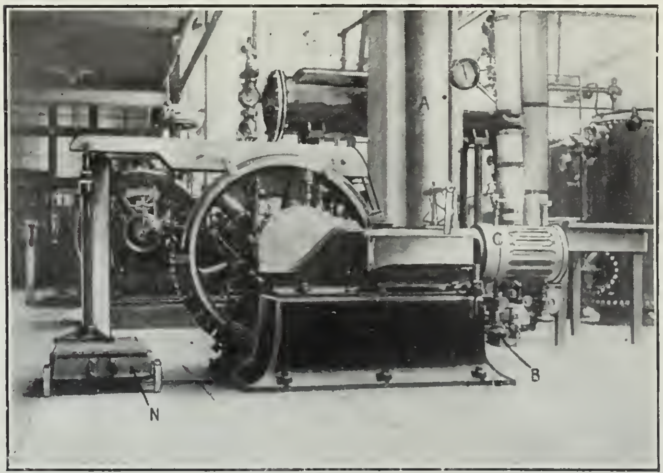


Fig. 3. Engine Rigged for Testing

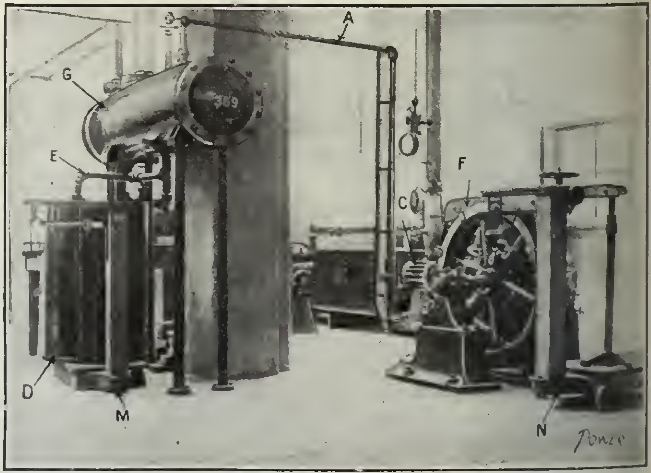


Fig. 4. Rear View, Showing Measuring Apparatus

Fig. 1 shows a cross-section of the original valve in its mid-position, while Fig. 2 shows a cross-section of the special valve and bushing in its mid-position. It will be noticed that the only difference between the standard valve and the special valve is that the length of the crank end of the special valve is 2 inches while that of the standard valve is $1\frac{5}{32}$ inches. The total length of the special valve is $7\frac{3}{16}$ inches while that of the standard valve is $8\frac{13}{32}$ inches; also the exhaust passage in the special valve has been closed. Except for the above noted changes, the special valve is exactly similar to the standard valve. When the special valve is in mid-position, the distance it would have to move to close the crank-end steam ports is $1\frac{1}{16}$ inches, and as the maximum valve travel is 2 inches, it is evident that the crank-end ports will never be closed.

The general arrangement of the apparatus and the connections are shown by the photographs, Figs. 3 and 4. Pipe *A* is connected at one end to the relief valve *B*, on the crank end of the cylinder *C*. At the other end it is connected to the condenser *G*. The tank *D* is supported by the platform scales *M* and is placed immediately beneath the condenser-discharge pipe *E*. The scales *N* and prony brake *F* are for determining the brake horsepower of the engine on which the test was made.

The method of procedure for determining the valve leakage was as follows: It was first necessary to design the special valve referred to above, and for this purpose the standard valve was removed and measurements were taken of the valve and bushing. The standard valve was now replaced and the length of the valve rod was adjusted until the best possible indicator cards were obtained with the engine running at 15 horsepower, which was its normal rating. Next, the valve travels at various loads from 2.5 to 15 horsepower were determined by measuring directly the travel of the valve rod when the engine was running at the above loads, and a curve was plotted from these measurements. From this data the special valve was designed as shown in Fig. 2.

The steam consumption of the engine when running at various loads from 5 horsepower to 15 horsepower was next determined, the engine exhausting into the condenser and the condensed steam being collected in tank *D* and weighed. A curve of steam consumption at the various horsepowers was then plotted.

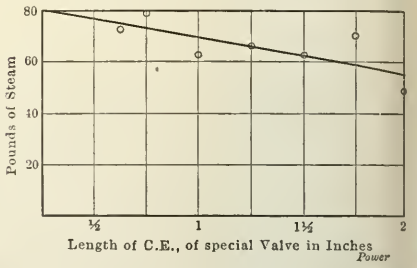


Fig. 5. Leakage with Various Lengths of Valve

The test for leakage through the crank end of the special valve was then begun. The standard valve was replaced by the special valve and the valve rod adjusted so that the head end of the special valve was in exactly the same position as the head end of the standard valve when in its best position, as previously determined.

The throttle valve was now opened and the engine run single acting by using the head end. The leakage traveled through the crank end of the valve, entering the crank end of the steam chest, thence through the crank-end steam ports into the crank end of the cylinder, and inally through pipe *A* into the condenser vhere the steam was condensed and veighed. After the engine had been unning long enough for the leakage to ecome constant, a one-hour run was begun, observations being taken at the beginning and end of the run, and at intervals of ten minutes as follows: Steam ressure, valve travel, revolutions per ninute, reading the scales *M* and also an ndicator card for the head end of the ylinder. Six other runs were made in . manner exactly similar to the above, hese runs being made with the length f the crank end of the valve 1¾, 1½, ¼, 1, ¾ and ⅝ inch respectively. he length of the crank end was shortned between each run by removing the alve from the steam chest and cutting ff the inner edge of the crank end of he valve to the proper length, the total ngth of the valve remaining the same.

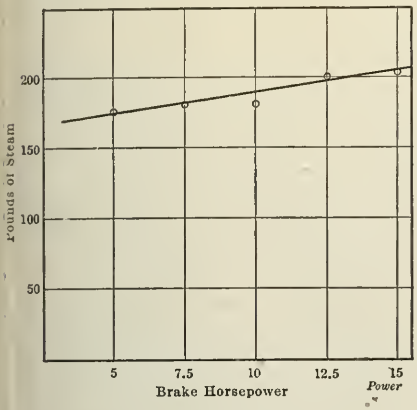


Fig. 6. Actual Leakage at Various Horsepowers

As the measured leakage included the akage through the piston, it was necesary to measure this amount so that the et leakage through the special valve ould be determined. The leakage through he piston was determined as follows: he special valve was removed and the eam ports at the crank end were plugged ith wooden plugs which were accurately tted and driven in. Holes were drilled the crank end of the special valve, hus opening a continuous passage hrough the interior of the valve, and the alve replaced within the steam chest. he engine was blocked in such a posion that the steam ports at the head end ere uncovered so that the steam could ave free access to the head end of the linder, and the crank end of the valve d not reach as far as the plugged steam orts. It is plain that the pressures on ach side of the plugs were practically hat of the atmosphere so that the tenency for leakage to occur was reduced a minimum. All other connections ere allowed to remain as before so that e leakage occurring was due to that hrough the piston alone. Half-hour runs ere made with the piston in various arts of the stroke, observations being ade at the beginning and end of the run and at intervals of 10 minutes as follows: Time, reading the scales *M* when balanced and the steam pressure.

A far more satisfactory method of measuring the leakage through the piston would have been to have measured this leakage when the piston was in motion and under running conditions as nearly normal as possible. Several attempts were made to accomplish this, but all ended in failure, due to the fact that the plugs which were used to close the steam ports could not be kept in place when the engine was running. However, as the piston leakage is small in comparison with the valve leakage, the error due to the method used is reduced considerably. In making the calculations the assumption was made that the leakage was proportional to the pressure.

The total leakage per hour through the various lengths of the crank end of the special valve, together with that of the piston, was reduced to equivalent leakage at 100 pounds pressure, and then the total leakage per hour at 100 pounds pressure was determined for the piston. The piston leakage being subtracted from the total leakage for each of the various lengths of the special valve through which leakage occurred, gave the net leakage through each of these various lengths, per hour at 100 pounds pressure. These results are shown by the curve, Fig 5.

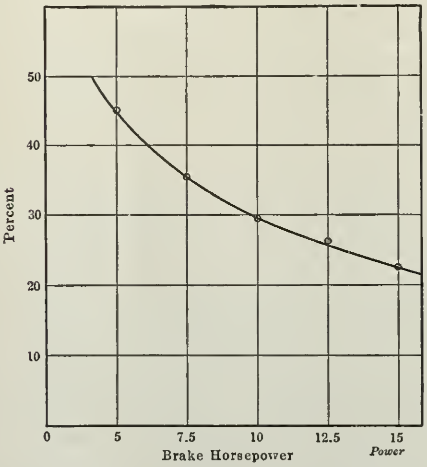


Fig. 7. Leakage in Percentage of Total Steam Used at Various Horsepowers

Next, the equivalent distance constantly leaked through when the crank end of the special valve was at each of the various lengths was calculated, the results being shown by the curve in Fig. 6.

The actual leakage per hour at 100 pounds pressure through the standard valve was now calculated. With the engine running normally, leakage occurred as follows:

1. Between the steam chest and the exhaust passage, through the spaces between the steam ports.
2. Through the spaces equal in width to the steam ports between admission and cutoff.
3. Between the cylinder and the exhaust passage, through spaces equal in width to the steam ports between cutoff and release and between compression and admission.

In order to determine the amount of these leaks, it was necessary to know the equivalent distances constantly leaked through and the pressure at which the leakage took place. The valve travel at each of the various horsepowers was determined, and knowing the valve travel, the equivalent distance constantly leaked through in each case, for each of the various horsepowers, was calculated.

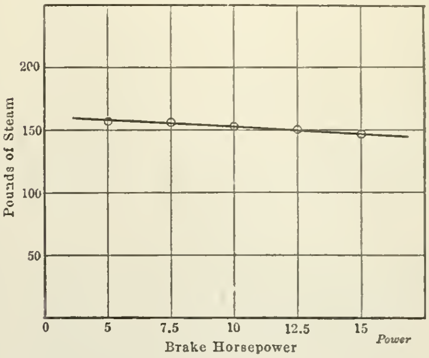


Fig. 8. Leakage of Steam Which Never Entered Cylinder

Leakage 1 occurred at a pressure equal to that of the steam chest as shown by the steam gage. Leakage 2 was due to this same pressure. Leakage 3, between cutoff and release, occurred at a pressure due to the average mean effective pressure of the expansion curves of the indicator cards for that particular run, and between compression and admission, occurred at a pressure due to the average mean effective pressures of the compression curves of the indicator cards for that particular run. Each of the above leaks was determined per hour at 100 pounds pressure by use of an equivalent distance curve.

This same method was used for each of the various horsepowers from 5 to 15, and the results were plotted as shown by the curve, Fig. 6.

The ratio of the total leakage through the standard valve to the total steam consumption at the various horsepowers was next determined, the results being shown by the curve in Fig. 7.

As the part of the leakage which occurred between the cylinder and exhaust passage consisted of steam which had done some work, another curve was drawn to show the amount of steam which never reached the cylinder at all, all the energy of which was absolutely lost. This curve is shown in Fig. 8.

In conclusion, the results show that leakage through the valve assumes enormous proportions. The total leakage increases slightly as the horsepower increases, and that the leakage of the steam which never enters the cylinder decreases slightly as the horsepower increases. The leakage in per cent. of the total steam used decreases as the horsepower increases. However, a leakage of 22 per cent., even when running most efficiently, is a tremendous loss of energy, and it is probable that the valve is the most imperfect part of the engine and that more energy is lost here than at any other place.

Since the leakage is so large, it is evident that every effort should be made to decrease it. However, there is every reason to believe that the valve tested was as perfect as it is possible to make this form of valve. The remedy does not lie in attempting to make a more perfect fit for this form of valve, but in designing some new form of valve through which the leakage shall be a negligible quantity or at least considerably less than 22 per cent.

Loss in Coal Due to Excess Air

By H. J. Westover

The proper amount of air for complete combustion is of utmost importance. The most practical way of determining it is to first examine a sample of the flue gases for the percentage of CO_2, and from these results the required amount of air may be figured. The article is written primarily to acquaint the fireman with the methods used in making these determinations.

The easiest and most reliable method of determining the quantity of air passing through the furnace is by obtaining an analysis of the products of combustion, preferably finding the percentage of carbon dioxide. The following table has been computed to show the loss in percentage of heat against the percentage of carbon dioxide, also the amount of money annually lost where the yearly coal bill is $1000:

LOSS DUE TO INCOMPLETE COMBUSTION.

Percentage of CO_2.	Loss, Per Cent.	Loss, Dollars.
14	no loss	.....
13	1½	15
12	2½	25
11	4½	45
10	6	60
9	8½	85
8	11	110
7	15	150
6	19½	195
5	26½	265
4	32	320
3	53½	535

If the percentage of CO_2 is known, one can see at a glance from this table, how much coal has been lost. These figures are not based on fictitious values but are figured on the actual amount of heat escaping to the chimney. There is no hidden mystery in the matter, whatever; a thermometer will show the temperature of these gases and the CO_2 determination will give the exact quantity of heat carried away.

These losses have not been figured from the best results obtainable, but from conditions that a good fireman can attain. If the stack temperature exceeds 525 degrees Fahrenheit, the loss will be greater than those given in the table, varying directly as this temperature increases.

The tests made by weighing both water and coal compare favorably with the above table. The figures shown in the table are a trifle higher for from 3 to 10 per cent. of CO_2, but above 10 per cent. there is practically no difference.

The method of figuring these percentages is given so that an intelligent fireman can get at the theory of his work and make the calculations for himself.

The work of the fireman has not been accorded the importance due it. The fireman is partly to blame for this. He should consider himself in the light of a manufacturing chemist, his business being to chemically combine carbon and hydrogen with oxygen, and the following remarks have been written for his benefit. He will find the work far more interesting when he understands the theory of combustion.

Coal is composed of carbon, hydrogen and several other elements, mostly unimportant. The only part of the coal we need to consider in this discussion is the carbon and the hydrogen. The sulphur and other elements, even when combustible, are not of much value.

At the start it is desirable to have a clear understanding of what is meant by "combination." If some red lead is mixed with white sand and soda, the particles of the mixture can again be separated, and after doing so the red lead, sand and soda would be the same as before. If, on the other hand, this mixture is placed on the end of a slice bar and put into the fire, the mixture will melt and upon removing it there will be a rough-looking piece of glass adhering to the slice bar, the red lead, sand and soda having disappeared. However, they still exist, but in a combined state, and it is no longer possible to take out a particle of sand or red lead. The new substance differs from any one of the original ingredients. This is what is termed a chemical combination and it differs in this respect from a mixture.

The carbon and hydrogen in coal combine with the oxygen of the air. This is a chemical combination and the result is a new substance, namely, carbon-dioxide gas and water. This new substance, CO_2, is totally unlike the coal, but at the same time every grain of carbon is contained in it.

It is a well known law in chemistry that the chemical elements combine with each other in exact proportions; for example, 1 pound of hydrogen will combine with 8 pounds of oxygen, forming 9 pounds of water. The same holds true with carbon; 1 pound combines with 2⅔ pounds of oxygen—no more, no less—to form carbon dioxide.

It is due to this indisputable fact, namely, that carbon, hydrogen and oxygen, combine in exact proportions, that the amount of air can be determined when we know the percentage of CO_2.

As already stated the sulphur and other minor ingredients are not important enough to be considered, and as no oxygen is required for the ashes no air is needed for the ash in the coal. We can also neglect the other combination wherein carbon unites with oxygen to form carbon monoxide, as there is at the present time very little of this gas formed in the ordinary furnace.

This is the gas formed when the damper is suddenly opened at a time when a very bright fire is burning. It is due to a shortage of air. However, as a rule, when looking through the holes in the fire door a light blue flame will be seen burning on the surface of the coal. This is the carbon monoxide meeting the leakage of air through the fire doors and burning to CO_2.

Analysis show that the soft coals in use throughout the United States consist of 94 parts of carbon to 6 parts hydrogen in every 100 parts of pure combustible, and if the coal has 10 per cent. ash it would require about 110 pounds of coal to furnish 94 pounds of carbon and 6 pounds of hydrogen.

The exact amount of air necessary to burn this 110 pounds of coal, consisting of 94 pounds of carbon and 6 pounds of hydrogen may be found. As previously stated, 1 pound of carbon requires 2⅔ pounds of oxygen to form CO_2 and 1 pound of hydrogen requires 8 pounds of oxygen to form water.

Carbon,	94 × 2⅔ =	250⅔ pounds oxygen
Hydrogen,	6 × 8 =	48 pounds oxygen
Total..............		298⅔ pounds oxygen

That is, 298⅔ pounds of oxygen are

required to burn the 110 pounds of coal. The oxygen is obtained from the air where it exists in the proportion of 23 pounds of oxygen to every 100 pounds of air. We shall therefore require

$$\frac{298.66 \times 100}{23} = 1298$$

pounds of air.

The air consists of oxygen and nitrogen and to obtain the oxygen it is necessary also to take the nitrogen. This is of no value; in fact, it is a decided loss, but it is unavoidable. The nitrogen exists in the proportion of 77 pounds in every 100 pounds of air. Hence,

$$\frac{1298 \times 77}{100} = 1000$$

pounds, approximately, of nitrogen in the air required for combustion.

The gases passing up the chimney therefore consist of nitrogen, carbon dioxide and some steam from the water formed by burning the hydrogen. The weights of these gases are as follows:

Carbon	94 pounds
Oxygen	250⅔ pounds
Carbon dioxide	344⅔ pounds
Hydrogen	6 pounds
Oxygen	48 pounds
Water	54 pounds

This makes a total of,

Carbon dioxide	344⅔ pounds
Water	54 pounds
Nitrogen	1000 pounds
	1398⅔ pounds = weight of gases going up the stack

So far, these quantities have been given in weight. For the convenience of measuring, they may be converted into cubic feet at atmospheric pressure and 72 degrees Fahrenheit. Further, in bringing these products of combustion to the machine for analyzing, we do not bring the steam or vapor as this is collected before the gases enter the measuring chamber. One pound of nitrogen occupies 13.8 cubic feet, and one pound of carbon dioxide occupies 8.8 cubic feet. These quantities will measure as follows:

		Cu.Ft.
Nitrogen,	1000 × 13.84 =	13,840
Carbon dioxide,	344⅔ × 8.8 =	3,028
Total		16,868

If exposed to a solution of caustic potash, the 3028 cubic feet of carbon dioxide would be absorbed, leaving the 13,840 cubic feet of nitrogen. The proportion of the total space to the space occupied by the CO_2 would be as 16,868 is to 3028, which is the same as 100 is to 18. Hence, the percentage of CO_2 gas is 18.

It will be noticed that the amount of nitrogen and oxygen required was 1000 and 298⅔ pounds respectively, making 1298⅔ pounds of air, and as 1 pound of air at 72 degrees Fahrenheit occupies 13.5 cubic feet, the 1298⅔ pounds of air would occupy 17,532 cubic feet. The ratio of the total volume of carbon dioxide to the volume of original air—17.3 per cent.—is not exactly the same as the ratio of carbon dioxide to the flue gas—18 per cent. This is due to part of the oxygen being taken for the hydrogen to form water. The larger the percentage of hydrogen in the fuel the larger the amount of air required.

In burning oil fuel more air is required for a given weight than for bituminous coal. On the other hand, less air is needed for anthracite coal than bituminous, because oil contains more hydrogen and anthracite less than bituminous coal.

The carbon combining with the oxygen does not increase the volume but it does increase the weight. It will be seen that if this fuel could be burned with just the quantity of air necessary for complete combustion the CO_2 would be 18 per cent.; but furnaces are not constructed so that every particle of fuel will come into contact with every particle of oxygen; consequently a margin is required and the greater the attention given to the handling of the fire the smaller will be the margin needed.

It has been found from practice that 14 per cent. of carbon dioxide is about as good as can be obtained, and it would be difficult to maintain this constantly with a changing load; however, this should be the mark aimed at.

To find the amount of surplus air for 14 per cent. of carbon dioxide proceed as follows: The products of combustion are proportional to the CO_2 gas; with 18 per cent. CO_2 the products of combustion measured 16,868 cubic feet; at 14 per cent. CO_2 the products would be 21,600; as 14 is to 18 so will 16,868 be to 21,600 and the value of these items will be

	Cu.Ft.
Nitrogen	13,840
Carbon dioxide	3,028
Surplus air	4,732
Total	21,600

Seeing that the products of combustion vary inversely as the percentage of CO_2 and it is desired to find what these will be for 6 per cent. of CO_2 we divide 18 by 6 and multiply by the products of combustion for 18 per cent. of CO_2, viz.,

$$\frac{18 \times 16{,}868}{6} = 50{,}619$$

Proceeding as before, we have

	Cu.Ft.
Nitrogen	13,840
Carbon dioxide	3,028
Excess air	33,751
Total	50,619

This represents an excess of 29,019 cubic feet over that required for 14 per cent. of carbon dioxide.

Now proceed to determine the amount of coal which would be required to heat this amount of excess air to the temperature at which it enters the stack and compare it with the original, this proportion being the percentage of loss.

Converting 29,019 cubic feet into pounds at 72 degrees Fahrenheit we have

$$\frac{29{,}019}{13.5} = 2150 \text{ pounds of air entering the}$$

furnace. The same weight of air leaves the stack at 525 degrees Fahrenheit. To raise 1 pound of the gases 1 degree Fahrenheit requires 0.236 B.t.u.; hence, it will take

$$(525 - 72) \times 0.236 = 107 \text{ B.t.u.}$$

to raise one pound from 72 to 525 degrees; consequently, to raise 2150 pounds of air to this temperature it will take

$$2150 \times 107 = 230{,}050 \text{ B.t.u.}$$

With a fairly good grade of bituminous coal having a heat content of about 14,500 B.t.u., and with 14 per cent. of carbon dioxide, 11 pounds of water can be evaporated from and at 212 degrees Fahrenheit per pound of coal. The actual B.t.u. required to evaporate 11 pounds of water under these conditions is 10,615.

Referring to the 230,050 B.t.u. required to raise 2150 pounds of air from 72 to 525 degrees, it will be found to take as many pounds of coal to furnish this heat as 10,615 is contained in 230,050.

$$\frac{230{,}050}{10{,}615} = 21.5$$

pounds of coal. This represents a loss of 19.5 per cent. of the original 110 pounds of coal.

A sample of gas showing 12 per cent. of carbon dioxide can often be obtained at some favored spot in the furnace when another sample at the same time taken at the breeching will show only 6 per cent. There is quite an art in obtaining a good average. It is quite possible to get a sample which is not a sample of the average gas and consequently the correct amount of air cannot be determined from such a sample. The CO_2 gas in most boilers is found richest in the center and poorer at the sides, front and end. This is due to the more rapid combustion at the center. As a rule the best average sample of gas for determining the loss due to excess of air is to be had at the breeching. The products usually are well mixed before entering the reduced orifice leading into the breeching and the stratified condition of the products often observed in the setting is well broken up at this point. The products at this point are carrying away heat at a temperature which can be measured without question and the quantity can be definitely determined.

For use during cold weather, when the trains of the Pennsylvania railway entering the New York station through the electrified-tunnel zone are disconnected from their steam locomotives and taken across the Jersey meadows and through the tunnels by electric locomotives, steam generated in electric boilers will be used to maintain the temperature in the cars and to keep the train connections from freezing. These boilers, which will utilize the 600-volt direct current from the third rail, are to be capable of generating 1000 pounds of steam at 80 pounds pressure.

A Homemade Reducing Motion

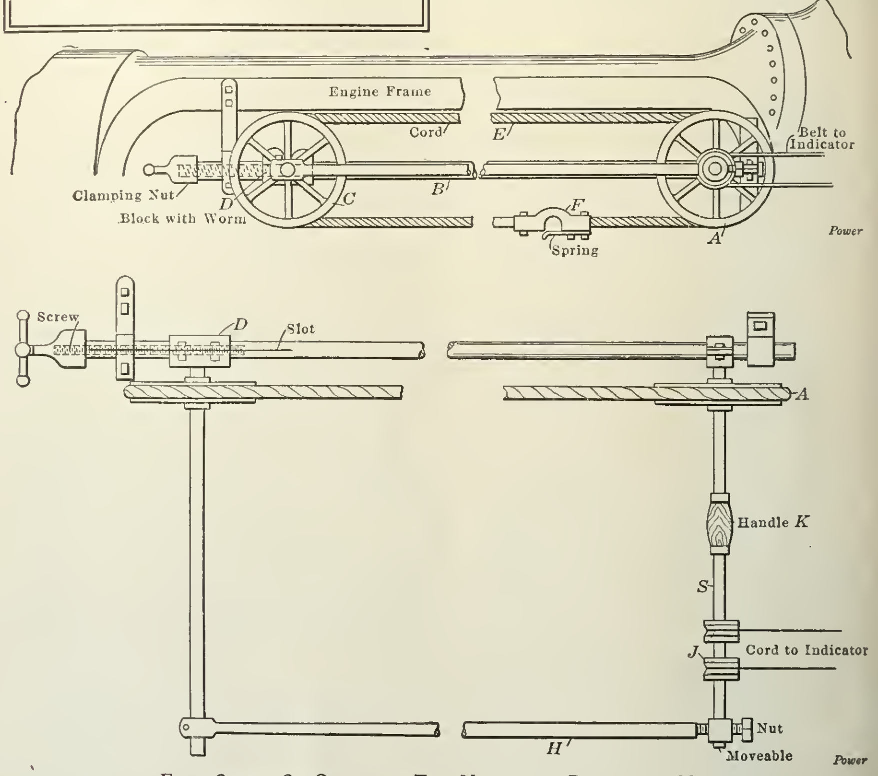
By Warren O. Rogers

Although the reducing motion is a homemade device, it is accurate and convenient to operate. When the device is applied to an engine it is a simple matter to take simultaneous diagrams from each end of the cylinder.

While visiting a cotton-mill power plant at Knoxville, Tenn., recently, the writer noticed a homemade reducing motion that was out of the ordinary, and yet of such a design that the reduction and formation of the diagram were practically perfect. The illustrations, Figs. 1 and 2, show the device attached to the low-pressure cylinder of a cross-compound Corliss engine.

The reducing motion consists of two brackets, one being secured to each end of the engine girder frame by means of screw bolts. A brass rod extends to and through these brackets, and is held in position by split-bearing clamps. A clamp is secured on each end of this brass rod, each clamp having a projection on the bottom in which a stud is fitted. On this stud a grooved pulley is loosely fitted, as shown in Figs. 1 and 2. The stud of pulley *A*, Fig. 2, remains fixed relative to the shaft *B*, but the stud of pulley *C* is adjustable. Inside of the rod *B* is a screw fitted with a crank handle, such as is used on the tailstock of an ordinary engine lathe. This screw engages with a female piece which is bolted to the clamp or bearing piece *D*. A thin extension of the female threaded nut is bolted to the clamp *D* and slides in a slot in the rod *B*. The purpose of this arrangement is to allow tightening the cord *E*, to prevent slipping. The bottom of the pulleys *A* and *C* come on a line with the center of the crosshead, in order that the pin projecting from the crosshead may engage in the hook link *F*. This connecting piece is made with a spring which fits to the under side; one end is fastened by means of screws, but the other end is free to give. The spring is turned down at the end so that the pin in the crosshead in striking the link forces the spring away from the engaging piece, and the crosshead pin, slipping into the pocket of the engaging piece, is held fast by the spring.

Fig. 3 shows a plan view of the device. The cord wheel *A* is fitted to a hollow shaft which runs on an inner shaft. This is for the purpose of disengaging the crosshead pin from the catch device while the engine is running. The outer shaft is fitted with a handle, and when it is desired to disconnect the reducing motion, the handle is pulled away from the crosshead. This draws the wheel *A* also, and the crosshead pin slips out of the engaging piece when it has reached a distance from the crosshead greater than the length of the crosshead pin.



Figs. 2 and 3. Side and Top Views of Reducing Motion

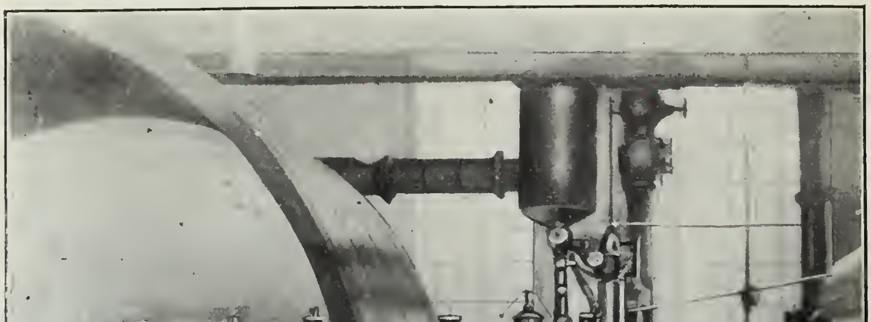


Fig. 1. Reducing Motion Attached to Engine Frame

The rod *H* is for the purpose of supporting the shaft *S*, on which two pulleys *J* are secured. These pulleys are of such size as to produce a diagram of the proper length. Two are used so that a diagram from each end of the cylinder can be taken at the same time. The only operation necessary after placing the paper on the indicator drum is that of moving the handle *K* in toward the engine, and then move the catch device to any position on the under side of the pulley, so that the crosshead pin can engage with the catch piece.

The cord running over the pulley is about ⅜ inch in diameter, which size prevents stretching or slipping. The device is interchangeable and can be used on any engine by changing the pulleys.

High Speed Compound Engine Design

By J. D. Speedium

The third and concluding instalment to the problem in English high-speed engine design. The engine frame, bearings, oiling system and governor are here dealt with.

CRANK SHAFT AND FLYWHEEL

The crank shaft is illustrated in Fig. 28. This is made from a solid forging of Siemens Martin acid steel having a tensile strength of not less than from 56,000 to 64,000 pounds per square inch. It is of larger diameter than is necessary to transmit either the torsional or bending stresses to which it is subjected, but this is absolutely necessary to provide ample bearing surfaces. Under ordinary working conditions the pressure per square inch on the bearings—taking the area equal to the diameter multiplied by the length—does not exceed 200 pounds. Four main bearings are provided, the one near the flywheel being considerably longer than the others, to carry the weight of the flywheel and part of the rotor of the dynamo, if the engine is used for that purpose.

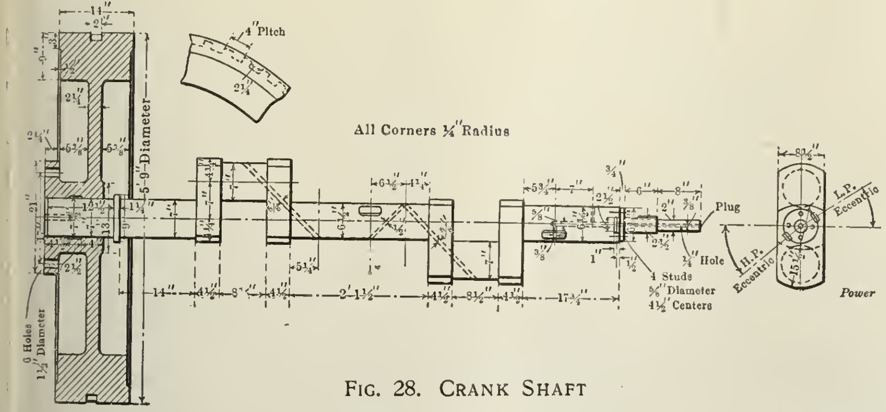


FIG. 28. CRANK SHAFT

To lubricate the crank pins a hole is drilled at an angle from the nearest main bearing through the web to the pin. A small extended shaft is bolted onto the end of the crank shaft at one end to carry the governor, this shaft being provided with a spigot to fit into a suitable recess in the main shaft. The governor is lubricated by a hole drilled up through the end of the shaft to meet a hole drilled from the outside on the nearest main bearing.

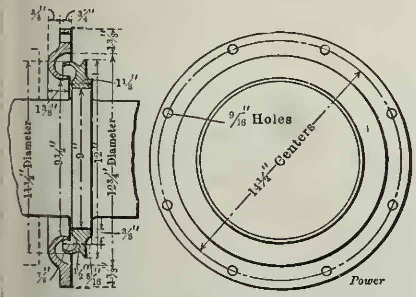


FIG. 29. OIL THROWER

Nearly all two-crank engines have the cranks placed at 180 degrees. This gives the best balance for running at high speeds, and at the same time reduces the load on the main bearings. In the case of engines running at very high speeds, balance weights are fitted to the cranks, these being arranged to balance the crank pin, with the webs, and in addition about 50 per cent. of the reciprocating parts.

Great accuracy is necessary in machining the crank shaft to insure a smooth-running engine; consequently, it is usual to specify that the crank pins and main bearings shall not be more than 0.002 inch out of round, and the crank pins not more than 0.002 inch out of parallel with the shaft, in a length of 12 inches. These limits are adhered to even for the larger shafts.

Owing to the high speed of rotation, the flywheel is usually of small diameter, being made in plate form, which no doubt is the strongest and most suitable design possible. A speed of 6000 feet per minute on the periphery of the wheel is generally adopted and rarely exceeded. For powers up to about 800 horsepower it is usual to force on the flywheels by hydraulic pressure, afterward drilling one or two round keyways half in the wheel and half in the shaft, see Fig. 28.

In the early high-speed engines, great trouble was experienced from flywheels working loose, as they were made only a fair fit on the shaft, and keyed in the ordinary way. Flywheels forced on the shaft, however, never give any trouble in this respect. For larger sizes of engine it is usual to provide a solid coupling on the end of the crank shaft, and bolt the flywheel plate between this coupling and the dynamo coupling, a small spigot being provided on the face of the engine coupling to center the flywheel.

As regards the weight of flywheels for ordinary stationary practice, a weight giving approximately 700 to 800 foot-pounds of stored energy per indicated horsepower is found sufficient, and for traction purposes, etc., a weight of 1500 to 2000 foot-pounds of stored energy should be used.

OIL THROWER

A small detail which has caused much trouble during the development of high-speed engines is the method of preventing oil from escaping through the frame through which the crank shaft projects to carry the flywheel. Again, owing to the temperature rise which occurs after an engine has been running some hours, it has been found necessary not only to

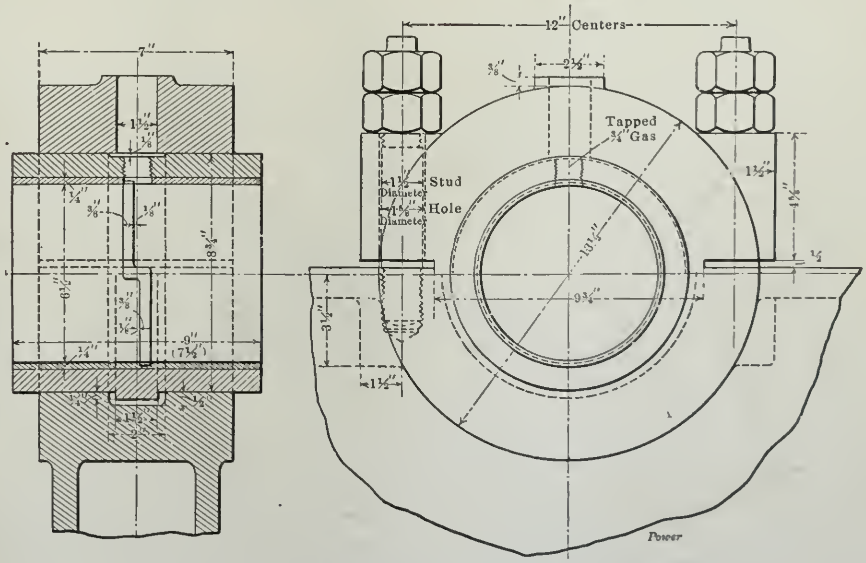


FIG. 30. INTERMEDIATE AND GOVERNOR END BEARINGS

prevent the oil leaking out, but to prevent the oil vapor and spray from leaking out and coming in contact with the electrical machines.

When the bearings of an engine are entirely inclosed in the frame, with an oil thrower fitted beyond the bearings, it is difficult to prevent the vapor from escaping. In the present engine, this defect is overcome by fitting a flange on the main bearing cap of the flywheel; this makes an oil-tight joint on the end of the frame, and the only oil or vapor which can leak out has to pass through the bearing. This is easily dealt with by means of an oil thrower of the design shown in Fig. 29; the oil collected in the outer ring or scoop being drained back into the engine base through a hole seen in Fig. 31; and it is only through this hole that oil vapor can pass.

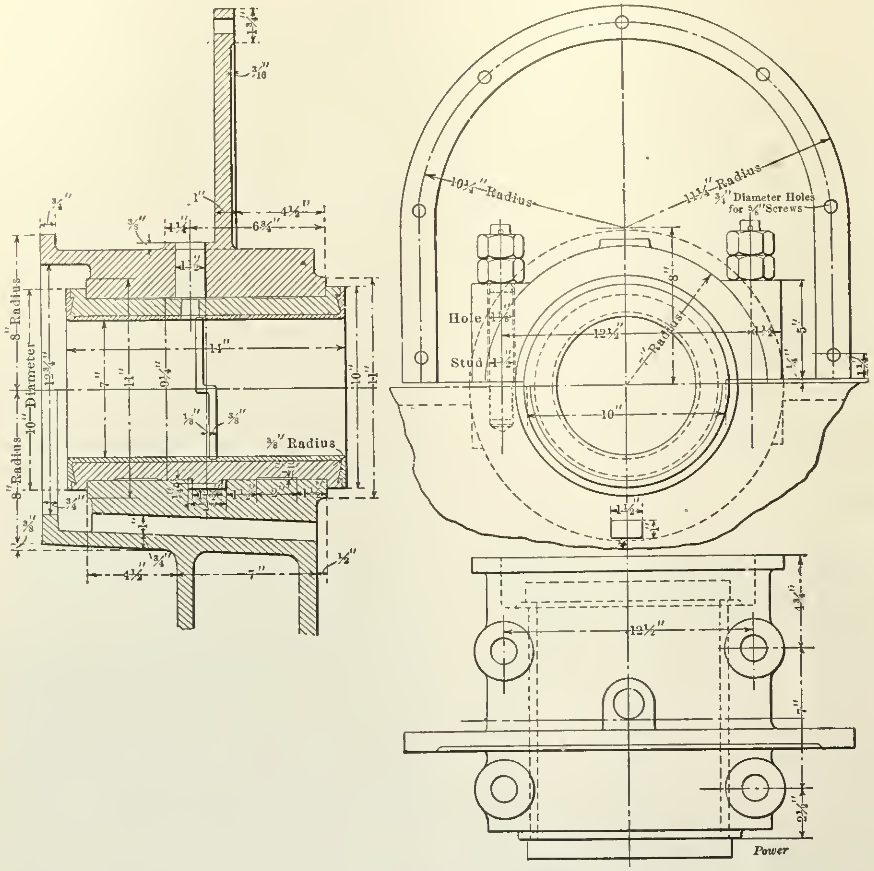


FIG. 31. FLYWHEEL END OF MAIN BEARINGS

MAIN BEARINGS

The main bearings for the crank shaft are shown in Figs. 30 and 31. The shells or bushes are usually made of cast iron, and are lined with antifriction metal. The caps are also of cast iron and are spigoted into the base as shown. This spigot fulfils two purposes: It centers the caps, and at the same time prevents the bushes from rotating. By making the bushes in this way, it is possible to remove the bottom halves without lifting the crank shaft, as they can be turned round the shaft and lifted out, when the cap and top-half bearing have been removed. The caps are secured by studs fitted with two nuts, one acting as a lock nut, and in addition a split pin is fitted.

The oil is fed into the top of the bearing, through a pipe which is screwed into the top half of the bush, and passes through a hole cut in the cap, the bearing being grooved to distribute the oil around the shaft.

The flywheel end bearing, Fig. 31, is used as a thrust bearing, and is accordingly provided with collars at each end, the antifriction metal being carried over the end faces which come in contact with the crank shaft. A central collar is provided on the bottom half of this bush to prevent it from rotating with the shaft. In the case of the other bearings this collar also prevents the bush from moving longitudinally.

ENGINE FRAME AND BASE

The engine base shown in Fig. 32 practically consists of a tank with sides of box-girder section. The two outer main bearings are carried by the ends of the casting, and the two inner bearings are carried by intermediate ribs, which span the center of the bed. Owing to the ample supply of lubrication required in high-speed engines, it is necessary to entirely inclose all the working parts; hence the frame and base are practically a sealed box when the engine is running.

The frame is shown in Fig. 33, and consists of a strong box, well ribbed to give stiffness and provided with doors to afford easy access to the working parts. The bottom is flanged where it sits on the base and is secured by closely pitched studs. The top of the frame is inclosed and is strongly ribbed to receive the distance-piece castings which carry the cylinders. The holes for the doors are made as large as advisable and have semicircular ends. Two doors are also fitted to the base, one at the back and one at the front, to give access to the oil pump and strainer.

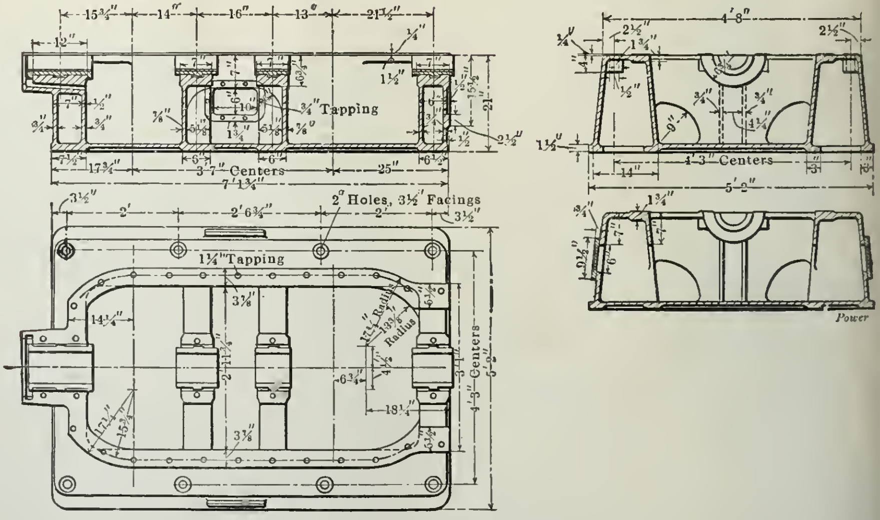


FIG. 32. ENGINE BASE

The foundation bolts, eight in number, are carried through the whole depth of the base and thus have a very firm hold on the engine. The right-hand side of the frame and base is worked to a circular form and is provided with a facing, to which the governor casing is attached, the governor also being entirely inclosed.

Oil Pump and Strainer

The oil pump and strainer are detailed in Figs. 34 to 36. The pump is of the valveless type and is actuated from a lug cast on the low-pressure eccentric clip; see the assembled drawing, Fig. 46. From this it will be seen that the pin, which works the oil pump, makes an elliptical path and causes the ram, together with the oil-pump barrel, Fig 34, to oscillate through a considerable angle on the trunnion, Fig. 35. During the in-stroke of the ram the port in the bottom of the barrel is open to the suction port in the trunnion, and on the out-stroke, the oscillation of the barrel puts this port in connection with the delivery port in the trunnion; hence no valves are necessary. The pump ram is usually of gun metal and is a good working fit in the barrel, which is of cast iron. No packing is necessary, and as the ram works under most favorable conditions as regards lubrication, very little wear takes place. The trunnion, Fig. 35, which is of cast iron, is divided into two compartments, most clearly seen in section in Fig. 35a. It is made a templet fit in the bracket which carries it, and is prevented from turning round by means of two pinching screws at the top. The suction pipe is connected by a flange to one end of the trunnion and the delivery to the other.

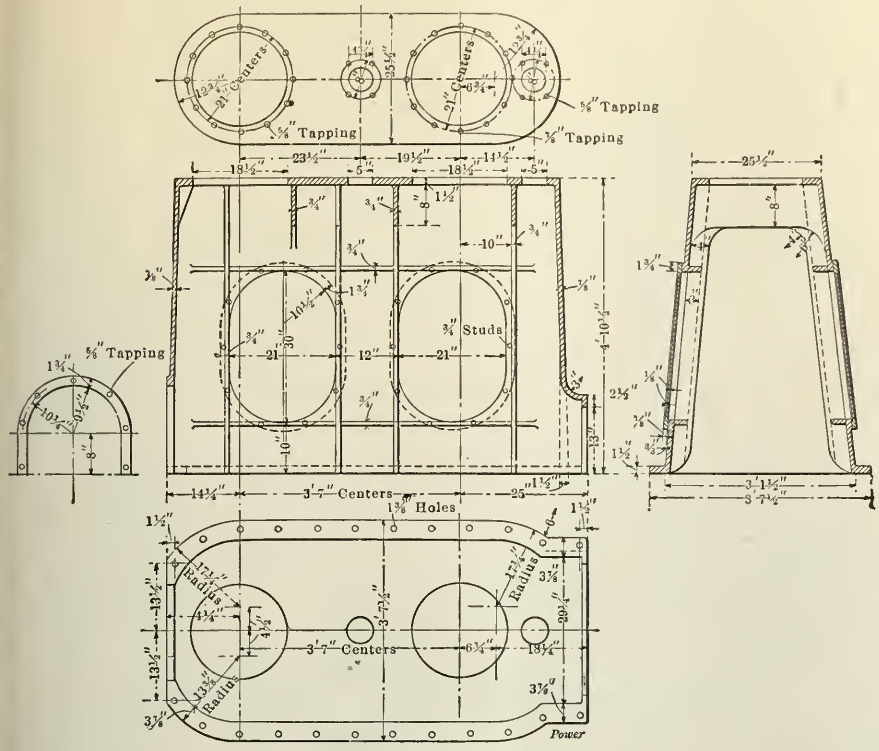


Fig. 33. Engine Frame

There is no fixed rule for determining the capacity of the oil pumps, as the capacity has to be determined by experience; they are usually made larger than required, so as to give some margin for wear of the parts and leakage. Pumps are fitted of such a capacity as to maintain an oil pressure of at least 20 pounds per square inch after considerable wear has taken place.

In Fig. 36 are illustrated the oil strainer and trough. This is a very important detail, as it is absolutely necessary to prevent any grit from getting into the suction pipe. A small amount of grit in a bearing on a slow-speed engine will only cause that bearing to heat up, but if grit gets into the suction pipe of a forced-lubrication engine it is immediately distributed over all the working parts.

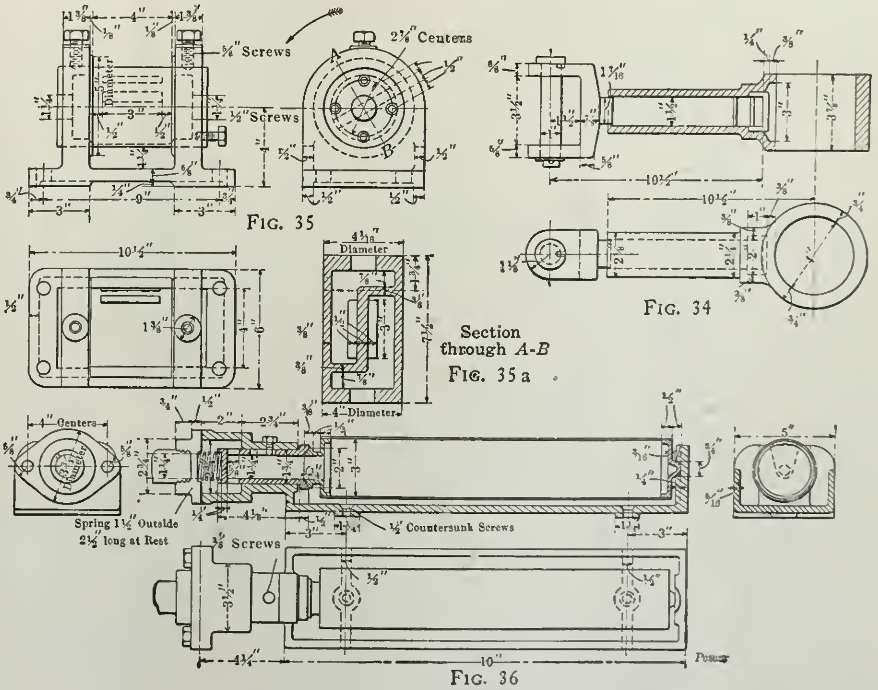


Fig. 35

Fig. 34

Fig. 35 a

Fig. 36

Figs. 34-36. Details of Oil Pump and Strainer

The strainer consists of a light, perforated tube closed at one end and fitted with an open branch at the other. The open end is made to engage with the conical seat of a spring-loaded piston valve containing one set of ports. When the strainer is in position the ports in the small piston valve are open to the suction pipe, but as soon as the strainer is removed, the spring pushes out the piston valve and closes the ports, thus preventing the pump from sucking oil direct from the engine base, should it be found necessary to remove the strainer while the engine is running. The perforated tube of the strainer is wrapped with fine copper-wire gauze. The strainer can be removed easily, as it is held in place only by the valve. The trough is fitted as an additional precaution against grit getting into the suction pipe. Also, the trough prevents water from being drawn into the pump. A pure mineral oil is used for lubrication and the same oil is often used for many months without filtering or changing. Engines of this type regularly run for six months without a stop and without change of oil.

Governor

High-speed, inclosed, forced-lubrication engines are universally governed by a throttle valve. The governor is of a simple centrifugal type and is fixed on the end of the crank shaft. Through the sleeve and bell-crank lever inclosed in the governor casing it is connected directly to the throttle valve. This valve is shown in detail in Fig. 37 and consists of a double-seated balanced valve with steam entering at the top. Both valve and seat are made of cast iron, but the valve box

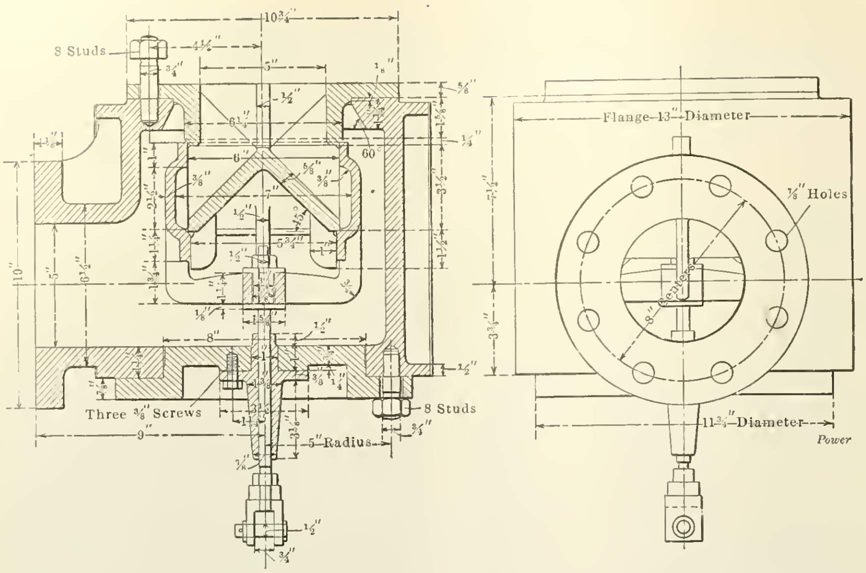


Fig. 37. Throttle Valve

is usually made of cast steel for pressures exceeding 150 pounds per square inch. The seat is let in from the top side and is held in position by the same studs which secure the steam-pipe flange. As the seat is not constrained in any way, only being supported at one end, it is not liable to be distorted; consequently the valve remains tight under all conditions of pressure and temperature. The valve is a floating fit on the spindle and can thus freely center itself on its seat. No packing is fitted to the throttle-valve spindle, but a long, closely fitting bush takes its place. This arrangement is preferable to a gland and stuffing box, as it cannot be tampered with. It is usual to adjust the governor so that the valve is pressed hard onto its seat when the governor balls are in their out position. The spindle is made adjustable at the knuckle joint just below the bush through which it works.

The details of the governor are shown in Figs. 38 to 42, Fig. 38 showing the bracket which carries the weights. The fulcrum pins are case-hardened and are lubricated by oil under pressure, a hole being drilled through the center of the crank shaft and making connection with the nearest main bearing; see Fig. 28.

The governor weights and levers are shown in Fig. 39. These are usually made of cast steel. The weight portion is made hollow, a cap being fitted at either end. Lead washers can thus be added in the pockets until the correct speed is obtained. A steel pin, fitted with hardened-steel ferrules at each end for attachment to the springs, is forced through the center of each weight. The other ends of the levers are fitted with pegs on which square brass blocks are placed, these engaging with suitable horns formed on the governor sleeve; see Fig. 40. At the other end of this sleeve are formed two collars, into which is fitted the trunnion ring, Fig. 40. This trunnion ring transmits the motion of the governor to the steel bell-crank lever, Fig. 42. The sleeve, Fig. 40, is of cast iron and is made a sliding fit on the crank-shaft end. Oil under pressure is admitted to the cavity in the center of the sleeve, and from this cavity the oil is led to the trunnion ring and sleeve blocks by the small holes shown. Thus, all the working parts of the governor are lubricated automatically by oil under pressure.

A detailed drawing of the governor casing is shown in Fig. 43. Two facings are provided at the top, one for the bracket shown in Fig. 44, carrying the bell-crank lever, and the other for the speeder-spring plate, see Fig. 45. A door is provided in the end of the casing to give

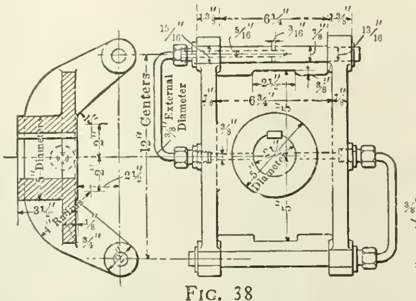


Fig. 38

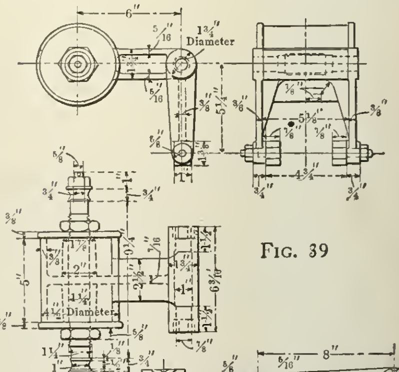


Fig. 39

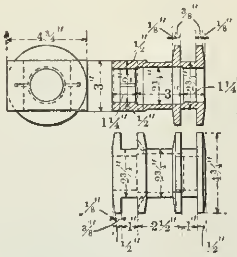


Fig. 40

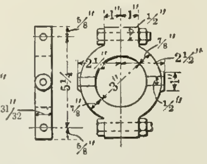


Fig. 41

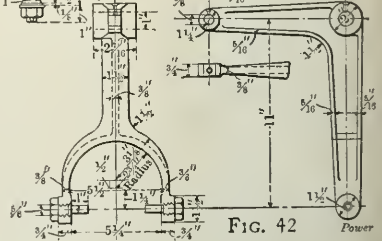


Fig. 42

Figs. 38-42. Governor Details

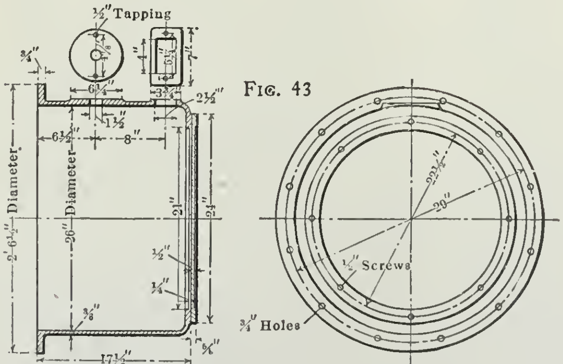


Fig. 43

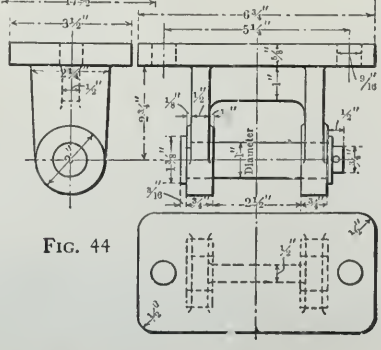


Fig. 44

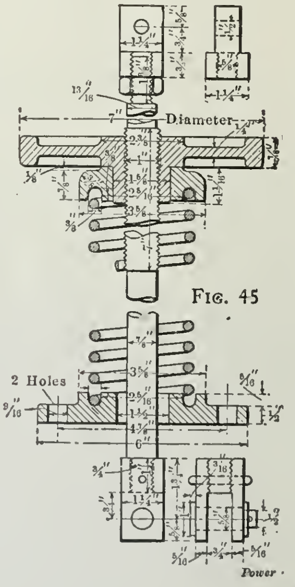


Fig. 45

Figs. 43-45. Governor Casing, Bracket and Speeder Gear

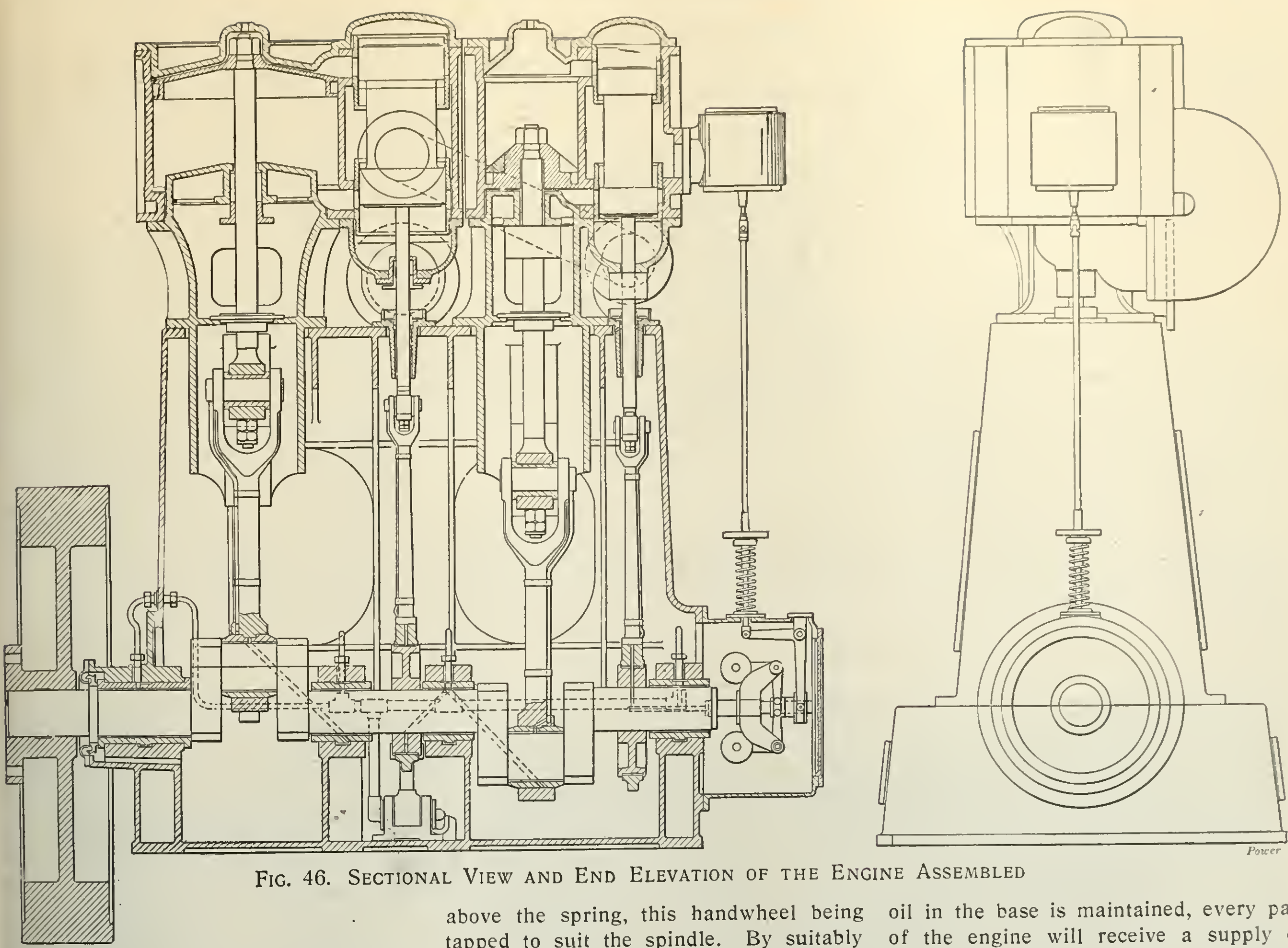


Fig. 46. Sectional View and End Elevation of the Engine Assembled

access to the working parts of the governor.

The speeder gear is fitted so that the speed of the engine may be varied 5 per cent. while the engine is running. It consists of a compression spring, surrounding the connecting link between the governor and the throttle valve, which always tends to close the valve, and so acts against the governor springs proper. The compression on the spring can be varied by means of the handwheel shown above the spring, this handwheel being tapped to suit the spindle. By suitably proportioning this spring with the governor springs, the variation in speed is obtained.

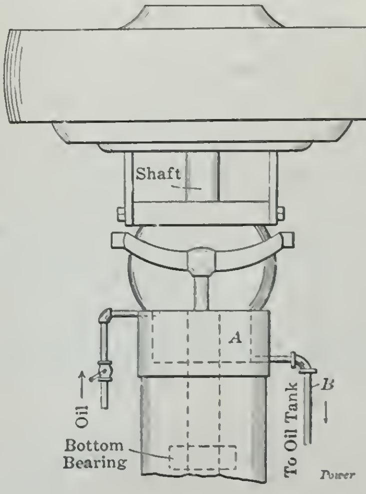
Lubrication

As previously stated, the lubrication of the engine is entirely automatic. Before starting the engine, the base is filled with oil to such a hight as to cover the top of the oil strainer about one inch. When the engine starts, the oil pump begins pumping; and so long as the level of the oil in the base is maintained, every part of the engine will receive a supply of oil under pressure. A pressure gage located outside the frame is coupled to the oil-delivery pipe, and by means of a small bypass valve, which allows the surplus oil to flow direct to the engine base, the pressure can be regulated at will. The usual pressure allowed is 20 pounds per square inch. The steam cylinders are lubricated by a separate mechanically driven force-pump type of lubricator. An assembled drawing of the engine is shown in Fig. 46.

Oiling System for Turbine Governor

The general practice in lubricating some types of turbine governor is to oil the top bearing supporting the governor spindle, just below the forked arm connecting the valve-regulating levers, and depend on the oil running down the shaft to lubricate the bottom bearing.

In one instance, the engineer in charge of a steam turbine in the city of Richmond, Va., worked out a means of effectually lubricating the lower spindle bearing of the governor on the turbine unit, by leading a small brass pipe from the return oil line from the main bearings to the upper bearing of the governor, as shown in the accompanying illustration. The incoming oil entered at the top of the collar *A* and discharged through the pipe *B* into the storage oil tank at the end of the turbine. In this manner the slight pressure of the oil in the return pipe from the bearings was sufficient to force the oil up through the pipe to the upper bearing of the governor, but not in such quantities as to cause the oil to fly. As the cup *A* is always partially filled with oil, so long as the bearing oil for the main bearings of the turbine is in circulation, the bottom bearing of the governor shaft is always well oiled, thus eliminating all trouble from lack of proper and sufficient lubrication.



Piping to Governor Bearings

It is always a good plan to keep the gage cocks in good order and use them. Place tees or crosses at the bend of both pipes from the boiler to the water column; remove the plugs and clean these pipes when washing out the boiler.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

A Small Power Plant with Modern Features

By L. L. Brewster

The plant herein described is one of several power stations of the Standard Oilcloth Company and is located between Peekskill and Montrose, alongside the tracks of the New York Central & Hudson River Railroad Company. The plant was installed under the advice and direction of the Loomis-Pettibone Company, consulting engineers, of New York, and the choice of apparatus was made before the designs for the building and the layout of apparatus were begun. The result of this procedure is a most complete and harmonious plant, the operation of which affords evidence of careful and competent design.

The building is a handsome structure of red brick and molded concrete trimmings, with a concrete floor and a reinforced-concrete roof supported by a steel skeleton frame. The wall columns support the track for a 7½-ton Cleveland crane, shown in the picture of the engine room. The engine-room walls are faced with light gray impervious brick, and four monitor skylights extend nearly the entire length of the room.

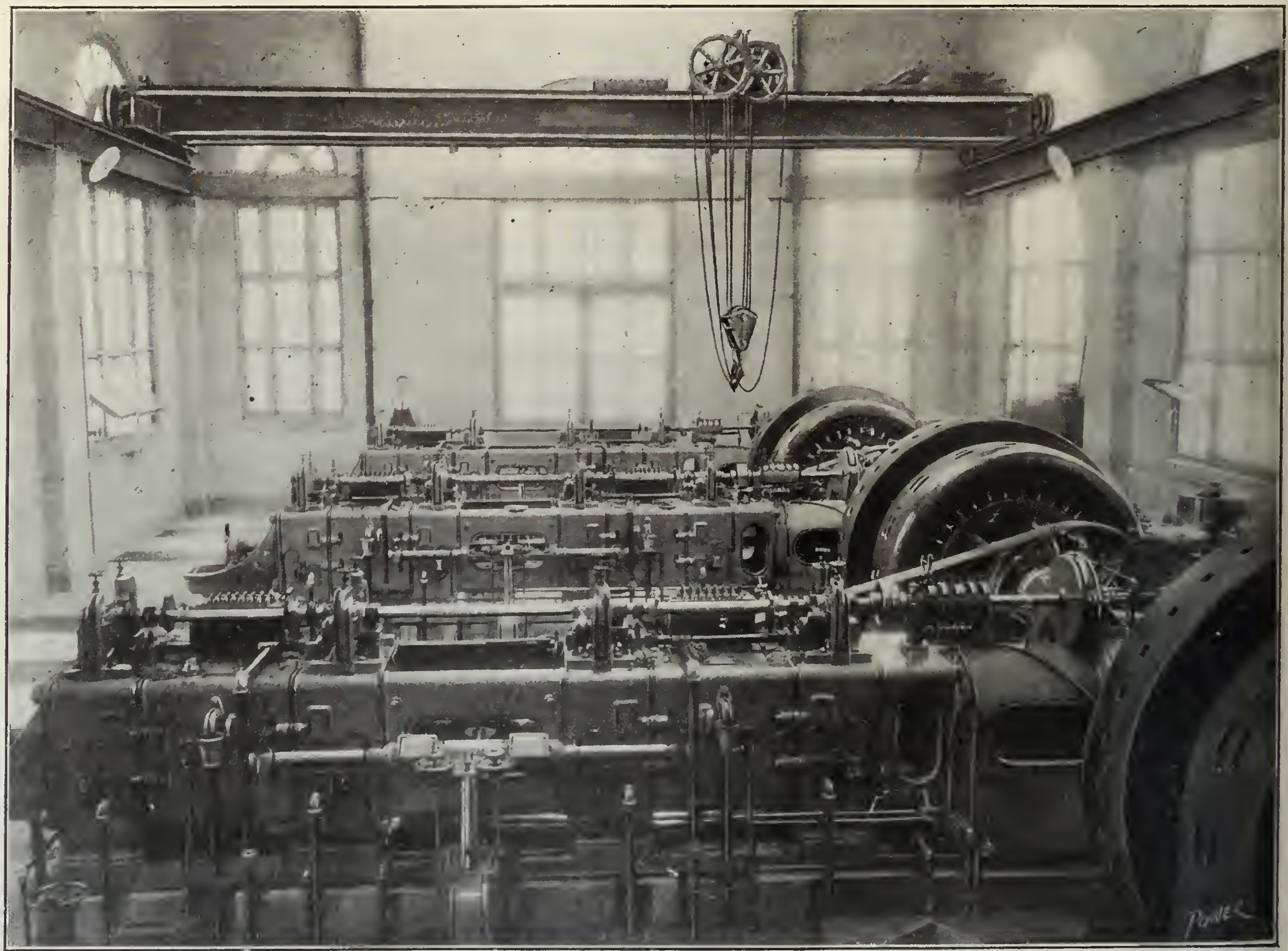


Fig. 1. The Generating Units in the Power Plant of the Standard Oil Cloth Company

On account of the character of the subsoil it was found necessary to lay a slab of concrete three feet thick entirely over the area inside the engine room.

Producers

The producers are operating on bituminous coal and are the Loomis-Pettibone down-draft single-generator type, built by the Power and Mining Machinery Company, of Milwaukee, Wis. There are two 7-foot generators, and each generator has a lining 13½ inches thick and is rated at 250 horsepower. The grate area of each generator is about 17.72 square feet, or a trifle over 0.07 square foot per rated horsepower. The generators are 12 feet 6 inches high. After leaving the generator, the gas passes first through an air preheater 3 feet 2 inches in diameter by 13 feet high, which serves not only to heat the air going to the generators but also to produce the steam to be admitted with the air to the fuel bed. The amount of water delivered to the preheaters is regulated by hand.

From the preheater, the gas goes to the wet scrubber or more properly cooler, 30 inches in diameter by 15 feet 6 inches high. The scrubbers are supplied with water through spray nozzles at a pressure of about 35 pounds per square inch.

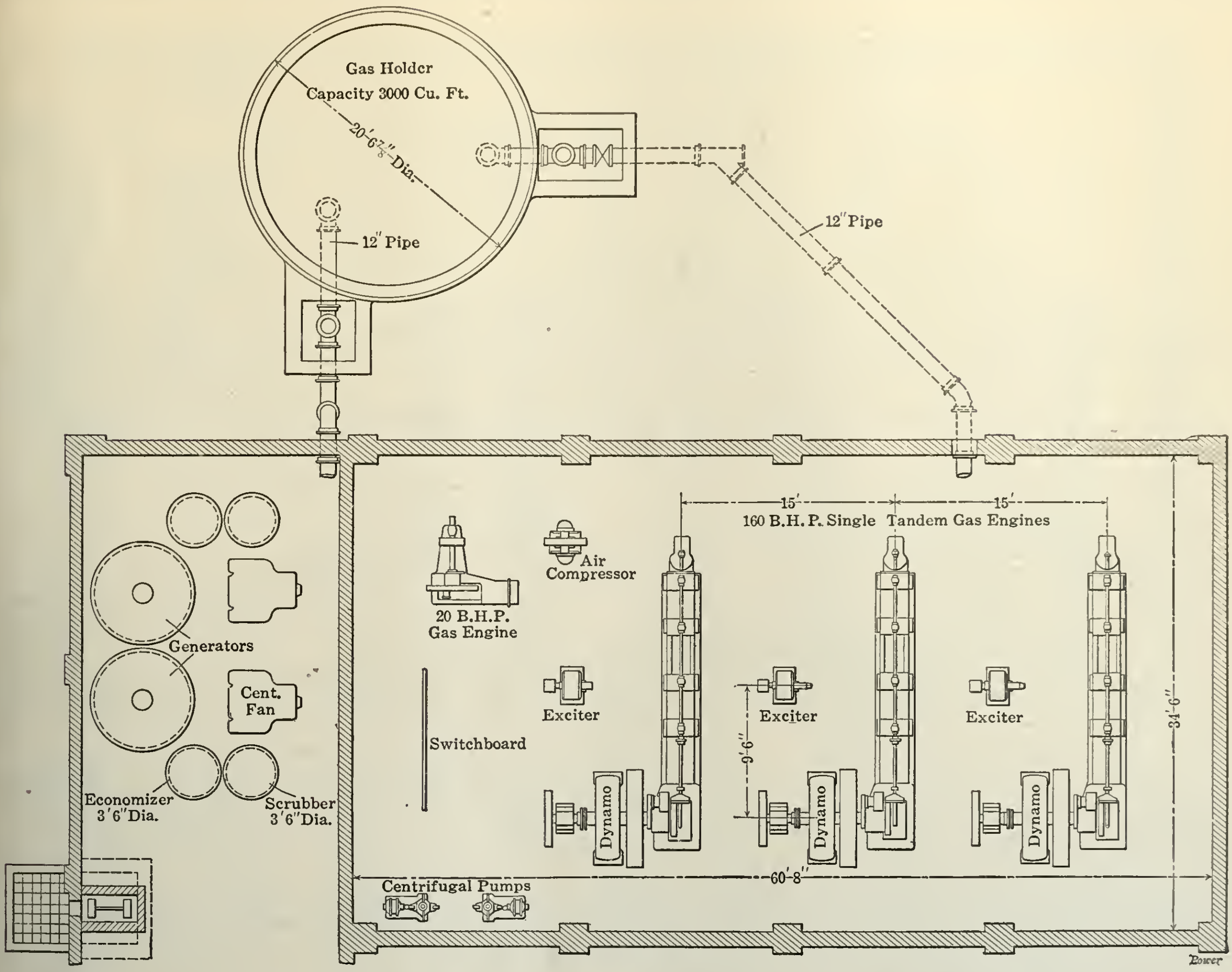


FIG. 2. GENERAL LAYOUT OF THE COMPLETE POWER PLANT

Several nozzles are used, and they are distributed around the body of the scrubber. No coke or other distributing material is used in the coolers, but the extremely well divided spray throws down a large percentage of the dust contained in the gas.

After the gas leaves the static scrubbers it passes to centrifugal washers and exhausters, consisting of turbine blowers, furnished by the General Electric Company, capable of delivering against any pressure up to about two pounds gage. These blowers run at 3600 revolutions per minute, and are driven direct by 20-horsepower induction motors. The blowers are equipped with atomizing spray nozzles and supplied with water at about 35 pounds pressure. Both as exhausters and gas cleaners they have proved very efficient.

It is very evident that the engineers have appreciated the importance of thoroughly cleansing the producer gas and have provided the requisite apparatus to do it well. Tests show the dust content to be very low; the figures are not before me, but they run considerably below 0.02 grain per cubic foot.

A study of the producer layout, Fig. 5, will show that either producer can be served by either exhauster or both producers and both exhausters can be operated simultaneously. Each producer has sufficient maximum capacity to supply the full engine capacity of 500 horsepower.

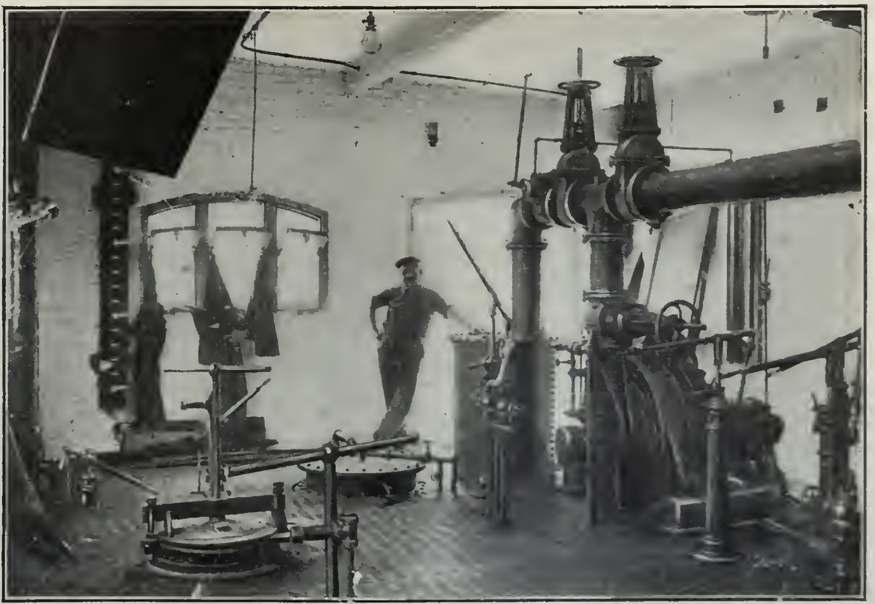


FIG. 3. PRODUCER CHARGING FLOOR

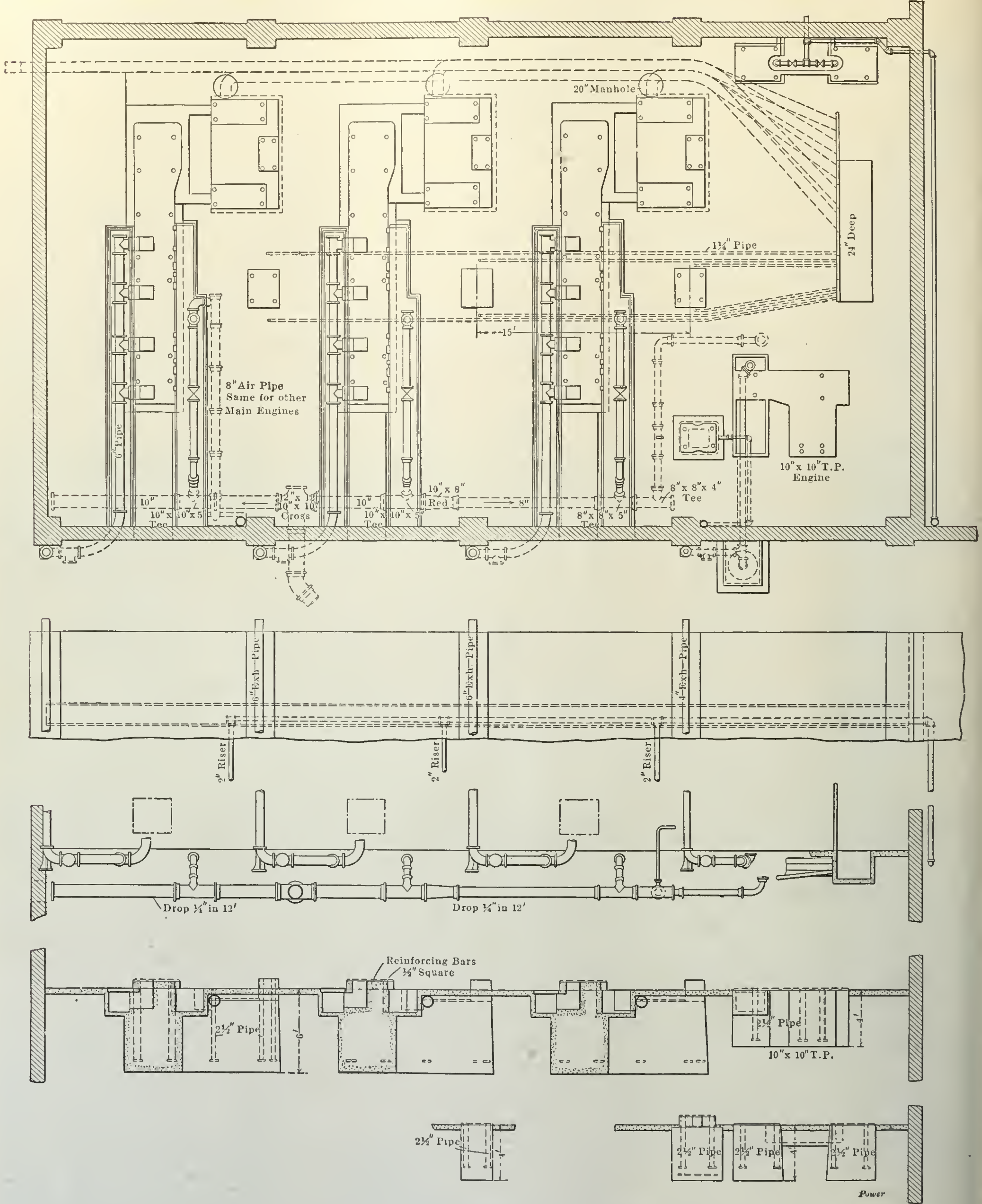


Fig. 4. Foundations and Piping Plan and Part Elevation

Coal is delivered in cars to the rear of the power house, where it is dumped into a hopper behind the producer room and elevated to the charging floor by a motor-driven bucket elevator furnished by the Link Belt Engineering Company. Spouts deposit the coal on the charging floor at any point desired.

The exhausters discharge into a common 12-inch main leading to a gas holder of 3000 cubic feet capacity, which serves more as a pressure regulator than as a reservoir; its contents would be consumed in less than four minutes by the three engines running at full load. Its function as a regulator is, however, very important, especially as the three engines drive direct-connected alternators in parallel, and with gas coming to each engine at constant pressure parallel operation is accomplished without any difficulty whatever, because one engine cannot rob another of gas.

The piping from the holder to the enines is of cast iron with bell and spigot oints. Outside the power house it is aid in the ground and inside the house t is placed in trenches which are covered ith steel plates. Figs. 4 and 5 give the ipe sizes.

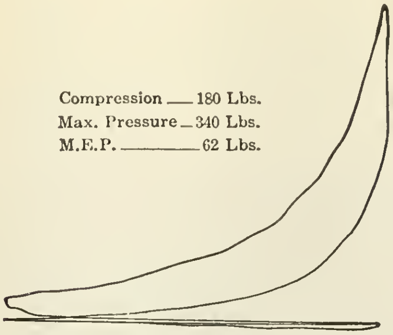


FIG. 7

COMPLETE AND STOP INDICATOR DIAGRAMS

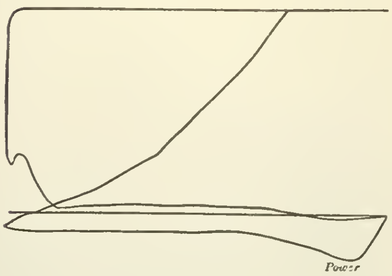
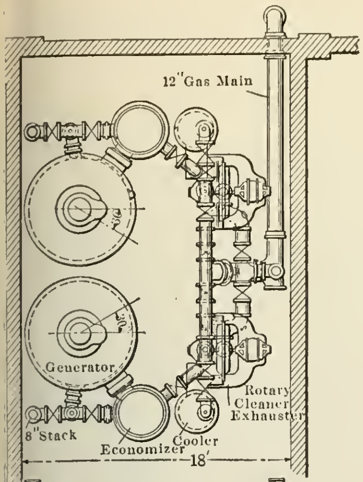


FIG. 8



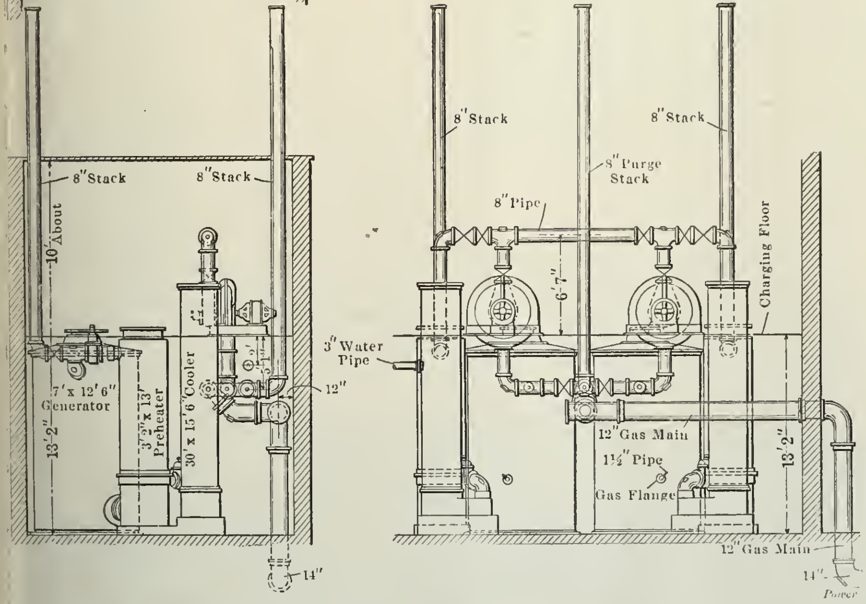


FIG. 5. ARRANGEMENT AND CONNECTIONS OF PRODUCERS

In the Engine Room

The engine equipment consists of three ouble-acting tandem units of 160 horseower each and a small auxiliary singlelinder single-acting engine of 20 horseower, all of which were furnished by ie Riverside Engine Company, of Oil ity, Penn. Each main engine is direct nnected to a 100-kilovolt-ampere alrnator and belted to an exciter unit; ig. 6 illustrates this arrangement. All ie electrical apparatus, including the vitchboard, was furnished by the Genal Electric Company.

The auxiliary engine is belted to a 6x6-ch Ingersoll-Rand compressor and also a 1-kilowatt, 125-volt direct-current nerator, as shown in Fig. 9. This genator supplies ignition current to the gines through the ignition switchboard own to the left of the main board in g. 10, but there are two other sources of ignition current, namely, a Westinghouse 120-volt storage battery and the exciter busbars. In case of a total shutdown, the battery would be used for starting and if sufficient gas had not been retained in the holder to supply the auxiliary engine, gasolene may be used.

All of the engines are ordinarily started by means of compressed gas instead of using compressed air as is commonly done. The belted compressor delivers compressed gas into a steel tank, 54x96 inches, at about 125 pounds pressure, and from this source the engines are started and the producers are reverse-blasted at intervals.

A view of the main engines is given in Fig. 1 and their relative locations are shown in Fig. 2. The cylinders are 15 inches in diameter, the stroke is 16 inches and the engines run at 225 revolutions per minute, which gives a piston speed of 600 feet per minute. They are of the standard Riverside design and are fitted with Jahns centrifugal governors.

The warm discharge water from the cylinder jackets is delivered to the piston rods at the intermediate crosshead, from which point the water flows in opposite directions through each rod and piston. A pair of telescopic joints convey the water to the moving crosshead. This use

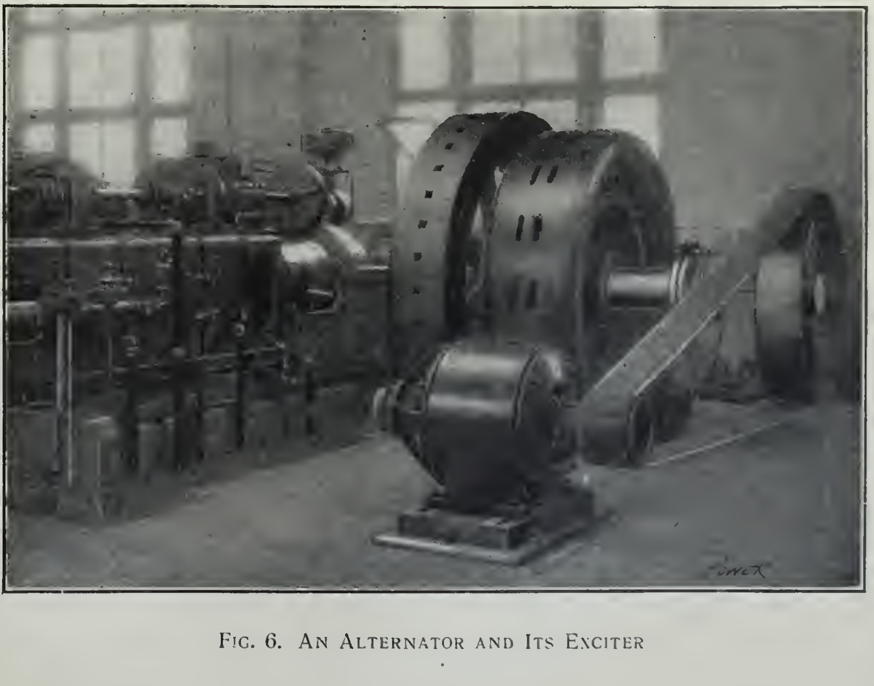


FIG. 6. AN ALTERNATOR AND ITS EXCITER

of warm, instead of cold, water in the piston rods, together with the use of a special steel in making them, has cured rod wear due to chemical action.

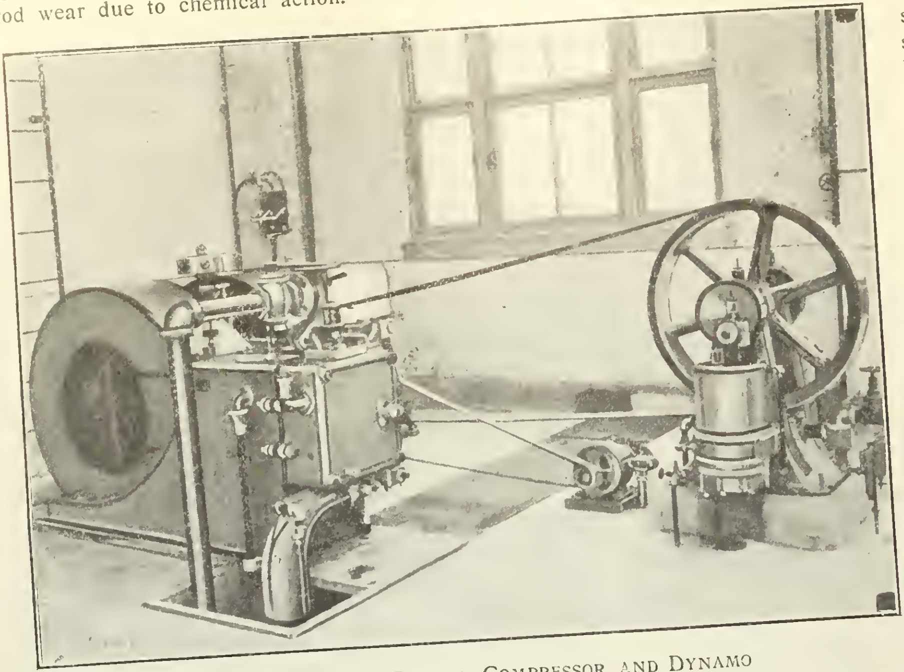


Fig. 9. Auxiliary Engine, Compressor and Dynamo

The gas made in this plant has a heat value of 110 to 120 B.t.u. per cubic foot, the hydrogen percentage being usually low. The explanation is that very little steam is delivered to the producers. With the high engine compression used, however, a very good efficiency is obtained and there is an almost complete absence of back firing. Advantage was taken of these gas characteristics to build the engines for high compression. Fig. 7 is an indicator diagram taken when the load was not maximum, the mixture being slightly throttled, as shown by the stop diagram in Fig. 8.

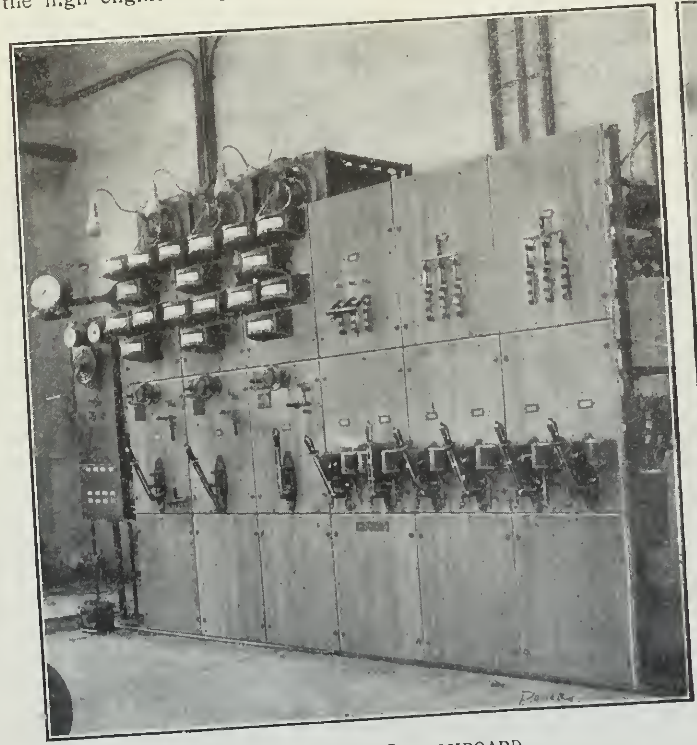


Fig. 10. The Switchboard

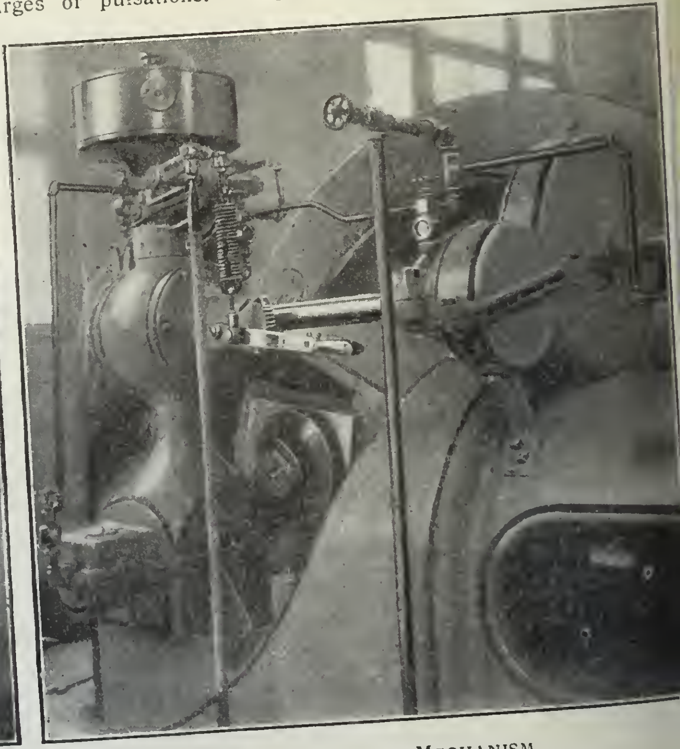


Fig. 11. Governor Mechanism

Any two or all of the three units may be operated in parallel without difficulty. This is, of course, due largely to the freedom from back firing, and also to the supplying of gas at uniform pressure, without pipe-line surges or pulsations.

The speed of each engine may be varied by hand over a considerable range by means of an auxiliary spring and adjusting lever connected to the governor as shown in Fig. 11. By moving the lever over the latching segment, more or less tension is put upon the auxiliary spring, which augments the main governor spring. Moving the lever downward increases the speed. This device facilitates synchronizing the generators and also serves to adjust the proportion of the load taken by each engine. The picture also shows the method of driving the governor and the cam shaft by a drag crank from the main crank pin. All of the driving mechanism is supported solidly on the heavy engine bed and is not affected by any misalinement of the main shaft. Furthermore, the speed of the governor is not affected by momentary irregularities in the cam-shaft velocity.

Water for the engine jackets and the scrubbers is furnished by a duplicate outfit of Watson-Stillman motor-driven twin-volute turbine pumps. Each of these pumps is rated at 9000 gallons per hour, delivering against 40 pounds pressure. The pumps draw water through about 325 feet of suction pipe with a total lift of about 10 feet. They can be primed one from the other, or by gravity from the water in the gas holder, to which they are also connected.

All the lubrication in the engine room is supplied from a Unit type oil filter furnished by the Burt Manufacturing Company, Akron, O. The filter is placed about 20 feet above the floor and the used oil is delivered to it by a small motor-driven Lobee rotary pump. The clean-oil compartment of the filter furnishes sufficient oil storage capacity. All oil us in the plant, except the cylinder oil, drained to a common tank below the fl level from which it is elevated by pump to the overhead filter.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Why the Regulator Wouldn't Regulate

By F. G. Heckler

It is well known that machinery depreciates when standing idle, but that a variable-speed controller for a 6-horsepower direct-current motor used for driving an Olsen oil-testing machine should fail to operate properly after three months' vacation was hardly to be expected.

The face of the starting box is shown in Fig. 1. The line current is brought in on the arm *A*; the field current is shunted through the contact strip *F*, passes through the magnet *M* and thence, by way of the binding post *B*, to the motor field winding. By moving the arm *A* to the right over the contacts *C*, *C'*, *C''*, *C'''*, the resistance in the field circuit is increased, which increases the speed of the motor. Attached to the arm is a strip *L* with a number of shallow grooves cut across its upper side. The field current in the magnet *M* magnetizes the core and attracts the armature *N*, pivoted at *P*, and presses the strip *L* against the knife edge *K*, which catches in the grooves across the strip *L* and thereby holds the arm in position for any desired speed from 800 to 1600 revolutions per minute.

The magnet held the arm in place, but it was impossible to get any increase of speed until the last step C''' was reached, when the speed would suddenly jump to its maximum value. It was noticed that in shifting from one contact point to another there was always sparking between the magnet core and the armature, indicating that there was current passing between the two, and this gave a clew to the trouble. There was a ground in the magnet winding which allowed the current to pass freely through *L*, *O* and *N* to the motor field winding, and the field current, therefore, was not decreased when the arm was moved to the right over the contact buttons to increase the speed. It happened that when the arm *A* was on the last contact button the air-gap between the core of the magnet *M* and its armature *N* was increased, preventing the passage of any current. This accounted for the increased speed at this point.

It was necessary to make at least a temporary repair immediately, and this was easily done by removing the extension *O* of the magnet armature and inserting between the two pieces, *N* and *O*, a piece of fiber, as shown in Fig. 2. The holes in the piece *O* were enlarged to permit the insertion of fiber bushings around the screws to insulate the shanks, and a strip of fiber under the screw heads insulated them there. This repair was made in a short time and avoided the delay of securing a new magnet.

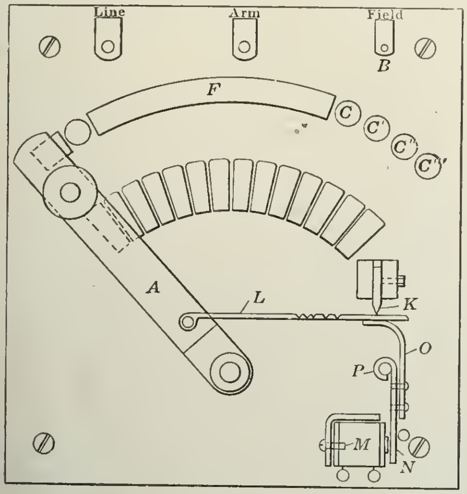


Fig. 1. Mr. Hecker's Motor Controller

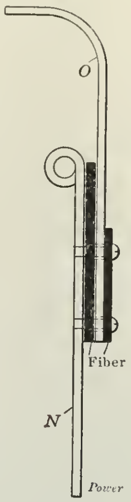


Fig. 2

Some Reminiscences of an Old Construction Man

By A. E. Walker

Some years ago I was engaged with a firm of electric-light contractors, who fitted up a number of small private installations mostly driven by gas engines supplied with town gas, there being at that time no public supply of electricity available in the town. In the course of my three years' experience with the company, I helped to install a number of these plants, and often had to run them for a few days until the gardener, coachman, errand boy or chief cook and bottlewasher felt quite at home with the equipment and could take charge.

Of course we were often sent for to find where the trouble was when anything went wrong, and some of the mishaps were very amusing.

One private house was in the dark all night because the gardener-electrician hadn't noticed that the bottom brush of the dynamo had fallen out onto the floor, there being only one on each stud and the machine being bipolar. We were, of course, called up in the morning, and the gardener related his woes, how he had put more oil on the bearings, put a new ignition tube in the engine, and tried every other stunt he could think of! His face was a picture when the real cause of the trouble was pointed out.

Another plant we fixed up consisted of a 15-kilowatt gas-driven dynamo and a storage battery. There was an automatic switch on the positive terminal of the dynamo which cut the machine in as soon as it came up to a high enough voltage to charge the battery, and cut it out if the voltage dropped.

A knife switch was also provided at the negative terminal to be opened when the dynamo was shut down, as a safeguard against the automatic switch being closed by accident.

One day the attendant shut down the dynamo and forgot to pull out the knife switch, and as luck would have it, someone got fooling around the engine room while he was away and pushed the automatic switch in. Results: Switch burned up, dynamo and battery fuses blown and a lot of plates in the battery badly buckled.

We once had a compound-wound dynamo sent into the shops to have a new coil put into the armature and be generally overhauled. When we got it together again and on the test block, belted to a gas engine, it refused to excite. The dynamo pulley was small for the belt speed of our driving wheel, so we were running the engine slowly to get the right speed on the dynamo. The foreman then had the engine run up to its normal speed which made the dynamo run about 50 per cent. fast. Still no signs of any juice on the bank of lamps or the voltmeter connected across the terminals.

All the wire ends were led to one terminal board on top of the machine; so it was easy to get them mixed. We had a council of war, and the foreman said the series winding was opposing the shunt, but I said it wasn't; however, he tried crossing over the series without any effect, so put it back; then he said she must have lost her residual magnetism. I suggested that perhaps the shunt current was opposing the residual magnetism, and after a lot of argument he said he would cross it over just to show me I was wrong. He did so, and immediately he got the last wire end into the terminal up came the voltage, and with the dynamo going at increased speed nearly blew all the lamps to bits, every lamp looking like a small arc.

Unbalanced Airgaps

By R. A. Cultra

The articles published in recent issues of Power under this title seem to apply to induction motors only. A good deal of trouble, however, can be caused by unbalanced airgaps in dynamos, especially direct-current machines. I had a 525-kilowatt direct-current generator in a power plant which behaved quite badly. The voltage fluctuated excessively and the brushes sparked some, and, although nothing indicated any very serious trouble, the machine ran very unsatisfactorily. After shutting down one night, while examining the brushes and brush-holders, as usual, I noticed that the bottom holders were much closer to the commutator than the top ones, and I then measured the distances from the armature core to the polefaces, top and bottom; the airgap below the armature was about $\frac{1}{8}$ inch and the one above it was about $\frac{3}{4}$ inch.

The engine was a cross compound and the bearing surface of the boxes had been criticized as insufficient by numerous mechanical engineers who had visited the plant. These boxes had always run very warm and the length of each one was barely two diameters of the shaft. The neglect of the oilers, combined with the inadequate bearing surface, had caused the babbitt to wear very thin in the bottom shells, lowering the armature and producing an unsymmetrical airgap. The armature did not strike the polefaces but it would have done so if allowed to run much longer.

Large shims were made to slide under the bottom halves of the boxes, and enough of these were put under the shells to center the armature between opposite polefaces. The result was much more than expected. Although the armature had been centered only for the purpose of preventing it from striking the magnet poles, we discovered that the machine ran very steadily and without sparking after this was done. Of course, after the difference in operation was noticed, it was easy to reason out the cause, which was the inequality between the electromotive forces generated in the different branches or paths of the armature winding.

LETTERS

Mr. Earle's Pump Motor Control

In reply to W. H. Earle's query in the September 6 issue, it is scarcely necessary to say that it is possible to rig up a water rheostat, but the expense and consequent trouble involved thereafter make it a very poor device for speed control. I would not consider it good practice at all to install a pump of the size he has without having a speed controller instead of the "usual form of starting box." An outfit of the capacity mentioned should have a speed range of from one-half to full speed. With a well designed controller, that is, one that does not increase in temperature too much at low motor speeds, the controller lever can be connected to a float in the sump so as to hold the water at practically any desired level. If the minimum speed of the motor is still too fast after the controller is put in, some dead resistance can be inserted in the armature circuit. Of course, it is much better to determine just what the minimum speed is to be and have the controller to regulate accordingly.

I would think that considerable trouble would result from trying to have a steady flow of water through the rheostat. I have yet to see the first circuit in a wet mine that was not troubled with numerous grounds, and the first time there was a ground on the opposite side of the line from the water rheostat, there would be a short-circuit on the line. If a water rheostat is used, it should be connected between the motor and the negative side of the line, as there will not be as much potential to ground as if connected between the motor and the starting box. I am assuming in this case that there is a trolley line in the mine and that the negative side of the system is grounded. If there is a two-wire circuit and no part of the system is grounded it will make no difference which side the rheostat is on.

I have one 8x8 triplex pump driven by a 30-horsepower 250-volt motor, and the speed range is from 500 to 1000 revolutions per minute. At the lowest speed the water is kept down nicely, but if for any reason it should be desired to run the motor at a lower speed, extra resistance could be inserted and the float and controller rigging could be put on, if it were desired to have the water level regulated automatically.

L. Earle Brown.

Roundup, Mont.

Mr. Earle is extending open arms to trouble for himself by considering the use of a water rheostat as described. I have had considerable experience with pumps operating under such conditions and have found that the best apparatus to be used is an automatic motor starter in the pump room, controlled by a float in the sump to start and stop the motor when the water has risen or fallen to certain predetermined levels in the sump. I know this arrangement will give satisfactory service if there is a good foot valve on the pump suction, whereas the water rheostat called for in Mr. Earle's letter will be a continual source of trouble.

W. H. Keller.

Ansted, W. Va.

Three Wire vs. Two Wire Generators

In his letter, published in the issue of September 20, Mr. Reynolds expresses the opinion that I am operating under conditions particularly favorable to the three-wire generator. This is not the case, even as to average conditions, and our conditions vary a good deal. I have at times an unbalanced load of 125 amperes and the voltage remains the same; we never see any fluctuations in voltage. At times an elevator load of 400 amperes comes on all at once and this is not even noticed. I do not think it good practice to install two machines, giving each one half of the load, when one generator can do the work.

The three-wire generator, with collector rings and choke coils, is as simple as any other generator but a man must study the machine the same as he would any other one. In the March 15 issue of Power the three-wire generators of the Westinghouse and Western Electric Companies were fully described and diagrams of both were shown. All young engineers and electricians ought to keep posted on these generators for they are the coming thing.

Walter W. Smith.

Chicago, Ill.

A pulley of compressed paper pulp made in the form of a ring, sawed in two and mounted with hubs and arms of either steel or wood was exhibited at Brussels. It weighs considerably less than one-half what an iron pulley does and furnishes an admirable surface for the belt.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Wrong Valve Setting

A certain plant has a 300-horsepower noncondensing tandem-compound McIntosh-Seymour engine, belt connected to a direct-current generator which is held in reserve and operated only at times of heavy load. It was noticed that whenever this engine was operated, it seemed to pull the steam right out of the boilers.

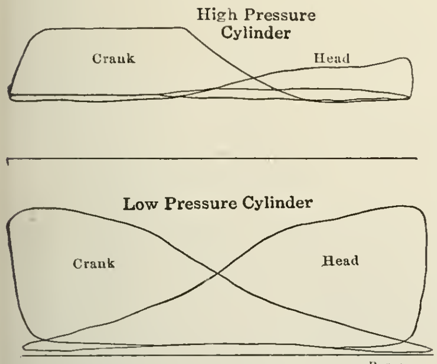


Fig. 1. Indicator Diagrams

The engineer, who had not had charge of this plant very long, undertook to find out why this engine consumed so much steam. Indicators were put on and diagrams taken, samples of which are shown in Fig. 1. The diagrams obtained on the low-pressure end were fair but those on the high-pressure end were very peculiar.

It was thought that the steam valve was cutting off too early, consequently that was lengthened, but with no better results. Next, the main valve was readjusted, but still the cards were about the same. The covers to the steam chest were removed and the steam valves set correctly but still the cards were not much better. The covers to the exhaust chamber were then taken off and it was discovered that the exhaust valves were set so that they were open during admission; this allowed the steam to blow through them into the receiver, with the result that the low-pressure cylinder was performing most of the work.

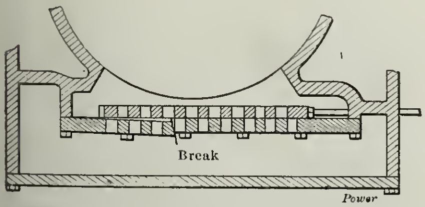


Fig. 2. Section through Exhaust Chamber

Upon opening the exhaust chamber on the low-pressure cylinder, it was found that the exhaust-valve seat had become broken, and as the tap-bolts which held the seat in place had become loosened, the seat had dropped slightly and let steam past when the valve was closed. Fig. 2 shows the broken seat. When a new exhaust-valve seat had been procured and the valves set properly, the engine ran very economically.

It was learned later that the man who had formerly held the position of chief engineer had had occasion to take out the valves in the high-pressure cylinder and upon putting them back had experienced great difficulty in making the engine run. When he did finally succeed in making it run he was satisfied to leave the valves as they were.

George F. Atwater.

South Norwalk, Conn.

Pump Packing Kinks

I had a pump with 8-inch water cylinders in which the packing wore out very rapidly. An examination showed that the brass linings of the cylinders had worn through on the bottom, leaving jagged edges which cut away the packing.

Removing the piston from the rod, I took some long cape and gouge chisels about ⅛ inch wide and chipped a groove through the remaining solid portion of the lining on the bottom and removed the lining. I then calipered the cylinder, measured the length of the sleeve and ordered a new lining. In order to use the pump I secured some tucks packing, about ⅛ inch thicker than usual, to make up for the thickness of the brass sleeve, and cut it into rings, beveling the ends to overlap each other. I then replaced the piston and the packing, put on the cylinder head and after packing the stuffing boxes started the pump which gave good service until the new brass sleeve arrived. Then, removing the piston from the water end as well as the packing in the stuffing boxes and the rocker-arm connection from the rod, I removed the rod through the steam cylinder and inserted a long 1-inch bolt, having a long thread at one end, into the steam cylinder in place of the rod. A large iron washer was placed against the end of the sleeve on the rod, and the nut screwed up until the sleeve was drawn into place. The bolt was then removed and the rod replaced, also the piston and packing, and the stuffing boxes were packed.

The second cylinder gave the same trouble a month later and the same treatment was applied, thus putting the water cylinders in excellent condition with the result that the packing lasted eight months. This will appear to the average engineer as a hard or impossible job to be accomplished away from a machine shop, but it is much easier when the piston has been removed.

R. A. Cultra.

Boston, Mass.

A Boiler Feed Pipe Repair

Upon taking charge of the steam plant of a manufacturing establishment I made a tour of inspection with the owner and upon entering the boiler room he remarked, "These two boilers blow off at 115 pounds pressure, the feed pipes leak badly at the boiler heads and they must be repaired. The boiler inspector is out of patience with us for neglecting them so long, and in his last report says they must be made tight before his next visit to our works."

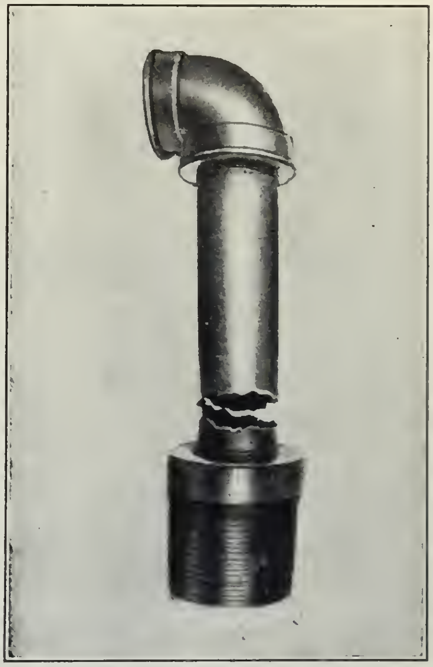


Fig. 1. Broken Feed Pipe

Upon examination I found that the leaks were at the point where the brass bushings entered the front tube sheets and, judging from the appearance of the iron feed pipes and the tube sheets, they had been leaking for a long time, as both were badly eaten away by the corrosive action of the escaping water mingled with the soot.

The next Sunday morning when I began to put in new bushings I was surprised to see the feed pipe break off when I undertook to disconnect it from the elbow. A picture of this broken feed pipe is shown in Fig. 1. At the point of fracture the pipe was only 1/32 inch thick on the under side and 5/32 inch on the upper side, and it is a mystery to me how the pipe withstood a pressure of 115 pounds per square inch without breaking.

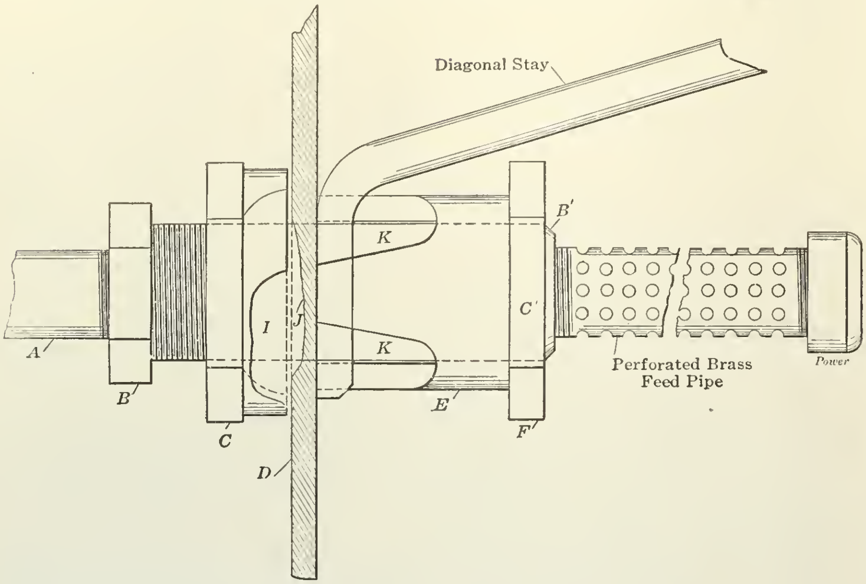


Fig. 2. Special Bushing for Boiler-feed Pipe

Upon removing the bushings I found a portion of the tube sheet to be eaten away as shown at *J*, Fig. 2. This sheet had been only ½ inch thick when new and as the bushing had eight threads per inch, only two threads were available; this was insufficient to make a tight joint even with the new bushings.

I accordingly had the arrangement, shown in Fig. 2, made at a local foundry and machine shop. The brass bushing *B B'* extends from *B* to *B'* in order that nut *C'* may clear the diagonal stays. Locknut *C* has a cavity *I* to receive the packing. Sleeve *E* is slipped over the bushing *B B'* and has openings *K* to straddle the foot of the diagonal stay, also to furnish a bearing against the tube sheet at three points to prevent rocking. Nut *C'* is screwed onto the bushing and presses against the sleeve *E;* this reinforces the tube sheet and prevents the bushing from being pulled out when nut *C* is screwed against the packing in the the cavity *I*.

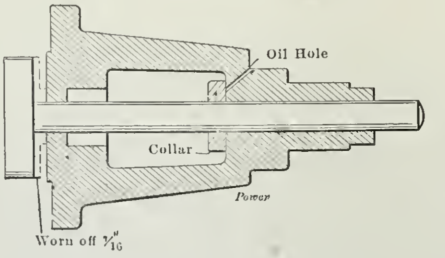
After the new fittings were installed, I wrapped the bushing with 3/16-inch packing, winding tightly in a clockwise direction and filling the cavity so that there was a space of 3/16 inch between the tube sheet and the nut; then the latter was screwed up tightly. We have had no more trouble with leaky boiler-feed pipes since making this change.

J. W. Parker.

Clinton, Mass.

Corliss Valve Stem Repair

The valve stem was on the high-pressure cylinder of a 26 and 42 by 34-inch cross-compound Corliss engine which was new. From the beginning the valve had shown a tendency to groan, but it ran a couple of months before we noticed that this particular valve stem was projecting more than the rest. It constantly grew worse, until by the time we had an opportunity to dismantle the engine, the stem projected at least 7/16 inch. Upon removing the bonnet and valve stem we found that not only had the collar been worn off but also about ¼ inch of the T-head of the stem, although the bonnet against which it had rubbed did not show the slightest sign of wear. The stem was steel and the bonnet was cast iron. The stem was packed with metallic packing which leaked a little and the only way we could explain such unusual wear was that the water leaking through the packing had washed away the oil and thus caused the damage. The surfaces of the valve and the cylinder showed that they had been receiving sufficient oil.



Collar Holding Valve Stem in Place

The method of repair is indicated in the accompanying sketch. We filed the inner surface of the outside bearing of the bonnet square with the stem and fastened a collar on the latter with a set screw arranged so that the end thrust was taken up between the collar and the bonnet. We also drilled an oil hole through the bonnet so that we might be sure that it was getting the proper lubrication.

Most engines are fitted with such a collar, or its equivalent, when they leave the shop, so I do not make any claims for originality, but I believe it is better to have the collar outside of the cylinder than to have renewable rings inside.

A. E. Mueller.

Mayville, Wis.

Filing Catalogs

Here is what I believe to be a good system of filing catalogs. First have a wooden box made about 5 feet long, 2 feet wide and 2 feet deep, in which to keep the catalogs. Pieces of white cardboard 2 feet square can be used to separate them. Number the cardboard partitions with large figures in red ink from one up to as many different classes of catalogs as will be handled. Then sort out the catalogs, placing those treating of the same subject in the same compartment.

For example—Suppose there are some catalogs on boilers, engines, electrical fittings, etc. Place those dealing with boilers after No. 1 cardboard, those dealing with engines after No. 2 cardboard, those on electrical fittings after No. 3 cardboard and so on. Another way is to place them in alphabetical order, in which case an index book should be kept similar to the following:

Name of Firm.	Subject.	Card Number.
Allis-Chalmers Co..........	engines	5
Allis-Chalmers Co..........	turbines	20
Allis-Chalmers Co..........	pumps	16
Babcock & Wilcox Co......	boilers	2
Harrisburg Mfg. & Boiler Co.	boilers	2
General Electric Company..	motors	13

Where there are catalogs of different firms treating of the same subject, it is well to place cardboard between each different catalog and mark each with the first letter of the firm's name. These cardboards should be made a little smaller than the others; this will make it easier to find the cardboard which is wanted.

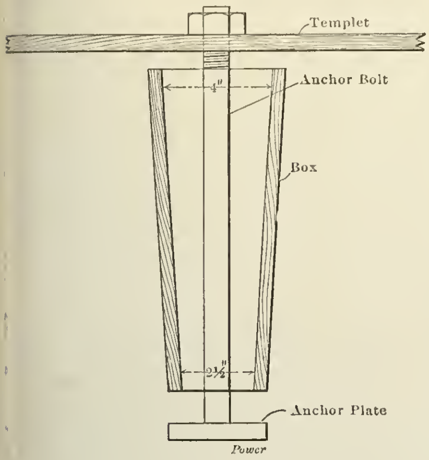
A system such as the foregoing saves considerable time as it does away with looking over the whole collection of catalogs to find the one wanted.

Henry H. Cantor.

Sheboygan Falls, Wis.

Inserting Foundation Bolts

Engineers use various forms of boxes and pipes around anchor bolts in engine foundations. About the best that I have seen came under my notice recently in the setting of foundations for two 360-horsepower engines. The erecting engineer had four boxes made of 1-inch boards which were long enough to reach



SECTION OF FORM

rom the templet to within 2 inches of he anchor-bolt plates; these boxes were 2½ inches square at the bottom, and 4 nches square at the top, inside measurements. After assembling the boxes they were put under water and left for two ays in order to swell them. Then they were slipped over the anchor bolts with the mall end down. As the foundation dried ut the boxes gave up the moisture in them and contracted, becoming so loose hat it was an easy matter to lift them out f the foundation. When the engines were set, every crevice around the bolts was filled with cement, this making a better job than by placing pipes around the olts.

W. H. KELLER.

Ansted, W. Va.

Municipal Ownership

Has a municipality the constitutional ight to own and operate public utilities or the benefit of its citizens; has it the ight to construct and maintain power lants for the purpose of supplying the ity with electrical energy for light and ower; has it the right to install a water ystem for the use of its residents; in ompetition with privately owned and established corporations?

The equity of municipal ownership, nd its limitations based on state-constitutional provisions, are in litigation in he courts of California. San Francisco with the hetch-hetchy water scheme, Los ngeles with the aqueduct hydroelectric-power project, the city of Imperial with esigns for a self-controlled water system and the municipality of Madera with ke contentions, are engaged in legal uits to test the interpretation of the law. The latter, the most diminutive of the group, has been assisted by the others in its water-works case, which directly involves the general principles for which all are contending.

The decision in this litigation given by Judge Wellborn, of the United States District Court, on September 13, is of significant importance to municipalities throughout the country. He says in part:

"Both reason and authority sanction the doctrine that whatever the nature and extent of a franchise, restriction of legislative power is in issue; no matter through what agency of government the franchise was conveyed, whether a constitutional convention, a legislature or a municipality, the rule of strict construction against the grantee uniformly prevails, it must be held that the complainant's franchise does not shut out municipal competition. By no rule of construction, however, whether literal or in favor of the public, or such as applies to the ordinary agreements between individuals, can a different conclusion be reached. If it were ever permissible to enlarge by implication the grants of special privileges affecting the general interests, where can there be found in the case at the bar any justification for such an enlargement?"

This considerate judgment is far reaching; it has given an impetus to the remaining actions and the outcome will be of wide general interest. As in the course of such events, the Madera case will find its way to the Supreme Court, which tribunal, up to this time, has always sustained the decisions of Judge Wellborn.

L. R. ALLISON.

Los Angeles, Cal.

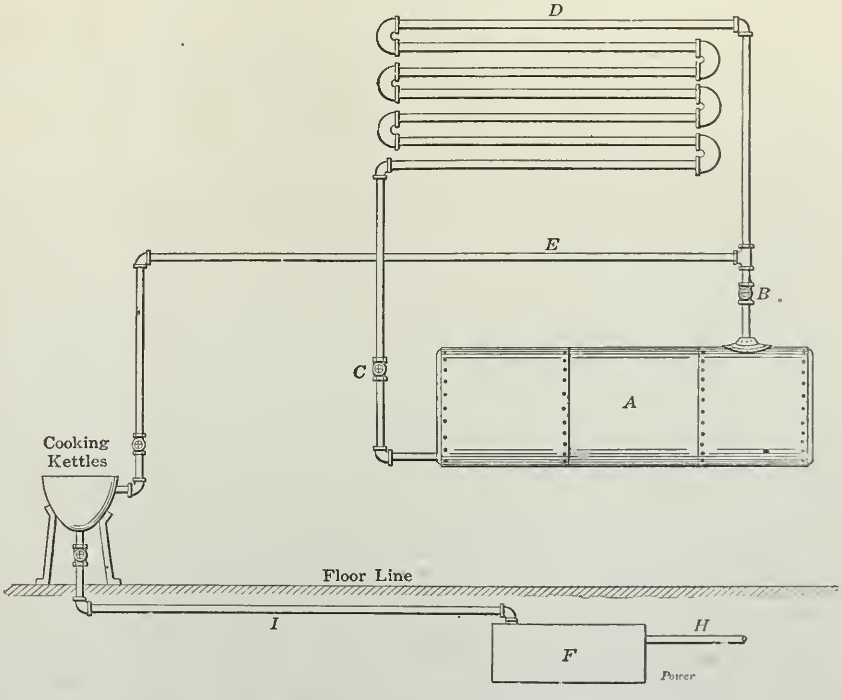
Disappearance of Water in a Boiler

An inexperienced man was placed in charge of a steam-heating plant, part of the equipment being arranged as shown in the accompanying diagram, in which *A* represents the steam boiler, *B* the main stop valve, *C* the main return steam valve, *D* the heating coils and radiators, *E* the steam piping leading to the jacketed cooking kettles, *F* the hotwell receiving the condensation, *H* the overflow to the sewer and *I* the return to the hotwell from the kettles.

To make some repairs in the pipe work, valves *B* and *C* were closed, and when the repairs were completed *C* was opened and through neglect on the part of the inexperienced attendant *B* was left closed. A number of the valves on the kettles were open and there were no check valves on the returns from the radiators. When the pressure began to rise in the boiler the water was forced up the return, through the radiators, along pipe *E*, through the kettles to the hotwell and thence to the sewer. As a result great excitement prevailed for a time in the building; no one could tell where the boiler water was going. The city pressure was turned into the boiler, but the water went out faster than it could be put in. The fire was hastily drawn and an experienced man sent for. When he arrived, he located the trouble in a few minutes, and the owner concluded that it was wiser and safer to employ an experienced man to look after the plant.

JAMES E. NOBLE.

Toronto, Canada.



ARRANGEMENT OF PIPING

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Frost in the Engine Room

In the August 23 issue, Mr. Fenwick states that he experiences much trouble with frost in his engine room. I believe this is due to the galvanized-iron roof; the vapor in the engine room is condensed, and in very cold weather it freezes on the underside of the iron covering. When the inside temperature is sufficient to thaw this accumulated frost, the water drips over everything.

I have the same conditions and experience the same trouble. The only remedy is a layer of felt under the iron roofing. Cooling the engine room by opening the outer door will stop the drip, but this is objectionable when the temperature is very low.

I would not, on any account, advise an air duct to the ashpit. Air at a very low temperature is not wanted below the grates; in fact, I believe that the air should first be heated in some way by the waste gases.

I have seen frost all around the ventilators on the furnace doors with the fires going full blast and all the doors and windows closed.

John L. Sheldon.

Prince Albert, Can.

In the August 23 issue, Mr. Fenwick raises the question as to the cost of frost accumulating upon the walls and ceiling of an engine and boiler room.

In the first place, what are the necessary conditions to produce frost? In the case of air and water vapor we find that for each particular temperature there is a maximum density and if we compress vapor beyond this density, a portion of it will be liquefied. This is known as the saturation pressure. Also, for each pressure there is a minimum temperature and if we bring the temperature of the vapor below this, condensation will take place. This temperature is known as the condensation temperature or dew point. To state this in another way, we may say that if air is at its dew point, any decrease in temperature or increase in pressure will cause liquefaction. If the surface on which this condensation falls is at a low enough temperature, heat will be withdrawn from the liquid until it congeals and forms frost such as may be seen at any time on a bare brine pipe in a refrigeration plant.

Applying the above to the problem under discussion, it is evident that the air in the building, which, due to radiation from boilers and engines, is presumably at a temperature of about 60 degrees, will, on rising and passing near the walls and ceiling, be cooled below the dew point by radiation from the walls, which are extremely thin, and by mixing with outside air entering through crevices as suggested by Mr. Fenwick. The moisture thus deposited on the walls and ceiling is rapidly turned to frost, due to the heat in the liquid being radiated through the walls.

The only method of preventing this deposit of frost is to so increase the nonconduction of the walls and ceiling either by another thickness of boards so placed as to have a 6- to 8-inch dead-air space between, or by good nonconducting sheathing, that the walls will not be able to radiate heat fast enough to bring the temperature of the air below the dew point.

W. L. Durand.

Washington, D. C.

Water in Superheated Steam

While there is no doubt that good judgment must be used in applying a calorimeter to a steam line if the average quality of the steam passing through it is to be correctly indicated, I think that W. H. Booth's statement on page 1501 in the August 23 issue, that "All calorimeter tests are, therefore, a delusion, unless they are made on superheated steam," is rather broad and misleading.

Why should not a calorimeter test of superheated steam be as liable to error as one made of saturated steam? It is evident that Mr. Booth wishes to imply, if he does not directly so state, that water and superheated steam cannot exist in a pipe line at the same time. This is, however, not a fact, as many engineers of wide experience can testify, for it is not an unknown fact that trouble has been caused by water in pipe lines which carry superheated steam. That the presence of water and superheated steam together in a pipe line does not conflict with any of the theoretical considerations of the subject, was most ably pointed out in an editorial published in the August, 1907, issue. As a means of illustrating the point, attention was called to the fact that ice cannot long exist in an atmosphere above 32 degrees Fahrenheit, but still we have the ice man with us, who delivers his goods in an open wagon through the streets, in the hottest days of summer. Water can be delivered through a pipe line that is carrying superheated steam in the same manner, the only requirements being that the water be not in a too finely divided state, and that the time of contact between the water and the steam be not too great. I do not believe that a thermometer can always indicate what is passing through a pipe carrying superheated steam as definitely as a good calorimeter, intelligently applied.

J. E. Terman.

New Haven, Conn.

A Problem in Economy

I read Mr. Lewis' letter, in the August 30 issue, inquiring how the efficiency of his plant may be increased. I think that it would be poor policy to use compressed air in the place of steam for forcing the fires, during the summer at least, as it would require more steam to furnish the additional air than would be saved by cutting out the steam at the jets.

No doubt there is an excess of exhaust steam as it is. If it requires more steam to heat the buildings in the winter than the compressor furnishes by its exhaust making it necessary to draw live steam from the boilers, then it would pay to use air in place of steam at the jets fo forcing the fires.

If the turbines exhaust into a con denser and there is no possibility of need ing more than the 350 cubic feet pe minute for some time to come, then i would probably pay to install a motor driven compressor. But the feed-wate temperature would be lower unless ther was sufficient exhaust steam from th feed pumps and pumps supplying con densing water to the turbine. Even i the turbines exhaust to the atmosphere would probably pay to get rid of the larg compressor and install a motor-drive one in its place.

In regard to the saving, there shoul be a substantial gain in replacing th steam-driven compressor with a moto driven one, as it should be possible t purchase the small outfit for consider ably less than the larger outfit shoul sell for if it has been kept in good con dition. Then the water rate for the stea turbine should be reduced considerabl Also, there should be a saving in cy inder oil and packing over the steam driven compressor.

The actual saving could not be com puted unless the water rate for both th turbines and the steam-driven compress for different loads and the cost of coal per ton were known. If the shop or factory to which his plant furnishes power should require more air and electric current in a few years, it would be best to let the machines run as they are, because they will also need more heat if the plant is enlarged; then the additional exhaust steam will come in handy for heating in the winter.

As he has not stated whether there is enough exhaust steam for heating without taking live steam from the boilers and also has not said whether the turbines run condensing or not, it is a hard matter to make a positive answer as to whether it would pay to make the changes he proposes or not.

GEORGE DREYER.

Gilsonburg, O.

H. M. Lewis, in the August 30 issue, asks if it would not be cheaper to use compressed air for forced draft instead of steam, thereby increasing the load factor of his air compressor so as to enable it to operate at higher efficiency.

While it would, no doubt, raise the efficiency of the air compressor, it would be a bad proposition on the whole, as it would be cheaper to use the steam direct than to use the steam to compress air and then use the air for blowing the fires. Even if the efficiency of the combined engine and air compressor were 100 per cent. which is, of course, impossible, there would be nothing gained.

As regards running the air compressor with a motor, and obtaining current from the turbo-generator sets, the economy will depend on what drive his air compressor is equipped with at present. If it is driven by a wasteful engine, the turbine-generator-motor outfit might have the best of it, but if driven by a Corliss condensing engine in fair condition, it would appear that it would be another case of "going 'round Robin Hood's barn."

LEAL DAVIS.

San José, Cal.

Bearing Lubrication

Intelligent criticism is illuminating and to me always most welcome. I am not averse to being intelligently criticized providing it leads to the development of an idea or shows the falsity of a certain reasoning. With respect to Mr. Taylor's criticism in the August 30 issue of my article on "Oil Grooves and Bearing Lubrication" in the July 17 issue, I wish to say that although a hollow crank pin carrying a tube of oil or grease may be "an old idea," it is, nevertheless, a good one and along the line of progress.

With respect to the hole weakening the pin, I wish to say that it is the best modern practice to use hollow crank pins and shafts as such pins and shafts are stronger than the solid kind, which can be easily demonstrated all things else being equal.

With respect to "soft boxes," they have an inherent disadvantage which is obvious and I need not dwell further on this subject. Since the introduction of "hard boxes" or bearings at this plant, armature repairs have been successfully eliminated. Previous to their use an armature winder was constantly employed. In the near future there will be no more "soft boxes" in use. An engineer who has spent 20 or 30 years in one plant is, no doubt, familiar with the practice in that plant but there his experience ceases, and he is not in a position to know of the methods adopted elsewhere. His knowledge must necessarily be limited to that one plant.

With respect to any "misleading information" in my article, permit me to add that I will give a five-years' subscription to POWER to any person that the editor of POWER may designate if such person can point out any "misleading information" therein.

WILLIAM KAVANAGH.

New York City.

The Divided House

On the frontispage of POWER for August 30 some statements were made in regard to an assistant engineer which are worthy of much thought. It is surprising to find conditions similar to those described in so many power plants.

Having been in a position to see the operation of jealousy among power-house employees from all sides, I find that many engineers seem to forget that while the owner of the plant may pay them, and they are working for him, the same man hires the chief engineer to direct the work and looks to him, not the assistants, for proper operation and results.

If results are satisfactory, it will not concern the company how they are brought about, always providing that they are accomplished with safety and economy. Now, the man with a "grouch," who imagines that he is the hardest worked engineer in the country and is not appreciated, will often advance the statement that he will run his watch regardless of what the chief will say, for he claims that he is not working for the chief but for the company. A chief engineer hires a man to run a watch or to do other duty about the plant and he is called an assistant. This word means a man who assists, not one who interferes or prevents, and every man in this position who does anything else is not an assistant but a drawback; also every man who tries to defeat the purposes of a good chief engineer is not working for the interests of the company.

I know of an instance a few years ago where a man imagined himself in the position of the one mentioned in the editorial. He had started as fireman and worked up to first-assistant engineer. Having had but little previous experience he was in no position to judge of what he should have, and in reality he had the best position possible for an assistant in that plant. However, he tried to make those who were working with him discontented, and also tried to go around the chief and get relief from the management. This man had reached the point where no amount of reasoning or argument would convince him of the proper arrangement of things, and that he had many troubles on his watch which could be traced to his personal neglect. He sank to such a position that when he left there was no feeling of loss or of any vacancy to fill. He has not had as good a position since.

In another case, the assistant was continually complaining about adverse conditions, class of machinery, division of help, etc., and he made the statement that he was in supreme command over his watch for he was responsible for its operation to the company. Now, this idea, I hold, is entirely wrong, as any company which places a man in charge of its plant should hold him responsible and not the assistants, and he in turn should demand of his assistants such faithfulness and obedience to his wishes as will insure good service.

I also believe that if any engineer goes into a plant with the idea that he is there to aid the chief in every way possible, and to anticipate his wishes even if he does a little more than is expected, he will gain both in reputation and self-respect. If, however, he is working for a man who does not appreciate his efforts and who cannot give him encouragement to continue, then let him give his notice honorably and without any useless argument, for such things always take something from a man's character.

It should be the aim of all men to conduct themselves in such a way that their employers will look upon them as honorable men, even if there is a difference in ideas such that it is impossible for them to agree. Many a firm has kept a man whom they knew was reliable, truthful and temperate rather than change for one who might know more about the business but who was unknown to them personally. As Mr. Wakeman says in one of his books, the engineer should do all these things, for "his reputation for honorable dealing is at stake."

On the assistant's side, I would say that the chief should always give his orders directly to him, as any other arrangement will give the men under the assistant the idea that he has no authority. If the chief supports his assistants even as he is supported by the management, the house will be united.

These statements will prompt many to say that such conditions are impossible in all plants; but if jealousy, deceit and other undesirable traits are cut out, the spirit of fair play and a desire to pull together substituted, the results will amply justify the change.

G. H. Kimball.

East Dedham, Mass.

What Causes the Scale?

I noticed in the August 30 issue the inquiry by Mr. Lynch who desires to obtain information as to how to prevent the formation of a black vulcanized rubber-like scale which accumulates on a steam-heated cast-iron form which he uses for bending, or shaping, certain kinds of wood, the wood being dipped in hot water before being used.

To illustrate how a black coating may be formed on heated iron under certain conditions, and by frequent application, or continuous action, be increased in thickness until it resembles a scale, a flat cooler in an artificial-ice plant will be taken as an example. The cooler, say, is of the atmospheric type.

The distilled water from the reboiler enters the cooler at a temperature of 180 degrees Fahrenheit, and the cooling water going over the cooler has a temperature of 70 degrees. If the cooler is allowed to operate without being blown out with steam, as they usually are for a certain purpose, and the cooling water has certain impurities in it, a green fungus will, after a few weeks, begin to make its appearance on some of the coils. It will extend about four pipes high from and including the bottom pipe, or to a point where the temperature of the descending cooling water has increased to such an extent by contact with the pipes near the bottom as to cause the formation of the growth.

This fungus is, I think, similar in nature to the substance which is giving Mr. Lynch trouble, only it is not "painted black" as yet. However, if the distilled water is cut off from the cooler and steam blown through, the steam-heated pipe causes the green matter to turn black, leaving a thin black coating on the pipes.

It occurred to me that Mr. Lynch might be using water containing impurities similar in nature to those in the water referred to. When the wood is dipped in the hot water the impurities contained in the thin layers of water which adhere to each section of wood may be thrown off by coming in contact with the heated form. If the form were operated at a lower temperature the scale might assume a different appearance; the high temperature naturally burns it black.

Layer after layer of impurities might be formed in this manner on the surface of the hot iron until a more or less thick scale resulted.

If Mr. Lynch is not using distilled water to dip the wood in, he might try boiling the water before using it.

W. W. Hailstone.

Youngstown, O.

Incompressibility of Water

Referring to the letter on "Incompressibility of Water," in the August 30 issue, I think Mr. Edgar has overlooked the fact that the pressure causing the decrease in volume of the water would also have a like effect on the iron. Although I can find no figures showing the relation between the volumes and pressures of metals in the solid state, most writers agree that solids are compressible, some even claiming that solids are about as compressible as water. This assumption is based on the fact that there is comparatively little difference between the volume of a body in the solid and in the liquid state. Therefore, it is reasonable to suppose that even to extreme limits of pressure, the ratio of the weights of equal volumes of iron and water will not differ much from that of 450:62.5, or 7.2, the specific gravity of cast iron. If the rate of change in volume is the same for both iron and water, the iron would sink until the bottom was reached, irrespective of the depth. Granting some difference in the compressibility in favor of the water, the distance at which a cubic foot of each would weigh the same as the iron would be a depth that would have no meaning so far as terrestrial distances are concerned.

W. G. Hawley.

Scranton, Penn.

Dissatisfaction and Jealousy

I have found that the first page of Power is always full of truisms. On reading this page in the August 30 issue, I just closed the book and patted it on the back for I have been there.

In the beginning of my career I was an ordinary fireman, shoveling coal into furnace doors along with several negro stokers in a Southern power plant. The chief was a Massachusetts man, highly technical and a strict disciplinarian. Often, unknown to me, he stood behind me as I was firing, and when I had finished he would insist that I open the furnace door and pull out all lumps of coal larger than a man's fist. This was hot work and it seemed so unnecessary that it kept me mad all the time until it finally occurred to me that in order to avoid hauling out these big coals I had better not put them in. I adopted this plan only to find that the chief was just as ready to find fault with me for other things; he continually called my attention to the dirty floor, and insisted that I brush it up after each firing. In view of the fact that jobs were not plentiful and times were hard, I put up with this "foolishness," but in order to avoid continual use of the broom I learned to handle a scoop without spilling coal all over the floor. The rims of the steam gages also came in for criticism, and he discovered that the feed-water line running across the boiler fronts was brass covered with a thick coat of black paint. I then had to get busy and clean these pipes till they shone like new and keep them that way, in addition to doing my share of the firing.

I knew that I was being imposed upon as the darkeys had much easier times than I did, and received the same pay. However, in a year's time the size of the plant was doubled and the position of head fireman was created; I was selected to fill this position at slightly increased wages but elevation seemed only to aggravate things; the chief was more exacting than ever, nothing seemed to suit him, he found fault with nearly everything I did, and if I did an extraordinarily good job he never noticed it but always found something that was not up to his idea of perfection. It took me about two years to find out that the chief was the best friend I had and that instead of imposing upon me he was doing me a favor every time he criticized my work.

However, I had ambitions to leave the boiler room and try oiling on one of the three big cross-compound engines. The three oilers on these engines had the distinction of calling themselves assistant engineers, while I was only the head fireman. It grated on my nerves to see these oilers leave the plant as soon as the wheels stopped while I usually had to stay from a few minutes to a few hours on some dirty repair job. On Sundays I had to see that the boilers were properly washed and filled for service; and often I was called out of bed at midnight on Sunday nights to straighten out some trouble. All this was unpleasant, in view of the fact that the boys in the engine room were always telling me of the good times they had Saturday afternoons and Sundays.

After a while the chief went back to Massachusetts and a new man took his place. During the change I secured the coveted position in the engine room. Then I found that keeping the floors clean and the bright work polished and chasing around after stray drops of oil with a bunch of waste was a very monotonous procedure; the chief and the machinist did all the engine repairing and about all I was learning was the fact that the former chief was a better friend than I had realized, and I was very glad to go to another plant in the capacity of head fireman. There I made up my mind that the next step was not to be an oiler but to be the chief myself, and I straightway began a systematic course of study, took a personal interest in the plant, voluntarily doing many dirty and disagreeable jobs that the chief himself never knew of, and in a couple of years I became so confident of my ability that I ventured to take charge of a small plant as superintendent, having secured a good recommendation from my former employers. In a couple of years more, by continually keeping after technical as well as practical information, I was able to satisfactorily fill a position in a larger plant, and have continued to progress until now I have the satisfaction of being consulted in mechanical matters where decisions involve thousands of dollars.

I am now thankful for the strict discipline imposed on me by those who were my superiors in my early days. I am also thankful that times were so hard in those days that I could not afford to get mad and quit.

In nine cases out of ten the mean boss is the best one; he is usually intent only upon the efficiency of the plant, and the efficiency of the plant is directly proportional to the efficiency of the men who do the detail work. A good fireman does not always rise to highly responsible positions but he stands a much better chance than the indifferent man and, whatever the duties, it always pays to do them cheerfully and willingly and in the very best manner possible. If conditions arise that do not seem just, determine whether they are for the good of the plant; if there is anything in favor of the plant then rest assured that no matter how unjust they may appear personally, the plant comes first. If the boss really "has it in for a man" in a personal way, he usually leaves no room for doubt, but discharges him. It rarely occurs that a man in authority will continue to employ a man in whom he sees nothing to build on, except in the case of the lowest menial laborers whom we are not considering.

A. C. Terlene.

Chattanooga, Tenn.

Piston Clearance

In the August 30 issue, P. S. Black gives his method of finding the piston clearance. This would not work with all engines, as there are a great many in which the piston rod is keyed to the crosshead instead of being attached by means of a screw and lock nut. The method I use and which will apply to any engine is to take out the crank-pin brasses, place the crank on the head-end center and push the crosshead back until the piston strikes the back head. Then, make a center-punch mark on the edge of the guide or frame to correspond with the corner of the crosshead or the center of the wristpin, whichever is most convenient. Do not use the crosshead shoe, however, as it may be moved later on, thereby throwing the marks out of the original position. Next, turn the engine over to the crank-end center, pull the crosshead up until the piston strikes the front head and make another point on the guide or frame to correspond with the point on the crosshead that was used before. Put the brasses back and turn the crank over to one center and measure the clearance on that end, which will be the distance from the mark on the crosshead to the mark on the guide. Then, turn the crank over to the other center and measure that end. With a rod keyed to the crosshead the clearance must be adjusted by putting in or taking out liners back of the brasses, depending upon the individual construction of each connecting rod. By using a square on the end of the wristpin or on some planed surface of the crosshead, the mark on the guide may be several inches from the one on the crosshead and still be exact so that the clearance can be measured as correctly in this way as in any other and the engineer will at all times have a permanent mark to check up by.

J. Case.

Hyattsville, Md.

The Expansion Valve

To continue under this heading used in the August 30 issue of Power, by J. J. Nash, considerably more information can be gleaned if we take the time to study the subject up. Many refrigerating engineers imagine that just as soon as the liquid ammonia passes the so-called expansion valve it immediately becomes a gas.

The conversion of water into steam through the transmission of heat into the water involves principles that are too generally understood to permit of discussion. If, however, we descend to degrees of temperature, which in everyday language imply cold, or lack of heat, the principle involved apparently assumes another aspect. Yet, there is no difference whatever in the causes which convert liquid anhydrous ammonia and water from the liquid into the gaseous state and no difference in the effect produced by each conversion.

Suppose that a thin stream of boiling water is continuously injected into a pipe coil which has been brought to a red heat. No one will question the fact that the injected water will immediately be converted into steam by contact with the heated pipe surface. Nor will one be surprised to learn that the pipe itself, beginning at the inlet end, will gradually cool down to the boiling temperature of the water. The water, still remaining at the boiling temperature, continues to flow on to where the pipe is still sufficiently hot to cause further evaporation of the water. The steam which is generated, in finding its way to the outlet of the pipe, though in intimate contact with the walls of the pipe, exerts but little cooling effect on them, owing to its limited capacity for absorbing heat or, in other words, its low specific heat.

When artificial refrigeration is considered, where the same process actually takes place though with different heat conditions, many engineers seem to be confused and take erroneous views of what is actually taking place. Such views are often held by men who have long been associated with this useful industry and who we would assume to be intimately acquainted with the process entrusted to their care and supervision.

No engineer has been known to call the valve through which he admits feed water to his boiler an expansion valve. Yet it bears the same relation to his boiler as the so called expansion valve bears to the system of freezing coils. In both cases its function is to regulate the feed. The simple experiment of evaporating a small quantity of liquid ammonia in a test bottle, where the conditions are more favorable for evaporation than those generally existing in a refrigerating plant, will convince the most skeptical that liquid ammonia does exist in the liquid state after passing the expansion valve. It should also convince him that the cooling effect is more intense on that portion of the bottle which is in direct contact with the liquid ammonia than it is on the portion of the bottle that surrounds the gas. This same process, which is so apparent in the bottle, takes place in the dark recesses of every freezing coil.

William G. Walters.

Stratford, Ont., Canada.

The Canton Boiler Explosion

The verdict of Coroner Harry A. March, as related in the editorial in the August 30 issue on the Canton explosion, seems to have caused an explosion of criticism. It is told that while Abraham Lincoln was engaged as a lawyer, he listened to a harangue by his opponent and after he had finished, Lincoln arose and said that his opponent reminded him of a Mississippi River boat that had a four-foot boiler and a six-foot whistle; when the whistle blew, all of the machinery was compelled to come to a standstill; similarly when his opponent began to talk his mental machinery for reasoning apparently ceased to act. It is my opinion that Mr. Harrington, who criticizes Mr. Allen in his letter on page 1698 of the September 20 issue, is in the same boat with Mr. Lincoln's opponent.

Mr. Harrington's criticism certainly must have been hastily prepared. He endeavors to make it appear that Mr. Allen advanced the theory that the explosion might have been caused by turning cold water into the boiler while it was red hot, when the most cursory examination of the evidence will show that the local boilermaker, Mr. Wheelan, was the one responsible for this theory, and not Mr. Allen. Mr. Harrington does not believe that waterhammer may be caused by turning steam into a boiler in which the pressure is lower than in the steam main. He may be interested to know that the production of waterhammer is far more certain in this manner than when the pressures are reversed. Mr. Harrington, although in the State of Massachusetts, and presumably holding an engineer's license to operate boilers in that State, seems to be badly mixed regarding the amount of pressure that would have been allowed on the Canton boiler should it have been located in the State of Massachusetts. The Massachusetts rules specify a factor of safety of five and three-quarters for lap-seam boilers over fifteen years old, and if the rivets in the Canton boiler were steel (which was doubtless the case), the Massachusetts rules would have allowed a pressure of 76 pounds, if the boiler was of the construction represented in the description in Power of May 31. This pressure is considerably above the "less than sixty pounds" given by Mr. Harrington. Even if the rivets had been iron, the allowable pressure would have been 68 pounds under the Massachusetts rules.

This boiler was operating on a factor of safety of about 4.4 at 100 pounds pressure, which can certainly not be severely criticized, for it is doubtless considerably in excess of the average factor of safety on which boilers are operated throughout the United States.

Francis B. Allen has long been engaged in the boiler-insurance field, and probably has investigated as many, if not more, boiler explosions than anyone else in the country, and under the circumstance it does not seem right to pass by unnoticed a criticism of his opinion based on such a lack of facts as is Mr. Harrington's.

I think that Coroner March deserves a great amount of praise for the good sense displayed in arriving at his verdict. Acknowledging his own lack of experience along the lines necessary to reach an intelligent verdict, he accepted the opinions of persons expert in the investigation of such accidents, rather than endeavor to give snap judgment on the testimony offered.

S. F. Jeter.

New Haven, Conn.

Advantages of License Laws

Yes, it is much cheaper to pay top wages to a first-class mechanic who can and will keep the wheels moving, than to intrust to a poor one several thousands of dollars worth of equipment simply to effect a saving of two or more dollars per day. The expense of one preventable shutdown might easily offset the difference in wages for an entire year.

In a no-license State how shall the employer decide upon the merit of an applicant who is unknown to him unless he judge by letters of recommendation which too often are worthless, as I have found by experience. If Mr. Incompetent can create a sufficiently favorable impression, he may be hired if a man is needed. Many engineers of poor ability possess a kind of cleverness which may appear like real wisdom to a prospective employer who happens to know nothing of the specific requirements of the position. Since a high salary is not always then a guaranty of worth, bearing in mind that the employer's point of view is that of the uninformed, is it surprising that the small owner prefers to hire "Mike," who is "handy" and who can be had for two dollars per day? He will say:

"Mike is handy; if anything breaks he can always patch it up somehow and manage to get along." But he spoils this in the next breath, when he says:

"Mike is always busy—he can always find something to do." This "being busy" is a point that wins much favor with this class of employer. He does not know that the engineer who has time to seat himself and talk to a visitor or to show him about the plant may nevertheless have his senses on the alert; that a proper degree of watchfulness and foresight will suggest the use of hammer and wrench *before* trouble grows serious enough to cause a shutdown, thereby preventing nine-tenths of the so-called accidents that would befall "Old Mike" in like circumstances. The man in the overalls has nothing to lose—whatever may happen to the plant in his care, he can collect his wages—the owner is responsible, and in case of accident he has no redress.

As long as the law will permit incompetent men to remain in positions where ignorance may jeopardize the lives of others, there will be certain people willing to hire them.

Recently one or two engineers have expressed the opinion in Power that license laws would have but little beneficial influence. They neglected to give any reason in support of this view, however, and should like to call their attention to the fact that with the enforcement of a proper law, an immense number of operating misfits would at once be obliged to leave the field clear for those who by virtue of a superior intelligence and ambition already possess the necessary qualifications.

There doubtless are state license and inspection laws that are excellent. Nevertheless it would seem that a national law would be highly desirable on account of the uniformity that would result and for the reason that the United States Government seldom does anything by halves.

Not long since, a firebox boiler was to be replaced by a new and larger one and the operator was asked for his opinion of the return tubular type. He had no idea what "return tubular" meant, but he was anxious to favor any possible preference the owner might have and so, to avoid committing himself irrevocably, he ventured:

"Yes, I've seen them there; but somehow—I don't care much for them." When the new boiler and other new equipment was ready for operation, this man was retained to take charge of it on the ground that he should have a trial because he had never had any serious trouble with the old outfit.

This plant was merely a side issue to the owner's other business, and in common with many successful small capitalists, he fully believed the care of such to be a trivial thing. This idea was gained by occasional visits to the power house and conversation with those of a smaller mental caliber than himself. For instance, a sudden repair job confronted the engine runner. The "Old Man" being present, was told that the defective part must be sent to the shop. On the principle that "two heads are better than one," he suggested an expedient that had not occurred to his mechanic, and time and expense were saved. After a few such occurrences the owner began to think that the engineering profession as it affected him, should easily be comprehended by a two-dollar-per-day man, and in proof he would cite his own genius in solving an unfamiliar question. Thus he surrounded himself with a wall of egotism which seemed proof against all argument.

Take again the employer who believes, on principle, in having the best of everything—equipment, supplies and operatives, and is willing to pay the price; here he may be between the devil and the deep sea. Please state, you non-license men, how you in his place, would judge the fitness of applicants without running the risk of placing your plant in the hands of one worthless man after another.

Let us hire good men, but let the Government weed out the ineligibles for the safety of the public.

G. W. Johnson.

Escanaba, Mich.

Pumping Layout

I read with interest the discussion of a pumping layout in the September 6 issue. Truman D. Hayes is decidedly wrong in his statements regarding the Pohlé air lift. He states that the water pumped by the Pohlé system must be allowed time to free the air.

As the mingled air and water in the pipe rises to the surface, the pressure gradually reduces until at the pipe outlet it is only that due to the atmosphere, provided the discharge is open to the atmosphere. As the pressure decreases, all of the air separates from the water except that which is naturally held by the water in solution and no amount of standing would serve to liberate any of the water so retained.

Mr. Monthau's discussion, in the same issue, is more nearly correct.

William G. Vedder.

Fort Logan, Colo.

The Largest Boilers in Existence

Editorial Correspondence

> *At the Delray station of the Detroit Edison company it is planned to operate each 14,000-kilowatt turbine with steam supplied by two specially designed Stirling boilers. Two are already installed.*

Lately there have been some interestng developments in the line of large-caacity boiler units at the Delray station f the Detroit Edison Company. These levelopments have taken place in the new ddition to the original plant known as No. 2 station, which, when completed, will onsist of two 14,000-kilowatt Curtis urbo-units and six Stirling boilers of arge capacity; the plan being to have wo boilers serve one turbine on the unit ystem, with one boiler for each unit lways in reserve. There are two of these oilers already in operation, one equiped with Roney stokers and the other vith Taylor underfeed stokers; a descripion of the former, however, will apply ssentially to the latter.

The boiler, as shown in the sketch, is louble ended with the elements facing ne another and the stokers under each nd. There are three steam and two mud rums, each 27 feet 9 inches in length, he central one being 54 inches in diamter and the others 48 inches. They are nade of 1⅛-inch steel plates having a ensile strength of 60,000 pounds per quare inch and are connected by banks f curved tubes as in the regular Stirling ype. The tubes are of the standard size –4 inches in diameter—and if placed nd to end would reach a distance of bout 4½ miles.

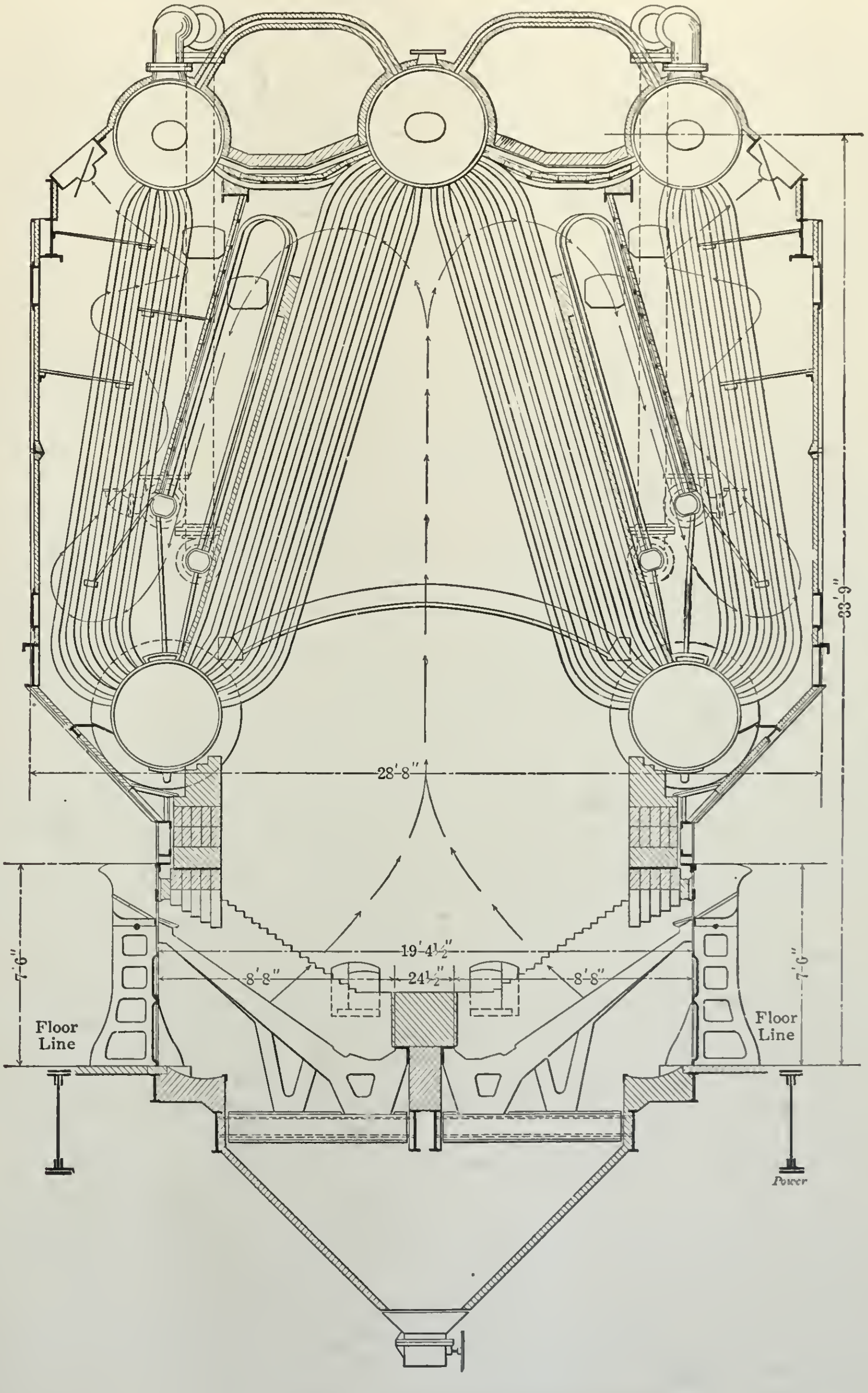
Steam is taken from the two side rums through 5-inch Hopkins-Ferranti traightway stop valves having contricted areas measuring 2½ inches in iameter. It then passes to standard Babock & Wilcox superheaters, one of which s installed in each end of the boiler, ence out to a Y-connection and into e line through a 10-inch Hopkins-Feranti motor-operated valve with a 5-inch utlet.

The boiler is equipped with four Roney tokers, each having a grate surface of 20 square feet, making a total of 480 quare feet with a ratio of heating to rate surface of 48 to 1. The normal orking pressure is 200 pounds with 175 225 degrees of superheat.

No official figures are available regardng the performance of this boiler, as ere has not been time enough to make areful tests. The largest load carried n the boiler to date is stated to be 9000 ilowatts on the main turbine in addion to about 300 kilowatts on the auxilries, making a total approximate load n the boiler of 9300 kilowatts or 12,400 orsepower. Allowing 14 pounds of steam er kilowatt-hour for the main unit and uxiliaries combined gives 130,000 ounds of water evaporated per hour at is load, which, divided by 23,040, repreents 5.64 pounds of water evaporated er square foot of heating surface.

When steaming at the maximum capacity, a kilowatt is generated on 2.48 square feet of heating surface. The circulation is toward the central drum, the water level being maintained much higher here than in the side drums. In fact, the latter act as steam drums, very little, if any, water remaining in them when under heavy load.

The feed water is introduced into the side drums, flowing downward into the second bank of tubes. A high efficiency is expected owing to minimizing the radiation and other losses per unit output.



Stirling Boiler Equipped with Roney Stokers

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Heat Transference

I have a feed-water heater arranged so that the steam and the water to be heated enter at the same end, that is, the hottest steam and the coldest water enter together. Now, if I should reverse the flow of the water, and have it enter the opposite end of the heater so it would leave the end where the steam is hottest, will it make any difference in its temperature?

W. H. R.

The transfer of heat from the steam to the water is proportional to the difference in temperatures, and where the hottest steam comes in contact with the hottest water the average temperature difference is greater than where the order is reversed and the coldest steam comes in contact with the hottest water. Therefore you would under ordinary conditions get hotter water with the contraflow.

Pressure of Superheated Steam

A boiler contains water under 200 pounds pressure and saturated steam at a temperature of approximately 387 degrees Fahrenheit. Superheat this steam 100 degrees, and the temperature would then be 487 degrees Fahrenheit. What would be its pressure?

H. D. Y.

Superheating does not change the pressure. By definition superheated steam has a temperature higher than that due to its pressure. If it is superheated 100 degrees it will have the original pressure but will have a temperature 100 degrees higher than that due to its pressure, whatever that may be. Steam at 200 pounds pressure superheated 100 degrees will have a temperature of approximately 487 degrees and still remain at 200 pounds pressure.

Spring Piston Rings

In getting new piston rings for a 13x12-inch high-speed automatic engine, what is the correct method with regards to taking the dimensions? Should rings be made thicker on one side than the other, or a uniform thickness all round? In taking out the old rings I find them to be $\frac{3}{8}$ inch on the bottom, tapering away to $\frac{1}{4}$ inch on the top where rings are split.

S. P. R.

Spring rings should be made $\frac{1}{4}$ inch larger than the cylinder for each foot in diameter and enough cut out so that the ring will just fit the cylinder without coming together at the split. There is a variety of opinion among engineers and designers as to whether spring piston rings should have uniform thickness or otherwise. It is really a matter of no importance as both kinds are used successfully. Unless protected by a keeper which will prevent the leaking of steam past the ring through the split, the split should always be kept at the bottom and not at the top of the cylinder.

Contents of Cylindrical Tanks

Please let me know how to figure the following: A horizontal tank 10 feet in diameter by 20 feet long contains 18 inches of liquid. How many gallons are there in it?

H. B. I.

A method of finding the amount of liquid in partially filled horizontal cylindrical tanks was given in the issue of August 9. The cross-sectional area of the liquid in the example given is 7.38 square feet, which, multiplied by 20, the length of the tank, gives 147.6 cubic feet of liquid. As there are 7.48 gallons in one cubic foot, there are

$$7.48 \times 147.6 = 1104.05$$

gallons of liquid in the tank.

Horsepower of Falling Water

A stream of water 4 feet wide and 18 inches deep runs over a dam and falls 14 feet. How much power can I get out of it?

M. G. F.

A stream of water 4 feet wide by 18 inches deep running over a dam would discharge 24 cubic feet per second; and the energy would be the weight of the water multiplied by the hight from which it falls,

$$62.3 \times 24 \times 14 = 20{,}932.8$$

foot-pounds per second. As 550 foot-pounds per second is the equivalent of a horsepower, the stream would deliver

$$20{,}932.8 \div 550 = 38$$

horsepower, 70 per cent. of which might possibly be realized, giving

$$38 \times 0.70 = 26.60$$

actual horsepower.

Required Pressure for Operating Pump

I wish to apply 200 pounds hydrostatic pressure per square inch to a boiler with a 6 and 4 by 6-inch steam pump. The suction is attached to the city water main in which there is a pressure of 35 pounds per square inch. What steam pressure will be required to operate the pump?

M. P. M.

The area of the 4-inch water piston is 12.56 square inches, on one side of which is the pressure of the water going to the boiler and on the other that of the city water. The difference between these pressures which must be overcome by the steam pressure against the steam piston is 165 pounds per square inch. The total pressure to be overcome is

$$165 \times 12.56 = 2072.4$$

pounds.

The 6-inch steam piston has an area of 28.27 square inches and to balance the pressure of 2072.4 pounds on a water piston, it will require

$$2072.4 \div 2827 = 73.3$$

pounds per square inch steam pressure.

How much higher the steam pressure will have to be to overcome the friction of the pump and the water will depend on the circumstances in the case and can only be determined by experiment. It is probable that 100 pounds steam pressure will do the work.

Keeping a Boiler Dry

Keeping a Boiler Dry

What means should be used to keep the inside of a boiler dry while out of service?

B. D. K.

Several methods are employed. Some engineers make use of the natural circulation of the air through the upper and lower manholes which are left open for this purpose. Others place pans filled with quicklime inside and close all openings.

Pressure Required to Turn Crank

With the same load on the engine, will the same steam pressure tend to turn the crank faster when the crank pin is on the upper quarter than when on the lower one?

M. C. R.

When pressure is applied to the crank pin the tendency to turn the shaft is the same whether the pin is above or below the center, and is proportional to the pressure applied, and there is no difference in the power delivered.

Power and The Engineer

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for September, 1910, *was* 134,000.

October 4.............. 33,000
October 11............. 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Editorial

Pay Firemen on B.t.u. Basis

Welfare work is being undertaken by great corporations on an extensive scale with the realization that any outlay expended in this direction will be well spent in quality and quantity of work returned by those who benefit by the system instituted. This idea is recognized in the power plant as well as in the various branches of industry. There has been scarcely a plant erected in the last few years in which some provision has not been made for the comfort of the operatives. Anything which tends to create a personal interest in the work is of value in increasing the efficiency of an employee. There is a great difference between the worker doing merely what he is paid to do and the interested employee attending to those endless and countless little details for the welfare of the business.

In the case of the fireman this is especially true. By many managers he is considered the most important factor in the operation of the steam plant. He shovels coal and coal means dollars, which, if saved, will mean increased dividends to the company.

In this connection, therefore, it is of special interest to notice the results of a bonus system for firemen employed in a certain plant, as brought out in the late meeting of the Central States Water Works Association. Here, on a daily fuel consumption of from twelve to sixteen tons, there was a net saving, after paying all bonuses to firemen, of from six to eight dollars a day, merely through a system which created an interest in the work being done.

If it is considered good practice to buy coal on the B.t.u. basis, why should it not be equally advantageous to pay a fireman according to the number of B.t.u. that he is able to transfer into useful work? The fireman is given a certain amount of coal containing a known number of heat units, and the measure of his efficiency and consequently his rate of pay should be the number of heat units he is able to transfer to the water in the boiler. With such a system in vogue, it can be taken for granted that the results will be as favorable as the fireman can make them with the class of installation and apparatus with which he has to deal—and he will have a more intelligent interest in and appreciation of that apparatus.

Boiler Feeding by Centrifugal Pumps

In Europe the practice of feeding steam boilers by means of centrifugal pumps, either turbine- or motor-driven, is quite common, but in this country it is the exception. Just why designers of power plants on this side of the water should lag in this respect is not clear, for there does not seem to be any good reason why boiler feeding by means of the centrifugal pump should not become common practice.

Perhaps the claim that a small steam turbine will consume more steam than a steam engine of the same output may have something to do in influencing engineers against taking up this system of boiler feeding, and not knowing exactly what the performance will be they hesitate to take a step into the dark. A noncondensing steam turbine uses less steam by far than a reciprocating steam pump.

Excessive steam consumption in boiler-feeding apparatus in condensing plants need not be considered as a serious drawback, because the steam passing through such units is frequently not enough to heat the feed water to a proper temperature. In winter, if the plant furnishes exhaust steam for heating purposes, and many of them do, the auxiliary exhaust steam is often insufficient and live steam is used from necessity.

It will be conceded by all that the turbine-driven centrifugal pump occupies but little space and requires scarcely any floor room and foundation other than enough to bring it to a convenient level above the floor. The advantages of such a unit where the available room is restricted is quite apparent.

For convenience of operation the turbine-driven centrifugal pump has no equal. There is no noise in its operation and it requires practically no attention after being started. The first cost is greater than any other type of steam-driven pump, but the upkeep should amount to very little and the bother of renewing packing is practically eliminated.

Cleanliness is another feature favoring this type of boiler feeder. Being compact it is easily kept clean, and to avoid leakage of water on the pump-room floor the drippings from the stuffing box may be easily caught and piped to a suitable drain.

The ordinary boiler feeder is never run at a constant speed. In small plants the water tender is almost constantly turning the throttle valve of the pump either open or shut. If a feed-water regulator is used, as is common in large plants, the speed of the pump is automatically controlled, but when the speed of the pump is reduced the amount of steam entering the feed-water heater is also cut down, and the water enters the boiler at a lower temperature.

In the case of a centrifugal boiler feeder the speed remains constant regardless as to whether one boiler is being fed or a dozen. The steam consumption may vary somewhat, but under ordinary conditions it will hold close to an average, and if a change does take place it will be so graded as to have no sudden effect upon the temperature of the feed water entering the boiler.

The pressure on the water main is constant (if the pump runs at constant speed) and cannot be increased by shutting off feed valves. Even if a stop valve on the main were closed the pressure in it would not go up, as the pump impeller would simply churn the water in the case. There is, therefore, no danger to the heater tubes, etc., nor to the piping. The flow of water is free and continuous and there is no clicking and slamming of check valves.

It is possible that the turbine-driven centrifugal pump for boiler feeding may have objectionable features with which the majority of engineers are not familiar, but if there are good reasons why such a pump cannot be used to advantage in feeding boilers, the readers of POWER should know about them. So bring out your paper and pencil and get busy.

Boilers of Large Capacity

In engineering circles it is now being recognized that the greatest economy is attained by working the heating surfaces to their limit, and that there is less loss with an intense fire and rapid circulation of the gases than with a large, slow fire, allowing gases to escape up the stack unburned. This practice of forcing boilers has been brought about by the demand for greater power. In many of our larger plants the distribution and class of service have been so carefully thought out that an almost constant load factor is attained. Where such conditions exist it is well to concentrate the load on as few boilers as possible and force them, thus minimizing the losses.

Unfortunately with the increasing demand for power and the consequent introduction of large turbine units, boiler practice has not kept pace as far as the capacity of the individual unit is concerned. But now comes the announcement of a bold step, page 1831, in this direction at the Delray station of the Detroit Edison Company. Here it is planned to have each fourteen thousand-kilowatt turbine served by two Stirling boilers of a size heretofore never attempted. Two of these boilers have been erected and put into service; and from a reliable souce we learn that one of these has developed as high as twelve thousand four hundred horsepower at the engine, although nominally rated at only about twenty-three hundred boiler horsepower. The public awaits with keen anticipation the publication of the official test figures showing the quality of steam produced, the economic evaporation and the coal burned per square foot of grate surface. The success or failure of this installation will have a potent influence upon future boiler practice.

Raising the Standard

Not the least encouraging feature of the evidence that the Institute of Operating Engineers will be successful in its aims is the attitude of the employers of operating skill toward the organization wherever they have been approached. It is beginning to be realized that the engineer holds an important position in production and transportation, as well as in all allied industries, and with this realization comes also the knowledge that the men who operate engines and power machinery must possess both skill and character.

As training in the vocation and cultivating desirable traits of character with habits of industry and sobriety is one of the principles upon which the organization is founded, it meets with a hearty welcome. It will be a powerful element in the "uplift" work, which in almost numberless forms is going on everywhere, for it will not only evolve better men but will force better working conditions just as fast as it is seen that only the best of conditions will attract the best men.

Besides the technical and mechanical qualifications some general culture will be expected from all who become members. It frequently happens that the employee who can, when called upon, give intelligent and concise replies to questions sometimes only remotely related to his work, is the one who is selected for a particular duty requiring tact and general information as much as the purely technical knowledge of the work. It is one of the aims of the Institute to aid men to acquire a more general education.

It is related that a peasant's excuse or reason for keeping the pig in the house was because the house had all the conveniences that a pig needed. In the growing industrial system employers of skill will not furnish houses for pigs, nor will pigs have any place in the equipment.

It is the aim of the Institute to evolve a class of workers to whom the employer will offer only the best conditions obtainable. There may always be inferior and ignorant workers and there may also always be employers who value a worker in direct proportion to the grime and gurry on hands, face and raiment, but the Institute aims to eliminate both. One class will not be in the membership and the other will not be numbered with the patrons. It is an attempt to give the calling of engine and machinery operation and management the place in modern industry which so manifestly belongs to it, but, which, through several causes, has not come into its own.

Carroll Balks at an Examination of His Carbonic Acid Gas Engine

We announced in our issue of July 12 that John Carroll, of carbonic-acid gas-engine fame, had been arrested by the United States Post Office authorities at Boston, on a charge of fraudulent use of the mails. It was decided to have his invention tested by experts appointed by the Government, in order to determine if there were any merit in it, or if it were a fraud. Arrangements were made by the Post Office department at Boston, in conference with Mr. Carroll, to have Professors Berry, Riley and Rowe, the first two from the Massachusetts Institute of Technology, and the latter from Harvard, examine the engine on May 27. When the time arrived, Mr. Carroll claimed to be sick and refused to allow an examination to be made. It was then decided to postpone the examination to a date to be named later, but Mr. Carroll later refused to allow an examination to be made at all, and was then arrested. He was forced by certain of his backers to ask for an examination, and accordingly, on July 21, E. G. Bailey, M. E., of the Fuel Testing Company, of Boston, Professors Berry and Taft, of the Massachusetts Institute of Technology, and Professor Rowe, of Harvard, made an examination of the engine.

The holes in the regenerating tank (evidently those referred to in our description of the engine on page 937 of the May 24 issue) appeared to be plugged up; the engine would not run. Carroll asked for time to repair it. This was granted and another test arranged for August 6. At this test Carroll had secured but a small amount of carbonic-acid gas and a test could not be made. Another date was set for August 16, at which time the holes spoken of were found to be open. The engine ran with a pressure apparently equal in both tanks, but it was apparent that he was exhausting from the charging tank and the outer tank of the regenerating tank into an inner tank and thence through the holes into the air. Soon after this Carroll was surrendered by his bondsmen and has since been in jail. On Tuesday, September 27, he was indicted by the Federal Grand Jury. He is also still under indictment in Philadelphia, where the original exploitation, exposed by POWER in the fall of 1907, was carried on.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Rotary House Pump

A new type of house-service pump, shown in Fig. 1, has recently been introduced into this country from Germany, where it has been operating successfully for several years. The American rights, including those for the controlling apparatus, have been secured by the Rotary House Pump Company, 103 Park avenue, New York City.

This system of water supply consists of a Pittler pump direct coupled to an electric motor and a bedplate on which the pump and motor are mounted. Air is compressed in the top of the air chamber to a predetermined pressure, when it actuates a small diaphragm throwing out an automatic switch and cutting out the motor. When the pressure falls in the air chamber, due to water being drawn from the system, the reverse action occurs, the diaphragm being actuated by a spring. The system is automatic, the motor driving the pump only when water is being drawn from the system. The water is never lifted higher than the faucet or outlet being used.

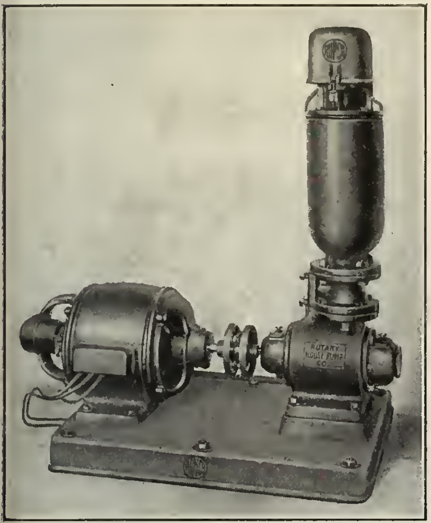


FIG. 1. ROTARY HOUSE PUMP COUPLED TO MOTOR

The pump consists of a rotating bronze cylinder secured to a shaft and revolving between two end cams having parallel curved faces. The cylinder has rectangular slots in its surfaces which are machined parallel to the axis and contain sliding vanes. The whole is inclosed in a casing containing ports connecting the pumping chambers with outlet and inlet pipes. The two end faces of the rotating cylinder are plain surfaces, perpendicular to the shaft. The end cams have special surfaces, which form a series of working chambers around the shaft at either end of the rotating cylinder, and the ends of the vanes enter these working chambers by reason of the cam surface at the opposite end. There is a port in the casing over the first part of each working chamber connected to the inlet.

Fig. 2 shows a diagrammatic representation of the pump projected on a plane. The vanes sliding between the two parallel and curved guiding pieces pass once the suction opening and once the discharge opening in the casing with each full rotation of the shaft. In position *I*, at the left, the upper edge of the sliding vane is just entering the suction chamber which is open to the suction port. When moving toward the right from *I* to *II* under the suction port the vane is surrounded by inflowing water with a balance of pressure on either side, the vane sliding endwise out of the rotating cylinder. At the moment this sliding vane reaches the end of the suction chamber (position *II*) the preceding vane passes the beginning of the discharge port, so that the first vane in moving toward the right forces the water in front of it into the discharge port. In moving from position *II* to *III* the vane forces the water ahead of it by its piston action to the discharge port, at the same time sucking water with the opposite side. During this period the vane does not slide in the rotating cylinder so there is but slight loss by friction. From position *III* to *IV* the vane slides under the outlet with balanced pressures on either side, the sliding of the vane against the curve taking place without any great resistance. From the position *IV* to *I* the vane is at rest, remaining entirely in the rotating cylinder, its purpose being to tighten the suction chamber against the pressure chamber.

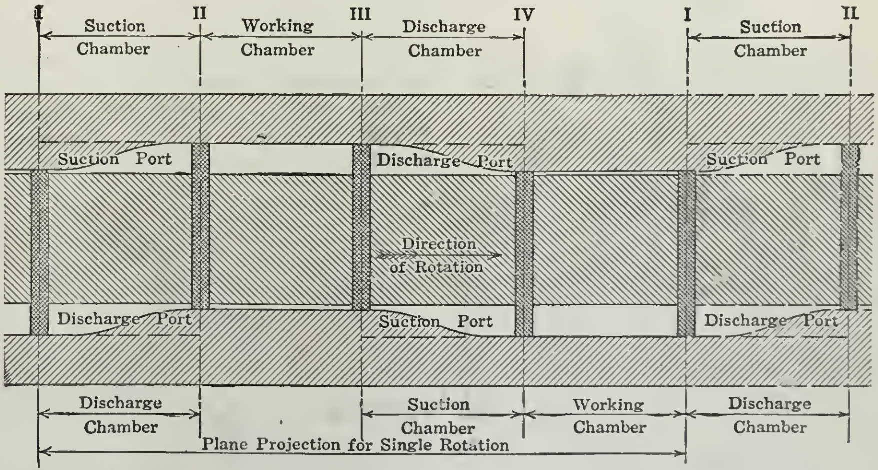


FIG. 2. LONGITUDINAL VIEW OF ROTOR AND CYLINDER OF HOUSE ROTARY PUMP

On the other side of the drum the same action takes place, only at a difference of 180 degrees. Each vane with each full rotation of the drum therefore sucks and forces a quantity of water which has twice the volume of the space inclosed between the two sliding vanes in positions *II* and *III*. At the end of the suction and forcing periods of one sliding vane, this one is immediately replaced by the full side of another one, and therefore an absolutely equal quantity of water is being constantly pumped or drawn in for every position of the shaft if it is rotating uniformly.

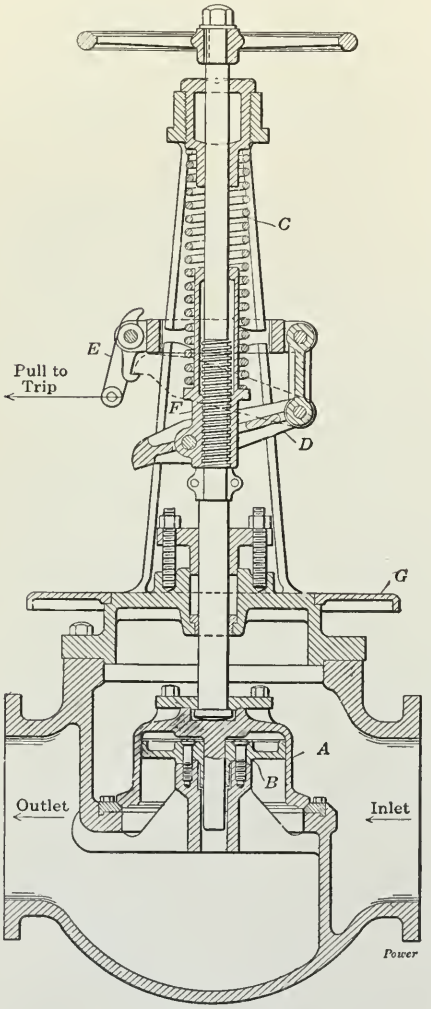
Nearly all water contains grit or solid matters of some kind, which causes a certain amount of wear on the ends of the vanes butting against the cam heads. For this reason the pump is constructed with vanes divided through the center. Pressure is introduced between the two halves by leading a small channel from the discharge to a groove, which entirely surrounds the rotating cylinder. By this means, the vanes are kept always in contact with the cams at either end and the wear on the ends of the vanes is automatically taken up, maintaining the efficiency of the pump. The rotating cylinder being of bronze, and the vanes of suitable material, the sticking or rusting together of the moving parts is entirely eliminated, it is claimed. The balancing effect of the pressures on the sliding vanes reduces ordinary wear to a minimum. With this split-vane construction, the pump maintains its volumeter efficiency even when pumping mud and sand, it is said.

This pump is applicable to all classes of pumping work, combining the advantages of rotary action, as with turbines or centrifugal pumps, with a positive delivery.

The only lubrication necessary is provided by ring oilers on the shaft, which are fed by the oil reservoirs at either end of the pump.

Low Pressure Turbine Trip and Throttle Valve

This valve is used as a combined throttle and trip valve for low-pressure turbines. The essential features are a means of throttling by a handwheel and screw spindle combined with the automatic tripping and closing of the valve by the turbine governor.



Trip Throttle Valve

The valve has a solid bell-shaped disk *A*, which forms a dashpot by moving over a fixed piston *B*. This prevents the disk from ramming the seat when it is tripped. The closing medium is a spring *C* in the yoke, which is sufficiently strong to close the valve when it is tripped and overcome the friction and light internal pressure on the spindle.

The pressure always enters the body of the valve above the seat in the direction of the arrow. When the valve is closed and the lever *D* is engaged in the hook latch *E*, the valve may be operated as a stop or throttle valve by the handwheel. The hook latch *E* is connected by a rod to the automatic tripping arrangement of the turbine governor and when this hook latch is tripped, the screw trunnion *E*, which is carried by the lever *D*, is forced down by the spring, carrying with it the screw spindle of the handwheel and disk, thereby closing the valve. To reset the trip hook, the handwheel is turned in the direction of closing or "clockwise;" this raises the trunnion and lever and compresses the spring without disturbing the disk of the valve on its seat. After the lever is engaged in the trip latch, the valve may be opened or closed by the handwheel as at the start.

The plate *G* is intended as a finish to the asbestos or other material with which the body may be covered.

This valve is manufactured by the Schütte & Koerting Company, Tompson and Twelfth streets, Philadelphia, Penn.

New High Duty Self Cleaning Stoker

The Under-Feed Stoker Company of America, Marquette building, Chicago, Ill., is now making a new self-cleaning stoker designed to be used in plants where the greatest possible horsepower is desired in the least possible space. This stoker is not intended to replace the standard Jones stoker, but has been designed to meet the increasing demands for a greater fuel-burning capacity.

The operation of the new stoker is the same as that of the standard design, the proportionment and regulation of fuel and air supply by means of steam pressure being retained. The parts of the new stoker outside of the furnace proper are counterparts of the standard stoker. Inside of the furnace the essential points of difference may be noted as follows:

The depth of the retort at its forward end is over double the depth of the retort used with the standard stoker, the depth gradually decreasing toward the rear until at its extreme rear the depth is practically the same.

The self-cleaning mechanism is operated by an auxiliary cylinder, which is located under the floor line. The location of this cylinder, however, is not fixed

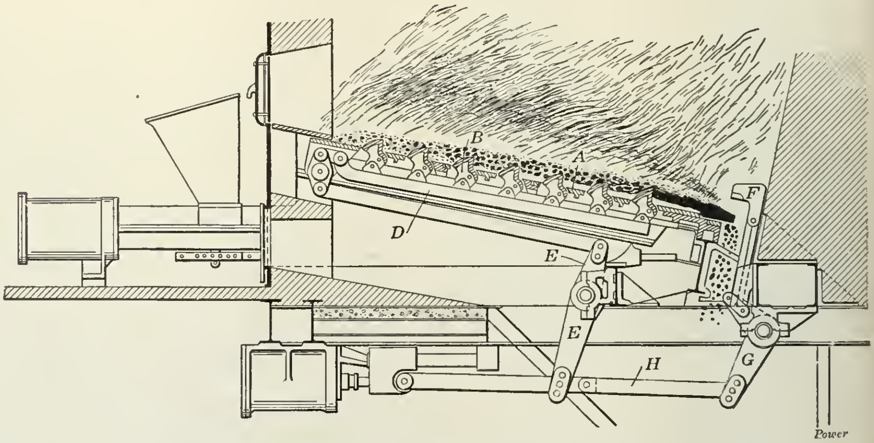


Fig. 1. Side View of Self Cleaning Stoker

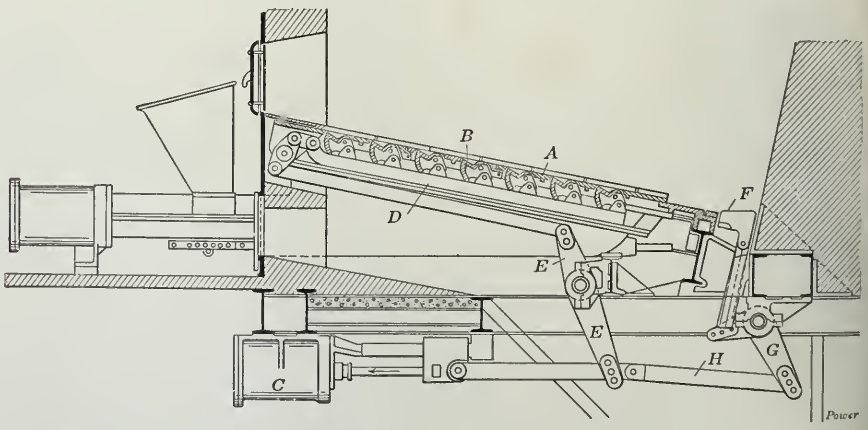


Fig. 2. Showing Dead Plates at Rest

and it may be located on the floor line at any accessible point.

Sectional dead plates are used, each section locking into its mate. These dead plates have a longitudinal movement varying from 2 to 6 inches or more, as desired. They contain rectangular openings for receiving the dogs *B*, Figs. 1 and 2, which are located at intervals of approximately 9 inches lengthwise of the plates.

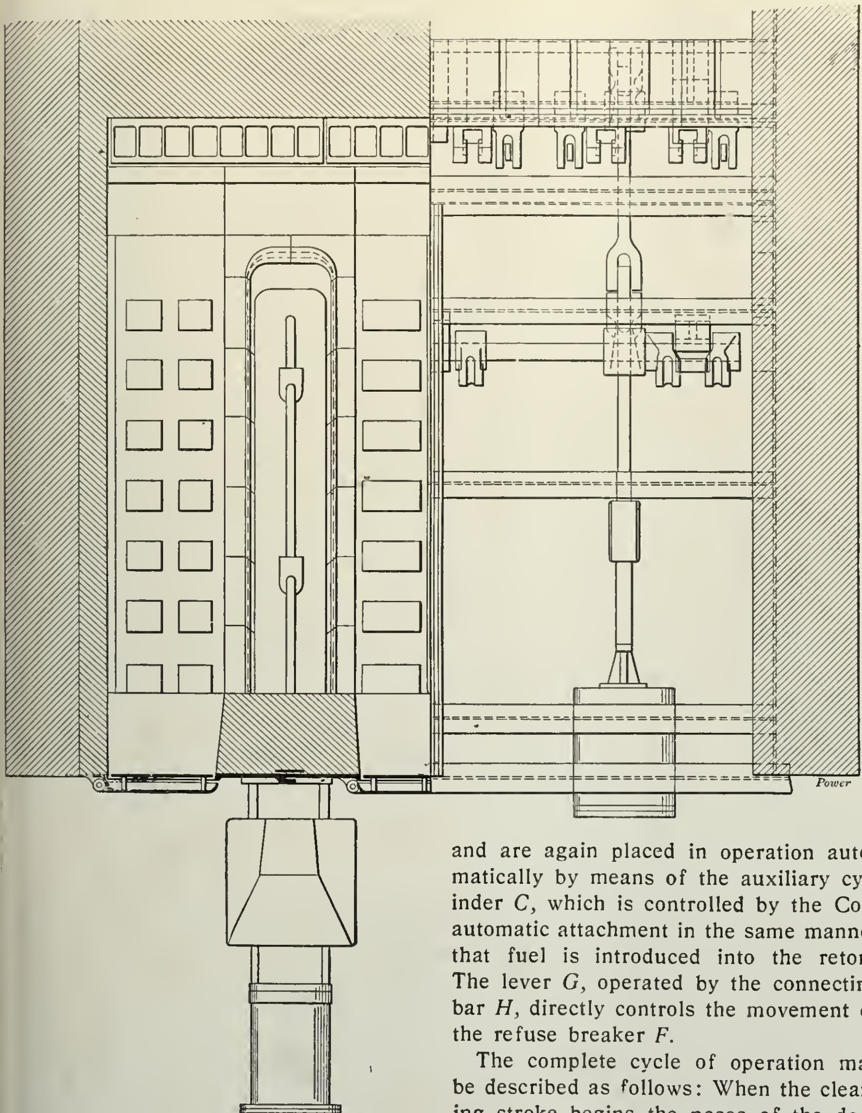


Fig. 3. Plan View of Self-cleaning Stoker

Beneath each dead plate and actuated y the auxiliary cylinder *C* there is a bar) connecting the lower portion of the logs together to effect a movement of all he dogs in the dead plates in unison with he movement of the dead plates themelves. Each connecting bar *D* is moved y the lever *E*. The dogs *B* are for the urpose of giving an upward and a forard movement to the refuse which acumulates on the dead plates, thereby assisting the forward motion of the clinker t the time when the dead plates are noving longitudinally forward.

The refuse breaker *F*, at the rear of the stoker proper, is operated in exact unison with the dogs and the dead plates. The dogs, the dead plates and the refuse breaker begin operating at about the same time, the movement of the refuse breaker first being upward, so that the clinker which is farthest downward will extend beyond the end of the retort and dead plates and under the refuse breaker *F*. The parts above referred to having completed their upward and forward movement, return to the starting point and are again placed in operation automatically by means of the auxiliary cylinder *C*, which is controlled by the Cole automatic attachment in the same manner that fuel is introduced into the retort. The lever *G*, operated by the connecting bar *H*, directly controls the movement of the refuse breaker *F*.

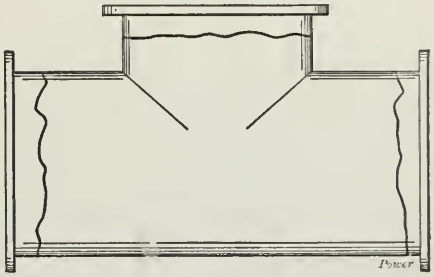
The complete cycle of operation may be described as follows: When the cleaning stroke begins the noses of the dogs rise above the dead plates about 1½ inches, facing the bridgewall. The dead plates themselves do not move until the dogs have risen to their extreme stroke, when the plates and dogs referred to move forward to the end of their travel, and carry the refuse a predetermined distance. The upward movement of the refuse breaker having been completed when the dogs and dead plates are at their forward position, then begins its downward stroke, carrying with it to the ashpit such refuse as may not have fallen therein, the dead plates and dogs returning to their original position. These movements are all accomplished by one forward and one backward movement of the auxiliary-cylinder piston. Fig. 3 shows a plan view of the stoker.

The increased fuel capacity of the self-cleaning stoker is not due to the self-cleaning attachment, but is due to the fact that the retort is materially longer than the standard retort; the tuyere area is, therefore, greater and in addition the tuyere blocks are made of greater hight, therefore containing greater air outlet. In other words, the tuyere area of the self-cleaning stoker, due to its greater length and greater air-outlet space in the tuyere blocks, has been increased fully 200 per cent., it is claimed.

The builders also state that with two of these self-cleaning stokers a smokeless consumption of 7000 pounds of bituminous coal per hour has been obtained in a water-leg furnace 5 feet 6 inches wide by 8 feet long, this consumption being at the rate of 160 pounds of coal per square foot of furnace surface per hour.

Receiver on Large Cross-Compound Explodes

On September 16, at 5:20 a.m., at the Harvard station of the Boston Elevated Railroad Company, in Cambridge, Mass., the attendant had completed the preparations for starting a 2700-horsepower vertical cross-compound condensing engine and was standing at the throttle when the receiver between the high- and the low-pressure cylinders exploded. The concussion blew out all the windows at one end of the building. The receiver dropped, carrying with it a considerable part of the valve gear, and crashed through the concrete floor, landing in the basement.



Location of Breaks in Receiver

The receiver was made of two cast-iron cylinders, 8 feet long, 3½ feet in diameter and about 2 inches in thickness; only one of these was damaged. It was equipped with a spring-loaded safety valve. It failed in three places, the middle flange casting and about 1 or 2 inches from the end flanges.

From observations and the little that could be drawn from the attendants, it is thought that the receiver failed from too high pressure, caused either by warming the cylinders preparatory to starting, or possibly from a leak in the reheater coil and the failure of the safety valve to operate. It is suggestive that one of the attendants said that more than one safety valve would be placed on the new receiver that is being installed.

Safe Working Rule for Belts

In *Phoenix*, a publication issued by the New York Leather Belting Company, Samuel Webber, of Manchester, N. H., gives some interesting data on the transmission power of belts which is worthy of repetition.

"I have had a working rule for many years, which was given to me by an old and experienced machinist, and may be thus expressed: Ordinary leather belting 1 inch wide, having a velocity of 600 feet per minute, will transmit one horsepower.

"Practice has shown me the safety of this rule, for which I have sought a reason and a formula.

"After an examination of the textbooks, I found that Morin's data gave me the clue to the truth of this rule, and that it was supported by other good authority. Morin says: 'Belts designed for continuous service may be made to bear a tension of 0.551 pound per 0.00155 square inch of section, which enables us to determine their breadth according to the thickness.' This is equal to 355 pounds per square inch of belting leather, and it is also equal to from ⅛ to 1/12 the breaking strength of the same, as given by Rankine and other authorities.

"From this I see my way to a simple formula: Substituting 330 pounds for 355 pounds per square inch, I strike the component part of a horsepower and deduce the following: One square inch of belting, at a velocity of 100 feet per minute, will transmit one horsepower with safety, and from these data get this rule: *"The denominator of the fraction expressing thickness of the belt in inches gives the velocity in hundreds of feet per minute at which each inch of width will transmit one horsepower.*

"Thus

⅙ = 6 × 100 = 600 feet; ⅓ = 3 × 100 = 300

feet, and so on.

"Now, ⅙ inch being about the ordinary thickness of a single belt, this shows me why my 'old rule of thumb' proved right, and a careful examination of many of the large belts running in our New England cotton mills within the last year or two confirms my opinion as to the safety of the rule.

"This gives a strain of 55 pounds per inch, and a belt speed of 50 square feet of surface per minute per horsepower as safe, ordinary practice for single belts; and I find the same velocity, with a strain proportioned to the thickness, works well with double belts. This, however, is applicable where there is a sufficient holding surface on the smaller pulley, if the arc of contact be small, a wider belt will be necessary, and I am not yet able to formulate a rule for belt contact. The nearest approach that I have made to it yet is to allow 10 to 12 square inches of pulley surface in contact with belt for each horsepower.

"It is generally conceded that the friction of a belt passing half around a pulley, is equal to one-half the strain on the belt; or that an inch belt, at 600 feet per minute, with a strain of 55 pounds would give a traction of 27.5 pounds and require a pulley which would give 1200 lineal feet per minute of surface contact, to obtain the horsepower to which the belt would be equal.

"The friction of a belt varies with the arc circle of the pulley with which it is in contact, and is only half as great on one-quarter of a pulley as on one-half of one; so that double the surface in square inches will be needed in the latter.

"Carrying out these rules, it will easily be seen that where high speed is to be obtained by the use of small pulleys, a much greater width of belt is necessary to get the frictional surface than is called for by the strength of the leather."

National Conservation Congress

At the second national conservation congress, held in St. Paul, Minn., September 5-9, 1910, President William H. Taft opened the congress and ex-President Theodore Roosevelt in his speech on the following day emphatically urged that the Federal Government retain control of water-power sites as well as coal lands and forests. Other speakers, among them James J. Hill, urged that these resources be left to the control of the individual States, believing the national machine to be too large and too remote and its operation too slow, cumbrous and costly.

The congress concluded its deliberations by adopting a platform in which it declared the rights of the people "to be natural and inherent and justly inalienable and indefeasible" and that these resources must be "developed, used and conserved in ways consistent both with current welfare and the perpetuity of our people." It further declared that the waterways should be administered in the interests of all the people, and that in doubtful cases there should be coöperation between State and Federal agencies; that use of the waters should be made with reference to all other uses and care be taken to prevent defilement; that the policy with regard to swamp lands be continued and extended; that a Federal commission empowered to deal with all water powers be maintained; that public lands be withdrawn pending classification and surface rights be separated from mineral, forest and water rights; that mineral lands be leased for a limited period; and that national forests be controlled by the Federal Government and the forestry service be given larger appropriations. The congress further expressed itself as opposed to child labor as detrimental to the life and development of the people physically, intellectually and morally, and recommended legislation suitable to this end.

A. S. M. E. Meetings in New York and Boston

The first meeting of the American Society of Mechanical Engineers for the coming season will be held in the Engineering Societies' building, New York, on Tuesday evening, October 11. The paper will be by Frank B. Gilbreth on "Fires: Effects on Building Material and Permanent Elimination." This paper was published in *The Journal* for May.

Meetings of the society in Boston are announced for October 19, November 11 and December 23. The October meeting will be conducted by the Boston Society of Civil Engineers, the American Society of Mechanical Engineers and other engineering societies of the city coöperating. The subject will be "An Account of the Destruction of Carthego, Costa Rica, by the Earthquake of May 4, 1910."

The November meeting will be conducted by the American Society of Mechanical Engineers and the December meeting by the Boston section of the American Institute of Electrical Engineers. It is expected that the meeting in January will take the form of a dinner of the character of the one held last January, at which there were some five hundred in attendance, representing the various engineering and architectural societies of Boston and vicinity and many of the technical schools and other organizations.

The annual meeting of the society will be held in New York, December 6-9, 1910.

Board of Education Public Lectures

In the new season of free public lectures for adults which was opened by the Board of Education, of New York City, on Saturday evening, October 1, special emphasis has been laid upon technical subjects, such as will appeal to workmen and to students in the technical trades. Of first importance is the course on "Power" to be delivered by Dr. Charles E. Lucke, professor of mechanical engineering at Columbia, who will give eight lectures on the topic on Saturdays at St. Bartholomew's hall, 205 East Forty-second street, beginning on October 1. His first lecture will consider "Mechanical Power and Machinery."

On October 4, at Public School 63, Fourth street, east of First avenue, R. J. Newton Gray, of the Manual Training High School, began a course of eleven lectures on "Physics," while on October 6, at St. Luke's hall, Hudson and Grove streets, Dr. Frederick E. Breithut, of the College of the City of New York, opened a course of eleven lectures on "The Chemistry of Common Things." On October 10, at the Stuyvesant High

School, Sixteenth street, near First avenue, Dr. William L. Estabrooke will give the first of his course on "Inorganic Chemistry."

During November, Prof. Bradley Stoughton, of Columbia, will deliver a course on "The Metallurgy of Iron and Steel" at Cooper Union and also at Institute Hall, 218 East One Hundred and Sixth street. While in December, Prof. Charles E. Pellew will give a course of lectures on "Dyeing" at St. Luke's hall.

In the Bronx, at Lafayette hall, Alexander avenue and One Hundred and Thirty-seventh street, on October 3, W. Wallace Ker, of the Hebrew Technical Institute, opened a practical course on "Principles and Practice in Electrical Engineering."

PERSONAL

Announcement is made of an important change in the management of the Goldschmidt Thermit Company, 90 West street, New York. On October 1, E. Stutz, vice-president and general manager, retired from the direction of the company, which passed under the management of William C. Cuntz. Mr. Cuntz brings to his position a thorough knowledge of the steel business and a wide acquaintance with the railway and street-railway officials of the country, having been connected for eighteen years with the Pennsylvania Steel Company.

BOOKS RECEIVED

KENT'S MECHANICAL ENGINEERS' POCKET-BOOK. Revised. By William Kent. John Wiley & Sons, New York. Morocco leather; 1461 pages, 6¾x1¼ inches; 218 illustrations. Price, $5.

HE CONSTRUCTION OF GRAPHICAL CHARTS. By John B. Peddle. McGraw-Hill Book Company, New York. Cloth; 109 pages, 6¼x9½ inches; 58 illustrations. Price, $1.50.

DESIGN OF MARINE MULTITUBULAR BOILERS. By James D. McKnight and Alfred W. Brown. The Technical Publishing Company, Ltd., London, England. Cloth, 48 pages; 6⅜x10 inches; illustrated. Price, $1.50.

HEAT ENGINES. By John R. Allen and Joseph A. Bursley. McGraw-Hill Company, New York. Cloth; 288 pages, 6¼x9½ inches; 157 illustrations. Price, $3.

NEW INVENTIONS

Printed copies of patents are furnished by he Patent Office at 5c. each. Address the ommissioner of Patents, Washington, D. C.

PRIME MOVERS

ROTARY ENGINE. William M. Hoffman, uffalo, N. Y., assignor to Frank B. Allen, oronto, Canada. 971,043.

ELASTIC FLUID TURBINE. Carl Richrd Waller, Trenton, N. J., assignor to De aval Steam Turbine Company, New York, . Y., a Corporation of New Jersey. 971,092.

STEAM ENGINE. Georje Psareas and Nick Parkas, Holyoke, Mass. 971,402.

TIDE MOTOR. Daniel M. Barr, Philadelphia, Penn. 971,343.

BOILERS, FURNACES AND GAS PRODUCERS

GAS PRODUCER. William B. Hughes, Cleveland, Ohio. 971,271.

GAS PRODUCER. William B. Hughes, Cleveland, Ohio. 971,272.

BOILER. Ferdinand W. Meyer, Kansas City, Mo. 971,296.

FURNACE. Peter Stipp, Scranton, Penn. 971,416.

POWER PLANT AUXILIARIES AND APPLIANCES

FLUID PRESSURE REGULATOR. Max Jaspersen, Hamburg, Germany. 971,275.

CUTOFF VALVE. Morgan D. Menear and Charles O. Shay, Independence, W. Va. 971,295.

CARBURETER. Edward J. Gulick, Mishawaka, Ind., assignor to the Simplex Motor Car Company, Mishawaka, Ind., a Corporation of Indiana. 971,038.

CONTROLLING VALVE FOR FLUID OPERATED MOTORS. Charles Algernon Parsons, George Gerald Stoney, Alfred Quintin Carnegie and Albert William Bowerbank, Newcastle-upon-Tyne, England; said Stoney, Carnegie and Bowerbank assignors to said Parsons. 971,069.

ROTARY PUMP OR AIR COMPRESSOR. George R. Gether, Milwaukee, Wis. 971,188.

FURNACE DOOR. Gustav de Grahl, Zehlendorf, near Berlin, Germany 971,266.

STEAM AND WATER GAGE. Edward Halley, Chicago, Ill., assignor of one-half to R. J. McDonell, Victoria, British Columbia, Canada. 971,268.

PISTON-ROD PACKING. Frank N. Mason, Pittsburg, Penn. 971,292.

COOLING DEVICE FOR INTERNAL COMBUSTION ENGINES. Ralph Brooks Vaughn, Wilkes-Barre, Penn. 971,328.

BOILER FEED-WATER REGULATOR. Howard E. Cade, Pencoyd, and David R. Knapp, Philadelphia, Penn. 971,357.

DEVICE FOR FACILITATING THE REMOVAL OF ENGINE VALVES. Oliver A. Hays, Southampton, N. Y. 971,375.

STEAM-PRESSURE REDUCING AND CONTROLLING VALVE. Nicolas Filiberto, Brooklyn, N. Y. 971,435.

ENGINE GOVERNOR. Charles B. Edwards, Massillon, Ohio, assignor of one-fourth to John H. Crooks and one-fourth to William Simonet, Massillon, Ohio. 971,431.

THERMOSTATIC OIL CUP. Edwin J. Grant, Sandy Point, Me. 971,440.

STOP VALVE. Charles O. Hawkins, Chesterhill, Ohio. 971,446.

VALVE MECHANISM FOR COMPOUND ENGINES. Paul Oluff Poulson, Brigham, Utah. 971,472.

STARTER FOR INTERNAL COMBUSTION ENGINES. David Smith Anthony, Durango, Mexico, assignor of one-third to Edward Williams, and one-third to George F. Habernigg, Durango, Mexico. 971,482.

STEAM CONDENSER. Henry L. Brandon, Leavenworth, Kas. 971,486.

HYDROCARBON BURNER. Willis W. Case, Sr., Denver, Colo., assignor to the Denver Fire Clay Company, Denver, Colo., a Corporation of Colorado. 971,489.

APPARATUS FOR REMOVING GASES AND VAPORS FROM SURFACE CONDENSERS. Emil Josse, Berlin, and Wilhelm Gensecke, Charlottenburg, Germany. 971,505.

ELECTRICAL INVENTIONS AND APPLICATIONS

SYNCHRONOUS INDUCTION MOTOR. Carl A. Lohr, Chicago, Ill. 971,056.

CIRCUIT BREAKER. Frank L. Sessions, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 971,082.

SYSTEM OF MOTOR CONTROL. Leopold Janisch, Berlin, Germany, assignor to General Electric Company, a Corporation of New York. 971,199.

PORTABLE ELECTRIC LIGHT. Joseph Block, New York, N. Y. 971,241.

ELECTRIC RAILWAY SAFETY SYSTEM. Antonio Casale, Scranton, Penn. 971,249.

SAFETY LIMIT SWITCH. William W. Pierce and George A. Pierce, Philadelphia, Penn. 971,307.

INCANDESCENT ELECTRIC LAMP. Anton Lederer, Atzgersdorf, near Vienna, Austria-Hungary, assignor to Westinghouse Lamp Company, Bloomfield, N. J., a Corporation of Pennsylvania. 971,453.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

One hears about thought transference—about projecting one's thoughts into another man's mind—and marvels.

As a trick or amusement it can't be equalled—

As an adjunct to a modern business it isn't worth one iota.

One cannot employ a telepathist to project a thought—without words—to 30,000 or 40,000 people, at so much per projection.

One cannot "wish" a man into buying anything without giving him some tangible "reasons why."

Yet there are manufacturers who unconsciously seem to have this idea.

They think that because *they* know a thing to be so—that their knowledge naturally is common knowledge.

They think that because they make a good device the world will become aware of it without their going to any trouble telling about it.

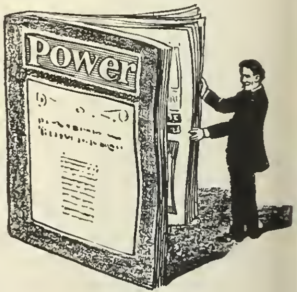
But this class of manufacturer is rapidly becoming converted—he is gradually coming to see that the only way to reap a real harvest is to harness modern methods to his business.

The bulk of the manufacturers of power-plant devices *know* that the power-plant man buys only upon solid, can't-get-away-from facts.

They know that power-plant men do their own thinking, that they do their own selecting, and that when they select, it is only because the device in question has been proven to be one they need—

For the saving it will effect, or for the improved work it will help accomplish.

And consequently these manufacturers tell you, in the Selling Section of your paper, the points that make their devices better, more serviceable, more economical.



They *show you* why their goods *should* be used, and they let *you* be judge as to whether or not their goods should be used in *your* plant.

These are the manufacturers who are "reaping the harvest."

They tell their stories in such a way that you can't help but get valuable ideas from them.

They put you in the way of thinking up ways and means of lessening expenses, and increasing efficiency—

Which means progress for you *personally*.

These are manufacturers you should listen to with much attention.

Their talks are meaty, full of real arguments—they do not depend upon "mental telepathy" to sell their goods—

They have the "reasons why" and are not afraid to tell them to *you*.

Go through this Selling Section carefully and prove these things to your own satisfaction.

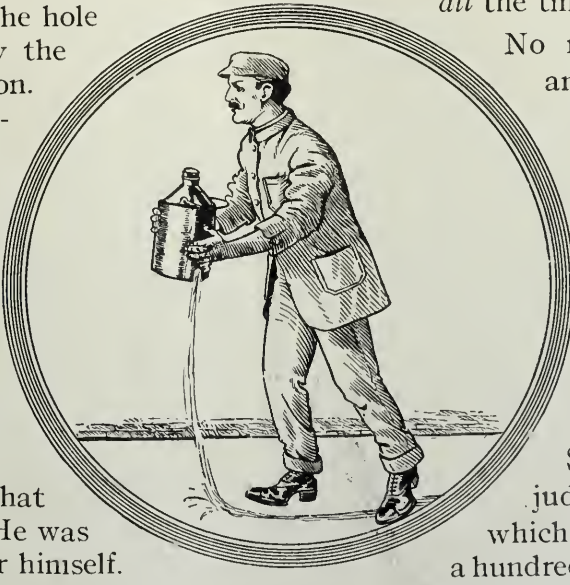
Power
and The Engineer

Vol. 32 NEW YORK, OCTOBER 18, 1910 No. 42

HERE is a true tale. The incident seems so far-fetched that there is considerable risk in using it. However, it is such a pat illustration of what we wish to bring out that the risk is worth while.

A certain artisan was discovered walking along the street carrying one of those cans that are often used for oil or other liquid.

In some way or another a hole had been staved in the side of the can near the bottom and our hero was having an awful and awkward time trying to dam the leak by holding one hand over the hole while he continued to carry the can in an upright position. He was only partly successful in his purpose, for he was leaving a trail of drops which leaked through his fingers. It never occurred to him that he could stop the loss by the simple expedient of carrying the can in a horizontal position with the hole facing upward.



That lad badly lacked what we call common sense. He was utterly unable to think for himself.

He could not reason out that the can might advantageously be carried in any other than the normal position.

Common sense is a curious thing inasmuch as everyone, secretly at least, firmly believes that he possesses it in its true form and suspects that about everybody else lacks it, at some time or other at least. What one man may consider to be good hard common sense is often judged by others to be the limit of foolishness.

Perhaps one way in which proof of a man's exercise of common sense may be given is by "delivering the goods." Had the man who has just been mentioned "delivered the goods," that is, all of the goods in the can, he would probably have escaped being the goat in this talk.

Engineers—real engineers, not near engineers—more than the men of any other class, must possess and continually use common sense. They must "deliver the goods" *all* the time.

No matter how much schooling and training a man has received and how much experience he has had to guide him, unless he seasons it all with a few grains of common sense he is a dead loss.

Common sense is often inborn; it comes to some as naturally as breathing. Such people can give snap judgments and perform acts which in ninety-nine cases out of a hundred are exactly right, although they may have absolutely no precedents to guide them.

However, common sense may be acquired. In fact, it is by effort and study that most of us come to possess the modicum that we have.

If you practise thinking for yourself and reasoning things out—making *use* of your brain—it will not be long before the practice will become a *habit* and men will begin to discover that you have plenty of good hard common sense.

Large Southern Cotton Mill Plant

By Warren O. Rogers

> *Cross-compound direct-connected units generate current for two large cotton mills having a total of 150,000 spindles. Current is stepped up from 600 to 1500 volts and transmitted two miles by aluminum wires.*

The cotton mill of the White Oak plant, Greensboro, N. C., is one of the largest in the Southern States; it gives employment to about three thousand persons and is rated as a 75,000-spindle mill, the spinning-mill building being 155 feet wide and 720 feet long.

Such a structure with its vast amount of machinery would alone require considerable steam and power. A second plant, known as the Proximity mill, of practically the same output, also obtains its power from this same plant, so that the installation at the White Oak mill ranks among the largest of its kind in the South. Both the White Oak and Proximity mills make blue indigo denims; the annual output is 150,000 yards per day, and the yearly cotton consumption is approximately 60,000 bales.

ENGINE ROOM

The engine and generator room is 68 feet wide and 263 feet long. It contains four Filer & Stowell high-duty cross-compound engines, each having 30 and 60 by 60-inch cylinders, and running at a speed of 75 revolutions per minute.

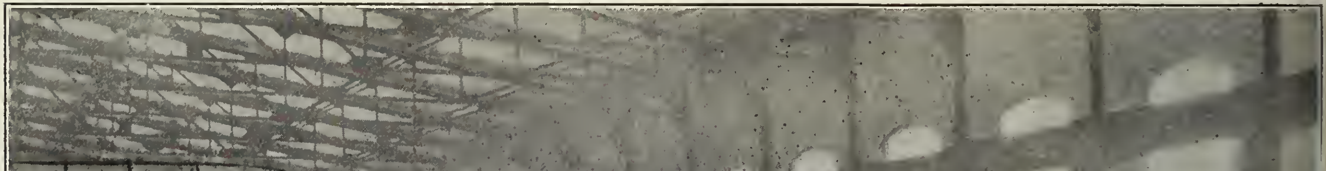


FIG. 1. VIEW OF ENGINE ROOM OF THE WHITE OAK COTTON MILL POWER PLANT

Two of the engines are connected to General Electric alternating-current generators of 1500 kilowatts capacity; the other two engines are connected to 1250-kilowatt alternators. All machines generate at 60 cycles and 600 volts. The engines run at 150 pounds steam pressure, with a vacuum of from 26 to 26½ inches. The receiver pressure carried averages seven pounds. Fig. 1 shows a general view of the engine room which is roomy and well lighted.

On the switchboard side of the engine room are placed three exciter sets, two of which are induction motor driven, the third being steam driven. The larger motor, which has a capacity of 187 horsepower, is direct connected to a continuous-current generator of 1000 amperes capacity. The smaller motor-driven unit consists of a 75-horsepower induction motor connected to a generator of 400 amperes capacity. The steam-driven unit consists of a 50-kilowatt direct-current generator direct connected to a 12x18-inch vertical steam engine, running at a speed of 280 revolutions per minute. The exciter sets, including the engine, are of General Electric make.

BOILER ROOM

The boiler house is 51 feet wide by 263 feet long, and contains 26 Heine water-tube boilers set in three batteries. Two of the batteries contain eight boilers, each of 200 horsepower capacity; the third battery has ten 250-horsepower boilers, which makes a total of 5700 nominal boiler horsepower. The overload capacity is 8000 horsepower. The average pressure carried is 155 pounds per square inch, but the boilers are capable of carrying 200 pounds pressure.

Each battery of boilers is equipped with a Green fuel economizer placed over the boilers and next to the wall. Fig. 2 shows a view of the boiler room, the method of delivering the coal and the arrangement of the steam pipes connecting to the header. Fig. 3 is a plan view of the engine and boiler rooms, and Fig. 4 is a cross-sectional view showing the arrangement of the jet condensers. the arrangement of the economizers an the location of the ashpit tunnel.

Back of each battery of boilers ar located three Hoppes horizontal ope heaters, one of 2000 horsepower and tw of 1500 horsepower capacity. Placed be

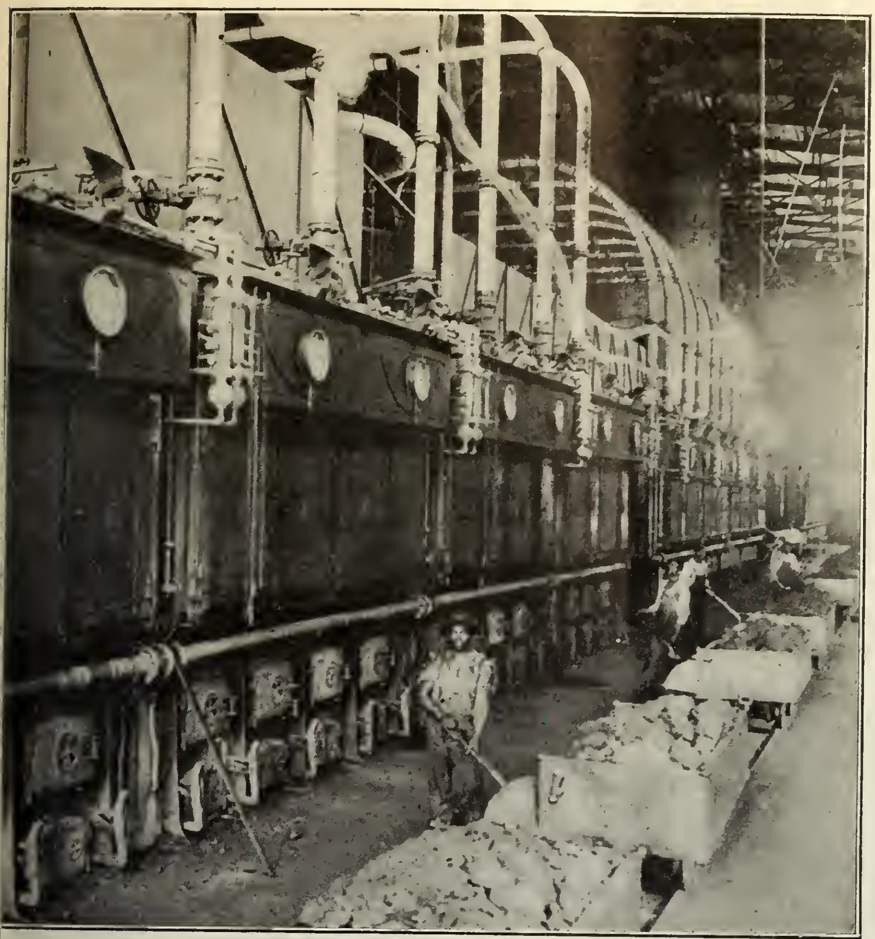


Fig. 2. View of the Boiler Room of the White Oak Cotton Mill Power Plants

veen the setting of the first and second attery of boilers is a Deane 20 and 10 y 18-inch horizontal simplex outside-

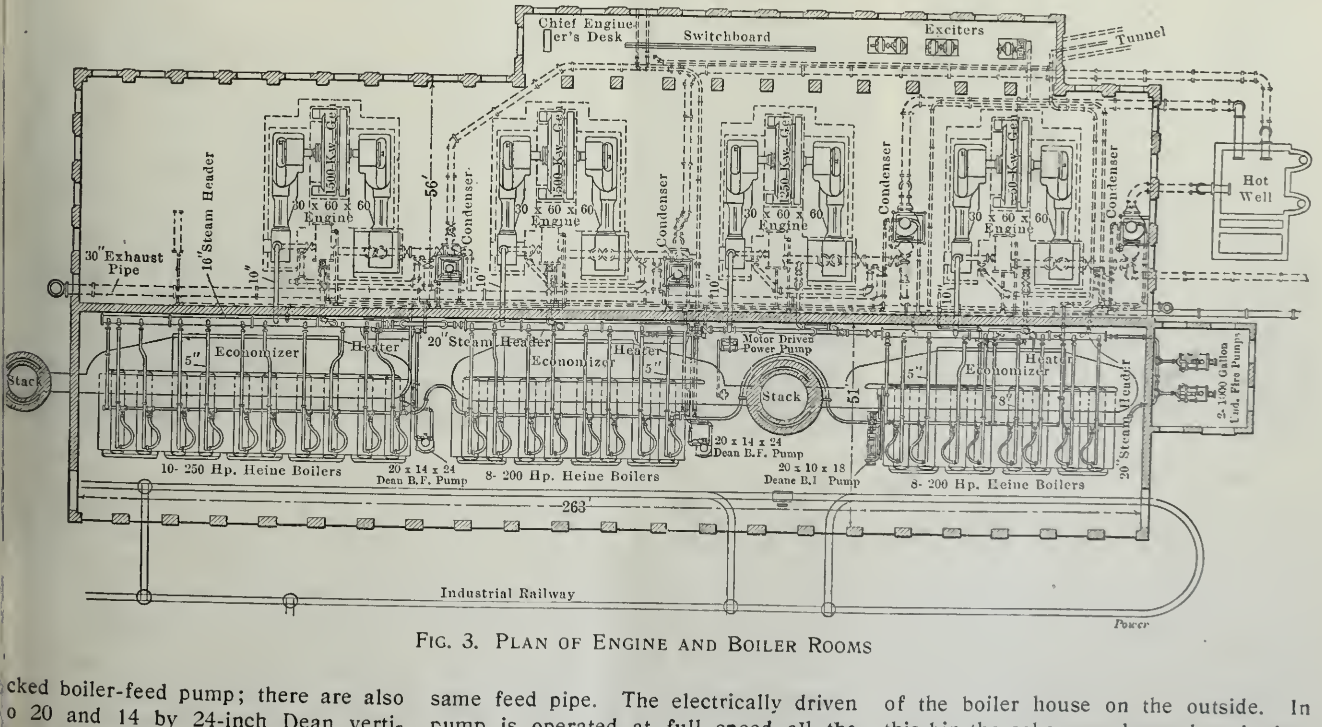


Fig. 3. Plan of Engine and Boiler Rooms

cked boiler-feed pump; there are also o 20 and 14 by 24-inch Dean verti- l boiler-feed pumps, Atlantic type, and e 9x12-inch triple-plunger Deane power pump driven by a 75-horsepower General Electric induction motor partially shown in Fig. 5. These pumps all discharge into the same feed pipe. The electrically driven pump is operated at full speed all the time and the vertical pumps are regulated fast or slow as the water level in the boilers dictate. The feed water to the boilers can be bypassed around the heaters when necessary.

The boilers of batteries Nos. 1 and 2 of the 200-horsepower units are connected to a 20-inch header; the 250-horsepower boilers are connected to a 16-inch header by long 5-inch bends, as shown in Fig. 2. From the steam headers to each engine, 10-inch pipe connections having long-radius bends are installed. A Sweet direct separator is placed above the throttle on each engine connection. The exhaust pipe from each of the engines to the condenser is 22 inches in diameter.

Handling Ashes and Coal

The method of handling the ashes is out of the ordinary. The ashpit floors are sloped from the back of the grates toward the front of the boiler setting, and cleaning-out doors are set in the brickwork forming the boiler front. Under the floor of the boiler room and extending its entire length, is an ash tunnel in which an ash car runs on a narrow-gage track. The ashes from the grates fall into the ashpit proper, and when it is desired to remove them from the ashpit a 2-inch pipe, to which is attached a hose, is placed in the ashes; after opening the cleaning-out doors, water is turned on and the ashes washed out either into the car or into the ash tunnel. In the latter case it is necessary to shovel the ashes into the ash-conveyer car, while in the former instance no handling is necessary. When the ash car has been filled, it is drawn to one end of the ash tunnel, and up an elevated track to a position over a concrete ash bin located at one end of the boiler house on the outside. In this bin the ashes are dumped, and when it is full a wagon is used to cart the ashes away.

The coal used in the plant is brought in hopper cars and delivered to an elevated bin from which it is loaded into small firing cars, and is delivered to the front of the boilers.

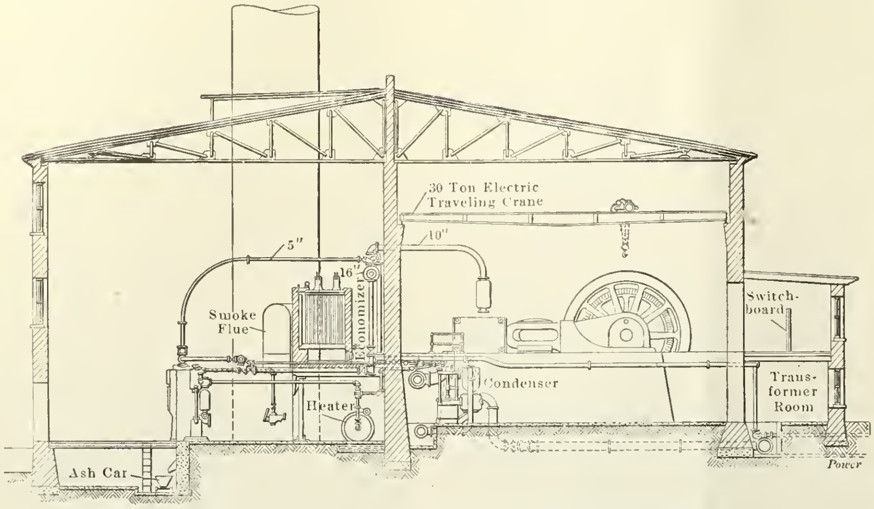


Fig. 4. Elevation of Boiler and Engine Room

Fig. 6 shows the location of the jet condensers, of which there are four, two being of the Deane vertical type, size 20 and 20 by 24 inches, and two of the Dean Brothers vertical type, size 20 and 38 by 18 inches. All are placed below the floor level in the basement.

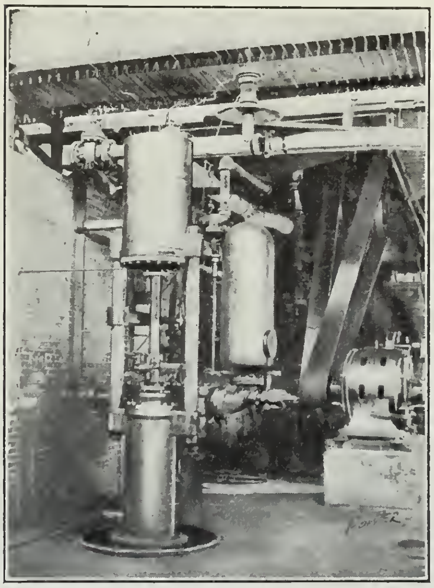


Fig. 5. Steam and Motor-driven Pumps

The engines are all equipped with a gravity-oiling system. The oil returns from the engine bearings to Burt unit oil filters located in the basement, which are arranged as shown in Fig. 7. After being filtered the oil is pumped to an overhead tank.

Piping

The main steam engines are piped to a single main header, while the auxiliaries receive steam from a separate header on which a pressure of 50 pounds is maintained. The auxiliaries may be supplied with steam from the main header if necessary. A nonreturn valve is used at each connection between the boilers and header. The exhaust pipe of each engine is so arranged that the high- or low-pressure cylinders can be operated either condensing or noncondensing. The exhaust pipe from each engine is connected to one atmospheric exhaust pipe, and the auxiliaries are connected to a single exhaust pipe so that the exhaust steam can be passed through the heater or to the atmosphere, as may be desired.

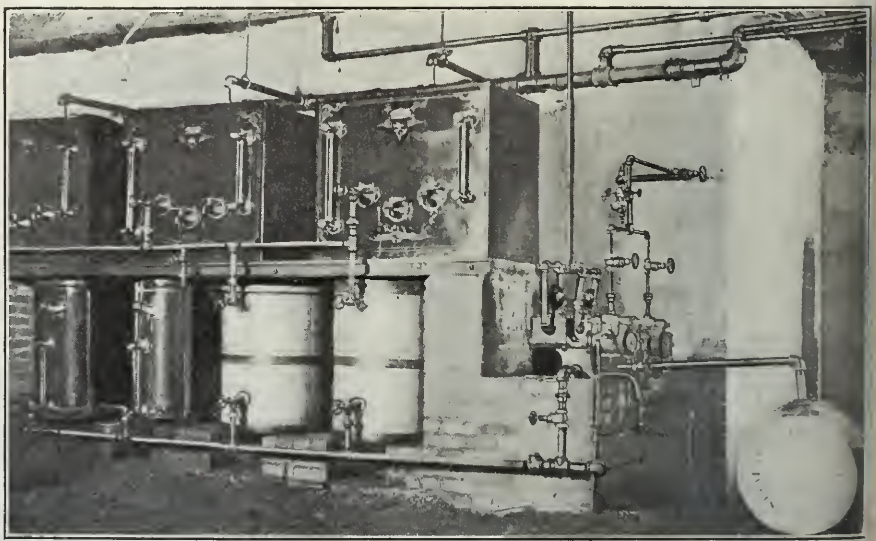


Fig. 7. Arrangement of Oil Filters in the Basement

Extra-heavy fittings are used on all high-pressure steam lines, and all pipes 5 inches and over are fitted with van stone joints. The boiler-feed pipes, fittings and valves are all of brass. The main atmospheric exhaust pipe, double galvanized and spiral riveted, is provided with extra-heavy fittings.

There are two 175-foot brick stacks, one 9 feet and the other 12 feet in diameter. The larger stack takes care of the gases from the sixteen 200-horsepower boilers, and the smaller stack serves the 250-horsepower boilers.

Outside of the boiler room proper are located two 1000-gallon Underwriters' fire pumps which supply the fire-sprinkler system with water. Water for all the departments of the mill is taken from a nearby creek which flows at a rate of about 4000 gallons per minute to a reservoir having a capacity of approximately 10,000,000 gallons.

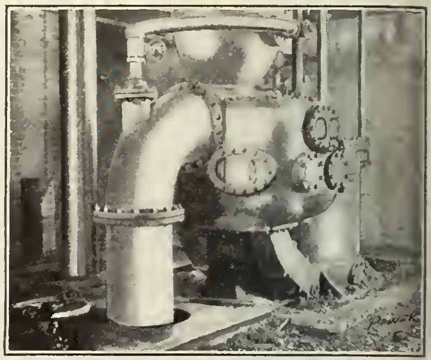


Fig. 6. Jet Condenser Set in the Basement

Outside of the power plant proper is a substation which transforms the current and distributes it to the Proximit mill, a distance of two miles, at 150 volts. The steel towers, carrying hig voltage aluminum wires, are spaced at distance of 600 feet, requiring in roun numbers about 20 towers.

The purchase of coal on the basis of i heat content is of the same utility as definite understanding regarding the qua ity of any other commodity of commerc or industry.

Roller Bearings in the Line Shaft

By John A. Schroeder*

> *It is claimed that roller bearings in the line shaft will cut the friction of babbitted bearings in half, thus reducing the total power bill by from 10 to 20 per cent, and this saving increases with the increase of load.*
>
> *Special requirements of line shaft bearings.*

One of the main causes of waste of power in a manufacturing plant lies in the transmission of power through line shafting, caused by the friction in the bearings. In all other departments vast improvements have been made, tending to effect savings and thus reduce the manufacturing cost. The means for transmitting power through the plant, however, where line shafting is used, are virtually the same today that they have been for years; shafting is still running in babbitted bearings and the only improvements in this line have consisted in introducing various devices for oiling the bearings.

A few years ago there was a tendency to abolish line shafting entirely, running all machines by individual motors. It has been found, however, that unless the machines thus driven are run only intermittently or each machine requires quite an amount of power, the low efficiency of small motors, together with the high cost of the great number of them, with wiring, starting devices, etc., render this system uneconomical. It is generally conceded that the most economical transmission of power is by what is termed "grouped drives." A number of machines are run from one motor, through one or more line shafts.

There is always a great deal of friction where shafting is run in babbitted bearings, even where the shaft is perfectly alined. In most manufacturing plants, hrough the settling of buildings, the oosening of set screws, and the springing of the shafts, this friction load is increased greatly, and the time has come when the manufacturer is asking himself f there is no way or means by which he can reduce his friction load.

In machinery of all kinds ball and roller bearings have been introduced and heir extensive use clearly demonstrates he saving they effect. Of the two, the ball earing, presenting only point contact, is he most efficient and should be used wherever possible. Under heavy and arying loads, however, and in places where bearings must be split, the roller earing is more successful.

When the question arises as to how to educe the friction load in a manufacturing plant, attention naturally is turned to all and roller bearings, and manufacurers of transmission machinery have ately given this matter a great deal of ttention. Ball bearings of several akes, designed for line-shaft purposes, ave been on the market for some time. is, however, impracticable to split a ball ace, and a line-shaft bearing should necssarily be split, in order to be practicble. The attention therefore turns to oller bearings, and some roller bearings successfully used on special machinery and in special places have been applied to line shafts. In some cases they have given satisfaction; in many others not, from the fact that in applying these roller bearings to this service the necessity of changing the construction was not taken into consideration. Recently roller bearings, constructed especially for line-shafting service, in the construction of which care has been taken to meet all requirements to be encountered in such service, have been placed upon the market, and will fill a long felt need and demand.

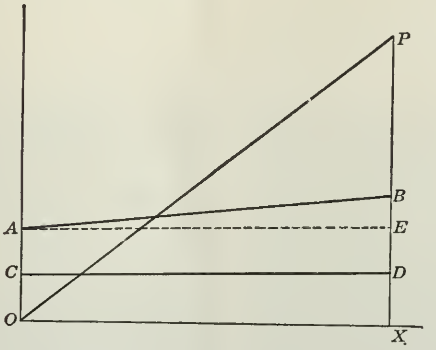


Diagram of Total Power Required and Increase of Friction with Load

The conditions and requirements peculiar to line shafting service are as follows:

Soft steel in the shafts.
More or less end play.
More or less springing in the shaft.

A commercial cold-rolled, cold-drawn or turned shaft is made from a low-carbon, therefore soft steel, and if hard rollers are run directly on this, it will soon show wear. A high-carbon steel bushing should, therefore, be provided, securely clamped to the shaft, so as to present a hard wearing surface for the rollers to revolve upon and to provide an absolute protection to the shaft.

In most roller bearings the rollers or roller structures have been revolved between stationary ends of the bearing boxes. If applied to a line shaft, the end play existing will force the rollers or roller structures against the stationary ends, causing friction and wear. In a roller bearing intended for line-shaft service, the roller structures should, therefore, be of the floating type, absolutely free and clear, and no revolving part should be allowed to run against any stationary part.

The belt on either side of line-shaft bearings will often cause the shaft to spring more or less, this springing causing the shaft to bend in the center of the bearing. If the rollers are run loose in the box, this bending of the shaft may cause them to get out of alinement, to bend and to bind. A properly constructed roller bearing for line-shaft service should, therefore, have the roller alined so as to be kept at all times parallel with one another and with the shaft. In a roller bearing for any class of service, the shorter the rollers are the more efficient is the bearing. The question then arises, how long should the rollers be in order to make the bearing most efficient, consistent with required strength. If a roller is alined so that it is at all times kept parallel with the shaft, it cannot be subjected to anything but an absolute crushing strain. It should, therefore, be of sufficient diameter and sufficient length to withstand a crushing strain equal to, or somewhat above the shearing strain of the shaft. If such is the construction, a roller cannot break unless there should be a flaw in the steel from which it is made.

Consideration must be given to the wear of the bearing. The bearing should be so constructed that the wear be allowed to come on such parts that can be most cheaply and readily replaced. If the rollers are made of a high-grade, high-carbon steel and they are case hardened and ground, they will show very little, or no wear, running against surfaces of softer material. It is a well known fact that steel running against cast iron gives as efficient results in regard to nonwear, as any two metals. If the box itself is made of a good grade of cast iron, there will be very little wear on the box itself. The bushing which protects the shaft is necessarily thin, and could not be made of cast iron. It should, therefore be made of a high-carbon steel, but not case hardened, so that the wear will come on this bushing and none on the rollers. The bushing, naturally, can be cheaply and easily replaced.

The friction load in a manufacturing plant is the power required to run the shafting with all machinery standing idle, in other words, with clutches thrown out, when such are used, or with belts running over loose pulleys, where tight and loose pulleys are used. It is very seldom that the friction load is less than 30 per cent. of the power load. Very often it runs 40 per cent. and above. With properly constructed roller bearings, properly installed, this friction load can be cut practically in half, which means a saving of from 10 per cent. to 20 per cent. of the total power used.

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*Mechanical engineer Royersford Foundry d Machine Company, Philadelphia, Penn.

As load is thrown on the shafts, the friction in the babbitted bearings is increased, due to the increased pressure between shaft and bearing. The friction at full load, that is with all machines running, is, therefore, greater than at no load; viz., where babbitted bearings are used. Where shafts are equipped with roller bearings, however, there is but a very slight increase in friction, if any at all, with increased load, so that the total saving of power by the use of roller bearings in place of babbitted bearings is even greater than the saving in friction load alone. The accompanying diagram will clearly illustrate this.

Line *OP* represents the power required for driving the machinery; with no load at *O*, and full load at *P*. The friction load is represented by line *A B*, with shafting equipped with babbitted bearings, and by line *C D* when equipped with roller bearings. At no load, the power required to drive shafting, the friction load, with babbitted bearings is *A O;* with roller bearings *C O.* As load is thrown on, the friction in the babbitted boxes increase until at full load it is *B X; P B* being the power actually used for driving the machinery. With roller bearings the friction increases very slightly, if at all, with increased load, so that the friction at full load, *D X*, is about the same as *C O* at no load. The saving in so called "friction load" is *A C;* the saving in power at full load is *B D*. Drawing *A E* parallel with *C D*, *B E* is the additional saving in power at full load. The total power required to drive machinery and shafting, with babbitted bearings is *P X;* with roller bearings *P X* less *B D*.

In every manufacturing plant, records should be kept of the power used as well as of the friction load. At regular intervals indicator diagrams should be taken from the engine, or readings of the motors, with friction load only, as well as with full load. If either is found excessive, it should be reduced and kept reduced. The line shafting should be kept properly alined and whenever possible, roller bearings should be used. No better investment can be made than substituting roller bearings for the babbitted bearings, if thereby a saving can be effected of from 10 per cent. to 20 per cent. of the total power used, the net result of lessening the friction load by one-half. A saving of 10 per cent. in the total power will, as a rule, pay for the cost of substitution in a year's time or less.

It must be remembered that roller bearings should not be used indiscriminately. When line shafting is run at unusually high speeds, or when the load is in the nature of shocks, such as in a forge shop, where drop hammers are used, they are unsuitable. The continual shocks, reverting back to the rollers, will gradually crystalize these until they finally break.

Explosion of Cast Iron Steam Pipe*

An 18-inch cast-iron pipe, carrying 100 pounds steam pressure, exploded as a result of a crack which had been started by tightening too much on the flange bolts. The packing was wholly within the bolt circle.

A section of 18-inch cast-iron steam piping about 5 feet in length recently exploded at the steel works of John Spencer & Sons, Newcastle-on-Tyne. The pipe had a flange 1⅜ inches thick at one end, stiffened by eight ribs, and the other end, which was plain, fitted into the stuffing box of an expansion joint. The thickness of the metal varied from ¾ to 1$\frac{1}{16}$ inches and the section was located near the engine, as shown in Fig. 1.

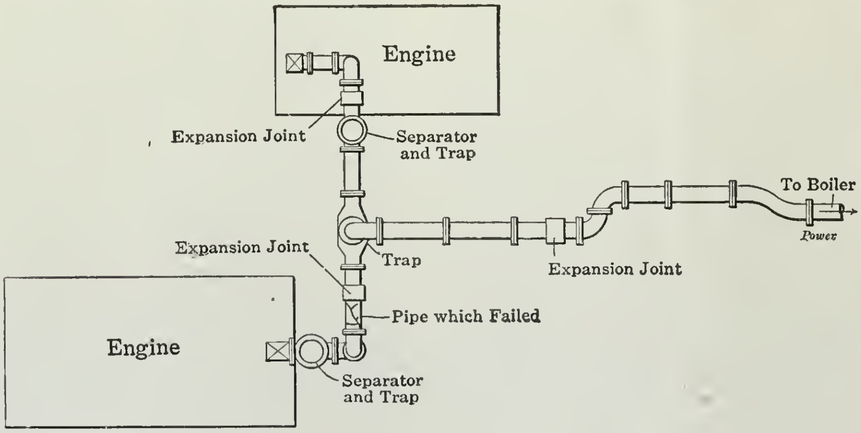

Fig. 1. Showing Location of Pipe Which Failed

The safety valve of the boiler was set to blow off at 100 pounds pressure, but the pipe had been designed for a pressure of 120 pounds and withstood a hydraulic test of 200 pounds.

The flange joints were of a type commonly found in cast-iron piping, a rubber ring about 1½ inches wide by ¼ inch thick being fitted inside the bolt circle. Three days before the explosion the joint was repacked; the attendant took out four of the bolts, slackened up on the other four, scraped the joint and inserted a new rubber ring. When replacing one of the bolts he noticed a crack in the flange, extending from the edge through the hole to the fillet and about 2 inches along the pipe. A 5x9-inch plate was fitted over the crack and held in place by one of the flange bolts and a band extending around the pipe. The flange bolts were then tightly screwed up by means of a spanner lengthened by a piece of tube till it had a leverage of nearly 4 feet.

When steam was turned on the crack leaked considerably, but it was though that the temporary repair would hold until a new section could be procured Three days later, however, an explosion occurred, entirely wrecking the pipe bu injuring no one. The fractured section is shown in Fig. 2.

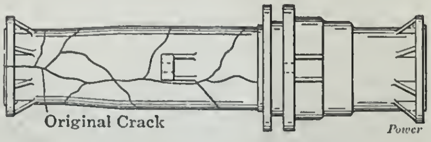

Fig. 2. Fractured Pipe

It is believed that the crack was caused by tightening too much on one of the flange bolts. Where jointing material is placed wholly inside the bolt circle, undue stresses are set up in the metal when the bolts are screwed up, and failures have often been due to this cause. It is preferable that the jointing material extend across the face of the flange.

In case of low water, cover the fir with wet ashes, or with green fuel if we ashes are not available, and close the fir and ashpit doors. Do not touch the safet valve. Do not turn on the feed wate under any circumstances. Let the steam outlets remain as they are until the boiler has been cooled off and examine to determine what damage has been done.

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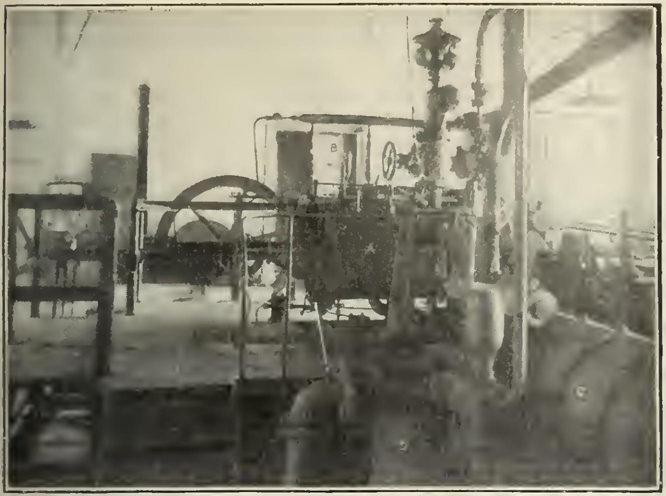
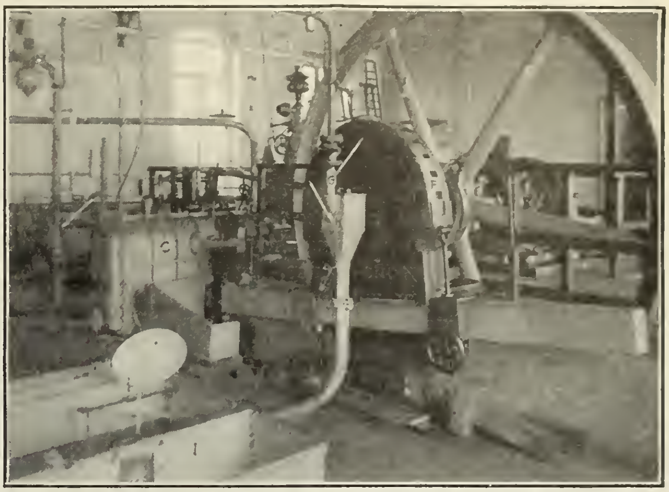
*From report of British Board of Trade.

Laboratory of Dwelshauvers-Dery

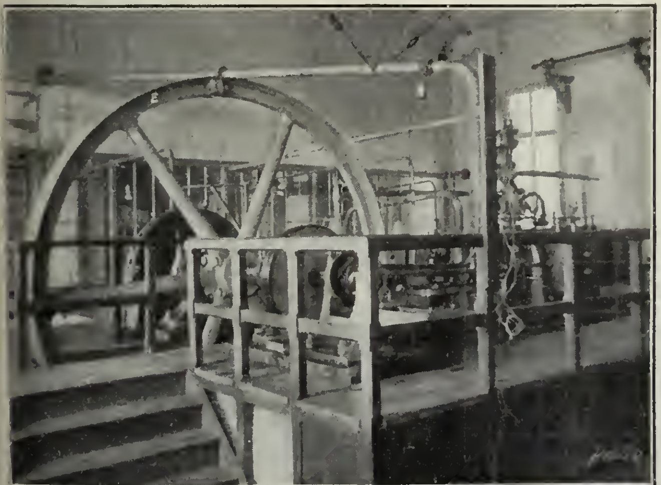
Since the article descriptive of our visit to the laboratory of the University at Liège appeared upon page 1617 of the issue of September 6, we have received from Professor Dwelshauvers-Dery the ccompanying views of the interior of that boratory, showing the engine and aparatus upon and with which were made e famous tests which showed that compression was not a cure for clearance, which tests were reviewed in our issue of June 28, with a complete description of the engine and apparatus here illustrated. The conclusions reached by Professor Dery were questioned by Professor Heck in our issue of September 13, to which Professor Dery replies in our issue of September 27. We should be glad to see the subject call out still more discussion and to see some of our laboratories take the matter up for still further experimental determination. In the meantime these photographs of the steam machinery and apparatus of the tests under discussion will be interesting, the engine in particular and what little can be seen of the original Richards' indicator.



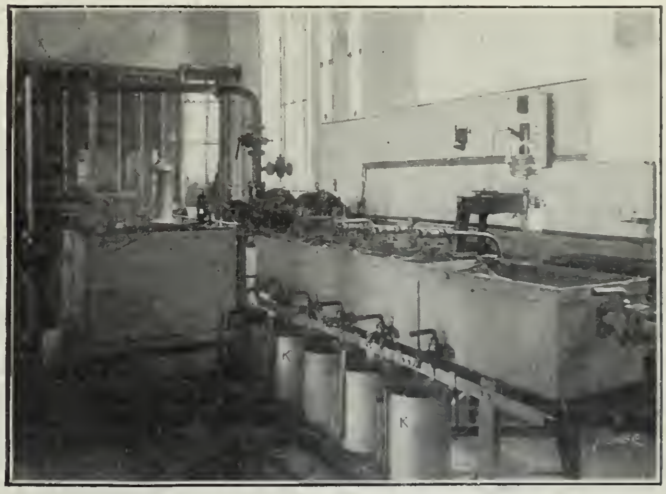
A, cylinder of the famous Beer engine. The guides of this engine carry a scale divided into thousandths part of the stroke (0.0234 inch), with a vernier, making it possible to locate the position of the piston to 0.002 of an inch. The riding cutoff valves have separate scales and verniers, all reading to 0.004 inch.
B, Richards indicator purchased at the exposition of 1862.
C, surface condenser.
D, air pump with thermometer at outlet.



F, brake pulley.
G, thermometer at inlet of water to brake.
H, thermometer at outlet.
I, gaging tank fitted with weir for measuring water supplied to brake pulley. The water level is measured with a scale which has to be read with a lens. The addition of four or five drops in a tank of 500 gallons may be at once detected.



Crank side of engine with brake *F* in background and flywheel to the fore.



K, measuring tanks for gaging water of condensation from jackets, valve chest and pipes.

EXPERIMENTAL STEAM ENGINE AND EQUIPMENT IN THE ENGINEERING LABORATORY AT LIÈGE

The concrete pipe line just completed r the Ontario Power Company is deared to be unique in its engineering feares among the great pipe lines of the ontinent. It is, in fact, the only one of s kind and size in the world. In diameter it is 18 feet, and a mile and a quarter in length. Costing over $1,000,000, it took but five months to complete it. Resting on a solid foundation of concrete, it traverses a section of almost every known variety of soil. Rock, gravel, loam and quicksand were encountered. The number of bags of cement used was 247,642, besides 2350 tons of steel. Its walls are fully 18 inches in thickness. The big pipe could easily be used as an underground-railway tube.

Standardization of the Boiler Room

By William A. Russell

A concrete example involving tests and investigations into the boiler-room conditions of a plant. A method is cited of making standardization tests of the performance of a boiler whereby coal can be bought on a B.t.u. basis to suit any market prices.

Many good articles and books have been written upon boiler-room economy; CO_2 has been discussed at great length, bitter war has been waged on the question of embodying a heat-unit basis in coal specifications, and much trade literature exists on the operation of stokers and mechanical-draft appliances; still there exists, to an almost unbelievable extent, the "happy-go-lucky" method of selecting the kind and grade of fuel to be used.

When the man responsible for the use of a certain fuel is asked why he uses a particular kind and grade, in the majority of cases he will give no other reason than the routine and the profound conviction that no other kind would be as economical as the one in use. In a few cases the excuse will be given that the men are accustomed to this kind and would not know how to use another, or that the fuel was changed some years ago and therefore must be the most economical. Usually the purchasing agent for the plant uses all his shrewdness and experience in trying to force down the price of fuel, while at the same time the "conviction" of the engineer and the absence of a little curiosity and experimenting on his part, may cost the owner much more than the shrewdest buying can save.

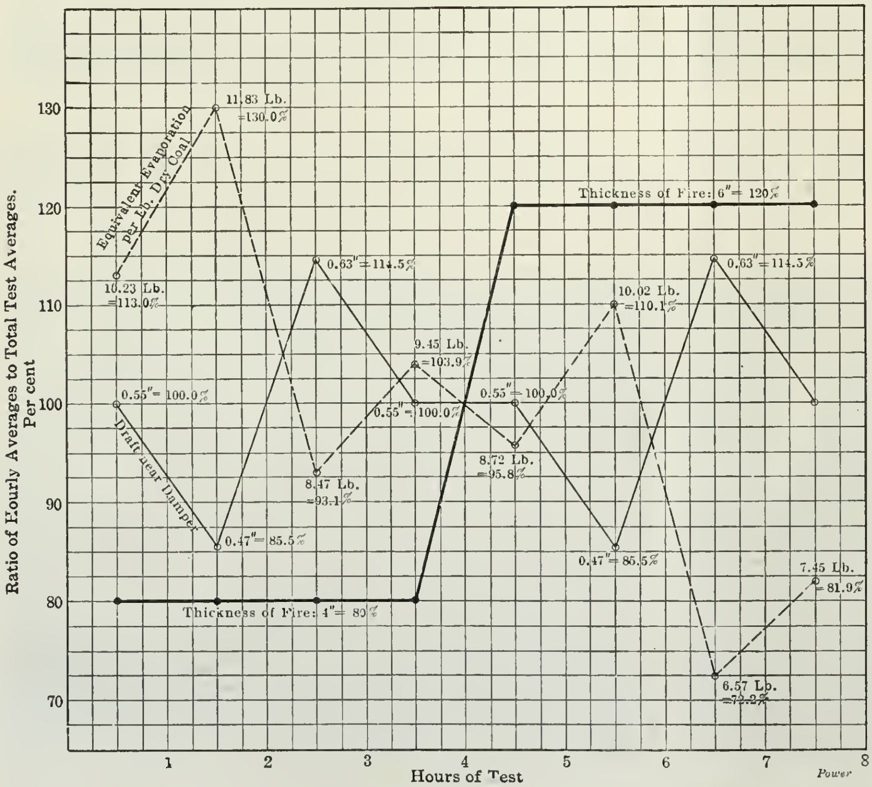


CHART OF STUDY TEST WITH BUCKWHEAT NO. 1

The process by which the seemingly complicated problem of selecting the most economical fuel can be solved so as to take into account the fluctuations in coal prices, is very simple and is easily accomplished in everyday practice. Of course, this method does not go into academic refinements, but by substituting experience and knowledge for pure guesswork, it tends to guide correctly the man who is desirous of saving his money. The writer believes that the method herein presented will be best explained by citing a concrete case where he was called upon recently to act in a consulting capacity.

The plant was a large water-pumping station, the boiler rooms of which contained twenty boilers having a total rated capacity of about 4000 horsepower. These boilers were distributed in two boiler rooms, each room having a separate stack. Boiler room *A* had twelve horizontal fire-tube boilers with inside furnaces, each with a rated capacity of 200 horsepower; and boiler room *B* had eight horizontal tubular boilers of the same rated capacity. In boiler room *A* it was found necessary to run not more than nine boilers at a time; two of them were usually out of service for repairs and at least one had to be kept in reserve for emergency purposes; hence, the ordinarily required capacity of this boiler room was 1600 horsepower. In boiler room *B* it was found possible to run six boilers at once, keeping two idle for repairs and reserve; hence, the capacity ordinarily required for boiler room *B* was 1000 horsepower.

The plant was hand fired when the investigation was begun and broken-sized anthracite coal was used. The particular conditions of the plant made the prevention of smoke a fundamental condition. Preliminary investigations of the fuel problem made it clear that the following combinations could be considered:

For hand firing,
- Broken coal having an average of 13,070 B.t.u. as delivered.
- Pea coal having an average of 11,930 B.t.u. as delivered.
- Buckwheat No. 1 having an average of 11,840 B.t.u. as delivered.
- Buckwheat No. 3 having an average of 11,750 B.t.u. as delivered.
- A mixture of 4/5 buckwheat No. 3 and 1/5 bituminous slack having a total average of 12,060 B.t.u. as delivered.

For mechanical stoking,
- Semi-bituminous "run-of-mine" having an average of 14,500 B.t.u. as delivered.
- Bituminous slack having an average of 13,300 B.t.u. as delivered.

Owing to the labor conditions, the firemen, although getting $3 per 8 hours' shift, could not be made to handle more than two boilers with the larger sizes of coal and one and one-half with smaller sizes, especially with the contemplated mixture. The cost of delivering coal to the boiler room and that of removing the ashes from it, had been taken into consideration as there were a number of so called "laborers" around the plant who did this work. With mechanical stokers installed it was found to be possible for one fireman to handle as many as eight boilers with the occasional assistance of these laborers.

The first step was to determine through a series of evaporative tests the maximum efficiency and the resulting capacity with each of these fuels. A careful inquiry among different manufacturers of mechanical stokers elucidated the fact that they would guarantee an efficiency in boiler room *A* of 72 per cent. (boiler and grate) and in boiler room *B* an efficiency of 65 per cent. for semi-bituminous "run-of-mine" with a capacity of 133 per cent. of the rating in both cases. In view of the stipulation that the combustion must be as smokeless as possible, these companies did not undertake to guarantee an efficiency of more than 65 per cent. in boiler room *A* and 60 per cent. in boiler room *B* with 100 and 110 per cent. respective capacities with bituminous slack. The depreciation of the mechanical stokers for boiler room *A*, including chutes, etc., reduced to a rate per 10,000 pounds equivalent evaporation and including a comparatively large estimated yearly cost for repairs, was placed at somewhat less than 2 cents. The cost of each fireman in boiler room *A*, receiving $3 wages per 8 hours' shift and reduced to a rate per 1000 pounds equivalent evaporation, amounted to 7 cents. For boiler room *B* these figures were 3 cents and 1.1 cents respectively.

A series of tests were run with the five grades of fuel previously mentioned under "hand firing." The tests were run in conformity with the standard code of the American Society of Mechanical Engineers, two for each fuel and each of eight hours' duration. These tests were all made on the same boiler and within three weeks, the weather conditions being constant during that time, and the soot being blown out after every second test. The first test for each fuel was in the nature of a study test.

Of the many variable conditions influencing the efficiency and capacity of a boiler, two can be easily regulated by the men firing the boilers: namely, the draft which can be controlled through regulation by individual hand dampers with the help of fixed draft gages, and the average thickness of the fuel bed. Therefore, with the general conditions during the whole period of these tests approximately constant, the writer decided to concentrate his entire attention on these two factors, and to study them in a systematic way.

TABLE 2.

No. of Test.	Kind of Coal.	Efficiency, Per Cent.	Capacity, Per Cent.
1	Broken...........	72	105
2	Pea...............	75	100
3	Buckwheat No. 1..	80	90
4	Buckwheat No. 3..	72	75
5	Mixture, buckwheat No. 3 and slack.	65	85

TABLE 3.

No. of Test.	Kind of Coal.	Efficiency, Per Cent.	Capacity, Per Cent.
1	Broken...........	70	110
2	Pea...............	72	100
3	Buckwheat No. 1..	75	85
4	Buckwheat No. 3..	68	70
5	Mixture, buckwheat No. 3 and slack.	70	85
6	Semi-bituminous "run of mine"..	65	135
7	Bituminous slack..	60	110

Each study test was subdivided for this purpose into halves of four hours each, and during each of these periods the thickness of the fire was kept constant and the draft, near the damper, was subjected to exactly the same variations in each half of the test. The fires were cleaned one hour before the start and one hour before the end of each test. Careful observations were made of the water and coal, firing being allowed only at fixed intervals and the weight of each charge being recorded.

Table 1 represents the data of a satisfactory study test, with buckwheat No. 1, and from this the accompanying chart has been compiled in a somewhat novel but rather convenient way. On the middle ordinate of each hourly period are marked off the average thickness of the fire, the average draft near the damper and the average equivalent evaporation per pound of dry coal. All of these are expressed in per cent. of the corresponding averages for the whole test:

TABLE 1. STUDY TEST.

Periods.	Thickness of Fires.	Draft near Damper.	Equivalent Evaporation per Pound of Dry Coal.
First hour...	4 inches = 80 per cent.	0.55 inch = 100.0 per cent.	10.28 pounds = 113.0 per cent.
Second hour.	4 inches = 80 per cent.	0.47 inch = 85.5 per cent.	11.83 pounds = 130.0 per cent.
Third hour..	4 inches = 80 per cent.	0.63 inch = 114.5 per cent.	8.47 pounds = 93.1 per cent.
Fourth hour.	4 inches = 80 per cent.	0.55 inch = 100.0 per cent.	9.45 pounds = 103.9 per cent.
Fifth hour...	6 inches = 120 per cent.	0.55 inch = 100.0 per cent.	8.72 pounds = 95.8 per cent.
Sixth hour..	6 inches = 120 per cent.	0.47 inch = 85.5 per cent.	10.02 pounds = 110.1 per cent.
Seventh hour	6 inches = 120 per cent.	0.63 inch = 114.5 per cent.	6.57 pounds = 72.2 per cent.
Eighth hour.	6 inches = 120 per cent.	0.55 inch = 100.0 per cent.	7.45 pounds = 81.9 per cent.
Averages....	5 inches = 100 per cent.	0.55 inch = 100.0 per cent.	9.10 pounds = 100.0 per cent.

The second test for each fuel was run with a draft near the damper and a thickness of the fire as close as possible to the ones which jointly had shown the best evaporation during the corresponding study test. For instance, during the second test with buckwheat No. 1 the draft was kept as close as possible to 0.47 inch and the thickness of the fire was kept between 3½ and 4½ inches. The result showed very high, giving an efficiency of the boiler and grate of 80.1 per cent., which corresponsd to an evaporation of 10.36 pounds from and at 212 degrees per pound of combustible.

The results of the five final tests are given in table 2.

The guarantees of the stoker companies, as heretofore mentioned, were efficiencies of 72 and 65 per cent. at capacities of 133 and 100 per cent. with semi-bituminous "run-of-mine" and bituminous slack respectively.

A similar series of tests in boiler room *B* gave the following data, including the guarantees of the stoker companies:

Tables 2 and 3 contain the standards for the boiler rooms *A* and *B*.

Of course, it might have been advisable to try one more fuel in both boiler rooms, namely, a mixture of ¼ bituminous slack and ¾ buckwheat No. 3, but this was refrained from because the firemen were not always able to prevent the formation of smoke of greater density than was permissible.

After having determined the standards as explained, the rest of the economical calculations is simple. It remains only to ascertain the prices for the different kinds and grades of coal, and from these and the average heat values and efficiencies, to calculate the cost of an equivalent evaporation of 1000 pounds; then by adding the labor costs, depreciation and maintenance charges on special apparatus, the problem is solved.

TABLE 4. BOILER ROOM A.

Kind of Coal.	Price per Gross Ton.	B.t.u. per Pound as Delivered.	Price per Thousand B.t.u. as Delivered.	Capacity in Boiler Horsepower per Boiler.	Number of Boilers Needed.	Number of Firemen Needed.
Broken....................	$4.95	13,070	0.1691	210	8	4
Pea.......................	4.05	11,930	0.1515	200	8	4
Buckwheat No. 1...........	3.10	11,840	0.1169	180	9	6
Buckwheat No. 3...........	2.05	11,750	0.0779	150	11	7
Mixture, $\frac{4}{5}$ buckwheat No. 3 and $\frac{1}{5}$ slack..............	2.24	12,060	0.0829	170	10	7
Semi-bituminous, "run of mine"..................	3.25	14,500	0.1001	266	6	1
Bituminous slack..........	3.00	13,300	0.1007	200	8	1

TABLE 5. BOILER ROOM B.

Kind of Coal.	Price per Gross Ton.	B.t.u. per Pound as Delivered.	Price per Thousand B.t.u. as Delivered.	Capacity in Boiler Horsepower per Boiler.	Number of Boilers Needed.	Number of Firemen Needed.
Broken....................	$4.95	13,070	0.1691	220	5	3
Pea.......................	4.05	11,930	0.1515	200	5	3
Buckwheat No. 1...........	3.10	11,840	0.1169	170	6	4
Buckwheat No. 3...........	2.05	11,750	0.0779	140	7	4
Mixture, $\frac{4}{5}$ buckwheat No. 3 and $\frac{1}{5}$ slack..............	2.24	12,060	0.0829	170	6	4
Semi-bituminous, "run of mine"..................	3.25	14,500	0.1001	266	4	1
Bituminous slack..........	3.00	13,300	0.1007	220	5	1

In this particular case, the following quotations were received per gross ton:

Broken	$4.95
Pea	4.05
Buckwheat No. 1......................	3.10
Buckwheat No. 3......................	2.05
Semi-bituminous "run-of-mine"........	3.30
Bituminous slack.....................	3.00

In calculating the cost of evaporating 1000 pounds of water the capacity standard must be taken into consideration; this will enable a determination of the cost of the firemen and the depreciation and maintenance charges for the mechanical stoker. The unit costs of these items have been already quoted in the first part of this article. Tables 4 and 5 are self-explanatory:

TABLE 6. PRIMARY COST OF 1000 POUNDS EQUIVALENT EVAPORATION IN BOILER ROOM A.

Kind of Coal.	Cost of Fuel, Cents.	Cost of Labor, Cents.	Depreciation of Mechanical Stokers, Cents.	Total Cost, Cents.
Broken	22.7	2.8	...	25.5
Pea	19.5	2.8	...	22.3
Buckwheat No. 1	14.1	4.2	.	18.3
Buckwheat No. 3	10.4	4.9	..	15.3
Mixture, $\frac{4}{5}$ buckwheat No. 3 and $\frac{1}{5}$ slack	12.3	4.9		17.2
Semi-bituminous, "run of mine"	13.4	0.7	0.2	14.3
Bituminous slack	15.0	0.7	0.2	15.9

TABLE 7. PRIMARY COST OF 1000 POUNDS EQUIVALENT EVAPORATION IN BOILER ROOM B.

Kind of Coal.	Cost of Fuel, Cents.	Cost of Labor, Cents.	Depreciation of Mechanical Stokers, Cents.	Total Cost, Cents.
Broken	23.3	3.3	...	26.6
Pea	20.3	3.3	..	23.6
Buckwheat No. 1	15.1	4.4	...	19.5
Buckwheat No. 3	11.1	4.4	..	15.5
Mixture, $\frac{4}{5}$ buckwheat No. 3 and $\frac{1}{5}$ slack	11.4	4.4	...	15.8
Semi-bituminous, "run of mine"	14.9	1.1	0.3	16.3
Bituminous slack	16.2	1.1	0.3	17.6

Tables 6 and 7 were calculated from the preceding efficiencies and cost data:

Under the particular market conditions stated in the foregoing, the semi-bituminous, "run-of-mine" coal proved to be the most economical for boiler room *A*. For boiler room *B*, the most economical fuel would have been buckwheat No. 3, but as this would have necessitated the running of seven boilers, which is more than the original conditions allowed, the somewhat less economical mixture of 4/5 buckwheat No. 3 and 1/5 bituminous slack was selected.

On the basis of this calculation, founded on the prevailing market prices at the given time, the writer advised the firm to contract for coal containing the following number of heat units:

Boiler room *A*,

$$\frac{1600 \times 24 \times 365 \times 34.5 \times 965.7^{*}}{0.72} = 648{,}564$$

million B.t.u.

of semi-bituminous "run-of-mine" coal at the guaranteed price of $0.1001 per million B.t.u. No penalties were advised, except that only analyzing below 13,500 B.t.u. per pound as delivered and above 12 per cent. ash in dry coal be rejected.

Boiler room *B*,

$$\frac{1000 \times 24 \times 365 \times 34.5 \times 965.7}{0.70} \times \frac{0.2 \times 13{,}300}{12{,}060}$$

= 91,961 million B.t.u.

of bituminous slack at the guaranteed price of $0.1007 per million B.t.u. and,

$$\frac{1000 \times 24 \times 365 \times 34.5 \times 965.7}{0.70} \times \frac{0.8 \times 11{,}750}{12{,}060}$$

= 324,973 million B.t.u.

of buckwheat No. 3 at the guaranteed price of $0.0779 per million B.t.u.

No penalties were to be imposed in this case either, only that coal analyzing worse than the following limits was to be rejected:

Bituminous slack, 12,800 B.t.u. as delivered and 15 per cent. ash.

Buckwheat No. 3, 11,000 B.t.u. as delivered and 18 per cent. ash.

Now, suppose that at the end of the year certain changes in the conditions of boiler room *A* permitted the use of eleven out of the twelve available boilers, and further, that the price of only the semi-bituminous "run-of-mine" had changed, having gone up to $3.75 per gross ton, which would be equivalent to a price of $0.1155 per million B.t.u.

Under these conditions next to the last line in table 6, as well as in table 7, would have to be changed so as to read respectively

$$15.5 + 0.7 + 0.2 = 16.4$$

and

$$17.2 + 1.1 + 0.3 = 18.6.$$

No further boiler tests are necessary; knowing the boiler standards, which are settled once for all as long as the equipment remains the same, the engineer in charge of the plant will presumably decide to change to buckwheat No. 3 straight, in boiler room *A*, thus saving $5320 as compared with the cost of continuing to use semi-bituminous "run-of-mine." Even allowing $1200 for a change of grates in all the boilers, this would leave a saving of $4000 for the year. Boiler room *B* would, of course, continue to use the same mixture as before.

The method of standardizing a boiler room as described in the foregoing concrete instance may seem somewhat cumbersome, but in actual practice it will very seldom, if ever, be necessary to standardize for so many different grades and sizes of coal. Two or three will usually cover the entire range of possibilities, in which case only from four to six evaporative tests will be necessary In the case described, the owners stubbornly insisted upon the larger sizes of anthracite, and to demonstrate clearly to them the wastefulness of these grades, it was necessary to run the complete series of tests. The important feature of this method is that the selection of the fuel to be used can be made scientifically, and independently of anybody's fancy, or biased opinion.

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*At the time this test was conducted the recently accepted value of 970.4 for the latent heat of vaporization at 212 degrees Fahrenheit, was not in general use.

Sid Diggles' ingin busted its guvner belt an' run away. Sid busted his suspenders an' all uv the sprintin' records doin' likewise.

Th' difference 'tween a Missourian an' an alternator is thet one has t' be cited, an' th' other excited 'fore they'll do bizness, b' gosh.

"Reddy" Gallagher sez thet he has th' best uv th' rest uv us when it cums ter goin' inter a biler. Says he leaves his hat off an' don't need any torch.

Th' inspecter called on old Hal Mossback, at th' ladies' rat factory, an' diskivered thet Hal hed bin runnin' fer 17 years on a barber's license thet he'd got when he graddyated frum th' barber college down in Punkville, Indianny.

Old Bill Grimes' firm put in a new biler-feed pump. Bill jined church th' next Sunday. Sed he'd wanted t' jine fer th' last 40 years but cudn't be a Christian an' handle th' dumd old feed pump et th' same time. There's a lot uv us in th' same fix.

Hi Jimson, th' ingineer at th' bustle factory, went up fer license tother day. Th' inspecter ast him what he'd do ef his ingin started t' run away. Hi used t' be a cowboy an' he sed that he'd lasso th' dumd critter an' put th' hobbles on it.

Th' inspecter writ him up a fust-class license, limited t' 23 hosspower.

Pete Blowoff kim over t' borrer sum cylinder ile uv me tother day. Sed he'd set th' cylinder-ile jug too clost t' th one he used hisself an' hed drunk it al up 'fore he noticed th' difference. Sed thet he prob'ly wudn't hev noticed th mistake only thet he'd filled th' lubricate out'n his own jug, an' th' ingin got te cuttin' up sumthin' scandulus.

Sum people think the're ingineers whe they get so they kin hit th' hole in th furnis wunst er twist in a while with scoopful uv coal, an' know enuf t' kee th' steam frum all leakin' out by hangi a grate bar on th' safety-valve lever.

These kind uv ingineers air thicker' fleas on a fat pup in sum parts uv th kentry.

The Fallacy of the Hydrodynamo

Once more the application of a system of floats has been brought into use for the ostensible purpose of creating power out of nothing. The device is described and its fallacy pointed out.

The principles of hydraulics have again undergone distorted interpretations to suit a device called the hydrodynamo which is being promoted by the Indiana Power Company, of Indianapolis. The term *perpetual motion*, which has long been accepted as an inventive nightmare, has been carefully avoided in reference to this device, although the claims are equally, if not more, fallacious than those f the former class. Here great power is upposed to be attained without the expenditure of any energy or of any energy-producing agent, and from the tone of he *prospectus* it would appear that hydraulic engineers have heretofore been groping in the dark as to the great posibilities of hydraulics.

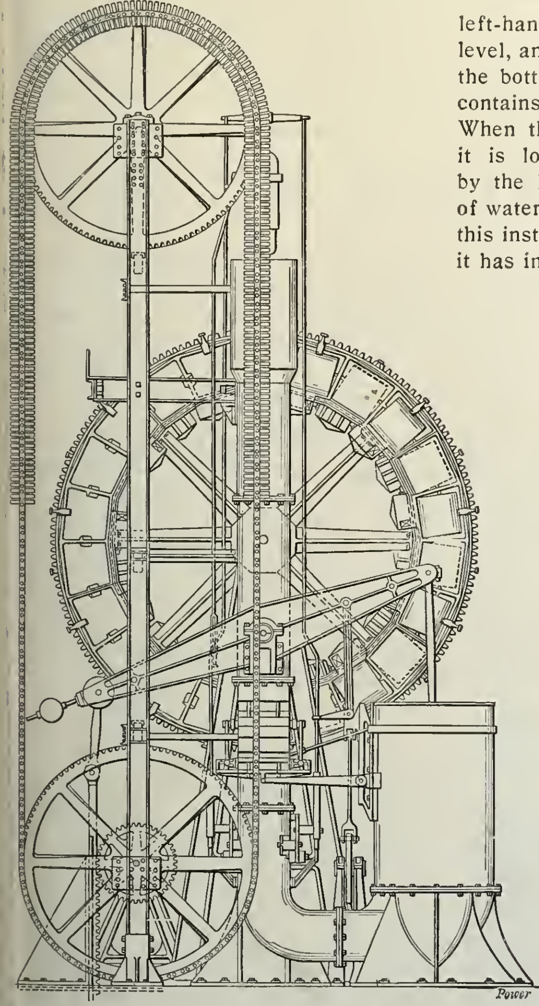


Fig. 1. The Hydrodynamo, Side View

To quote from the prospectus regarding a description of the device, "The prinipal elements of construction are a hydrostatic balance, a series of floats dapted to be raised in the balance, a riving chain operated by these floats, and number of links connecting these parts ogether."

The front view, Fig. 2, represents the left-hand column of water at the receded level, and a float *F*, falling by gravity from the bottom drum into the cylinder which contains air at atmospheric pressure. When the float has entered the cylinder it is locked securely in the pocket *P* by the latch *L*. The right-hand column of water (see the front view, Fig. 2), is at this instant at its highest level. In rising it has immersed a float which was locked in position until the column reached its highest level, at which instant the float was released and, being much lighter than the water which it displaces, it rises and compresses the spring buffer at the top of the column, dropping out of the top by gravity into the waiting compartment of the driving drum. In entering the driving drum, the float trips the latch which holds the drum and the latter moves forward.

In the act of turning forward (the drum being clutched to the hydrostatic balance beam) the hydrostatic balance of the left-hand column, Fig. 1, causes the water to rise to the highest level, while the right-hand column of water is falling, the cylinders being connected by a crossbeam. When the water in the left-hand column is at its maximum hight the float is released, rising to the top and rolling into the compartment at the top of the drum. While the float in the left-hand column is rising, a float has been brought into the floatway at the lower part of the right-hand cylinder and drops by gravity to its locked position; in this way the machine continues to operate.

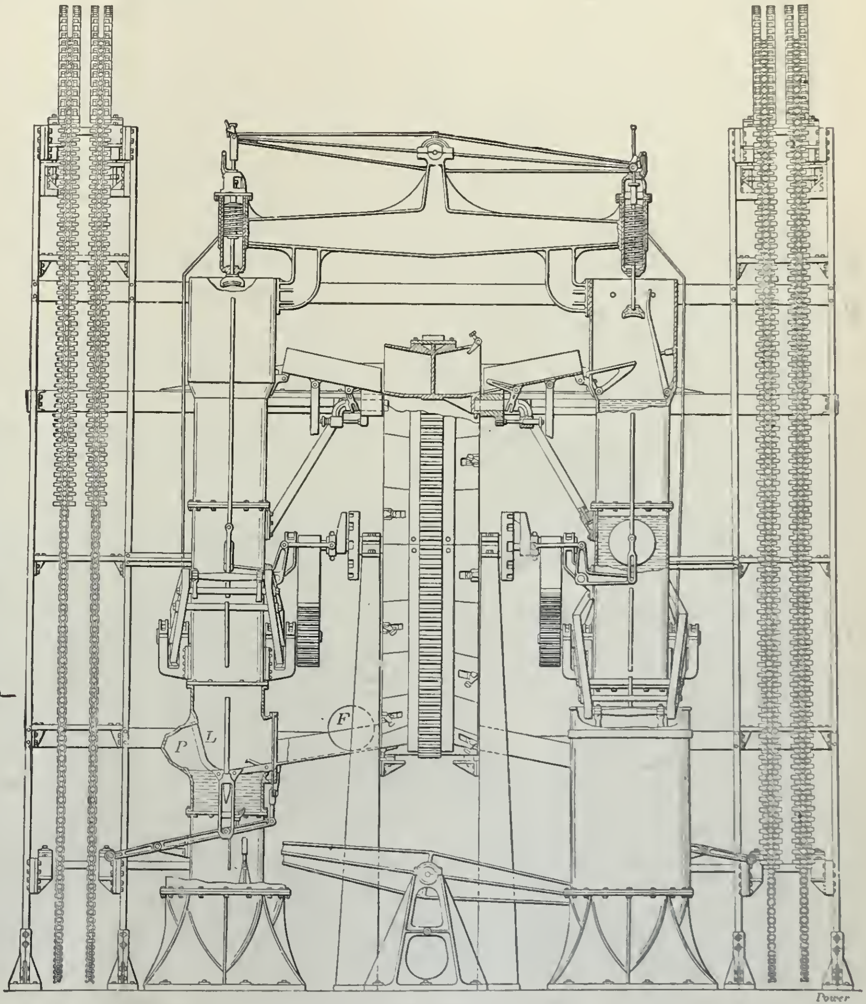


Fig. 2. The Hydredynamo, Front View

Eleven floats are always in the front side of the drum and no floats whatever are in the rear side; this, according to the promoters, *develops a great mechanical torque producing power available for useful work.*

The mechanical construction of the device, which seems somewhat complicated, is of small importance except that there is apparently considerable friction loss. It is the hydraulic operation wherein lies the "nigger in the woodpile."

The supposed power is derived by the weight of the floats acting through a distance equal to the diameter of the drum. At first glance it might appear that no energy is expended in raising these floats and that the energy derived through their fall is pure gain, but a closer study of the problem will show that when the float *F*, Fig. 2, has rolled into the pocket *P* and is held in position by the latch *L*, while the water rises, the volume of water displaced by the float, which is many times heavier than the latter, must be raised a greater distance than if the float were not immersed. This additional distance is equal to the difference in the levels of the water before and after the float has been released, and it requires power to lift this water. It may be said that this power is furnished by the weight of the eleven floats in the drum. True, but the force of these eleven floats acting through one-eleventh of the total fall is equivalent to one float falling the entire distance, or the eleven floats falling the entire distance would have to raise the column of water eleven times. Again, the fall of one column of water will not raise the other column the entire distance and the extra work of raising the water the total distance is at the expense of the force produced by the floats in the drum. The question naturally arises, "What power is left to accomplish external work?" When friction is taken into consideration the balance of power will undoubtedly be found to be a negative quantity. It would seem almost a physical impossibility for the machine to run light, much less than with a load.

There has not yet been any of these machines constructed, all the claims being based upon hypothetical assumptions and drawings, with the promoters content to offer stock for sale under the most alluring pretensions. It is said, however, that one is now being built; but until such time as a practical demonstration is made and the claims substantiated, we shall continue to maintain our present attitude regarding its impracticability.

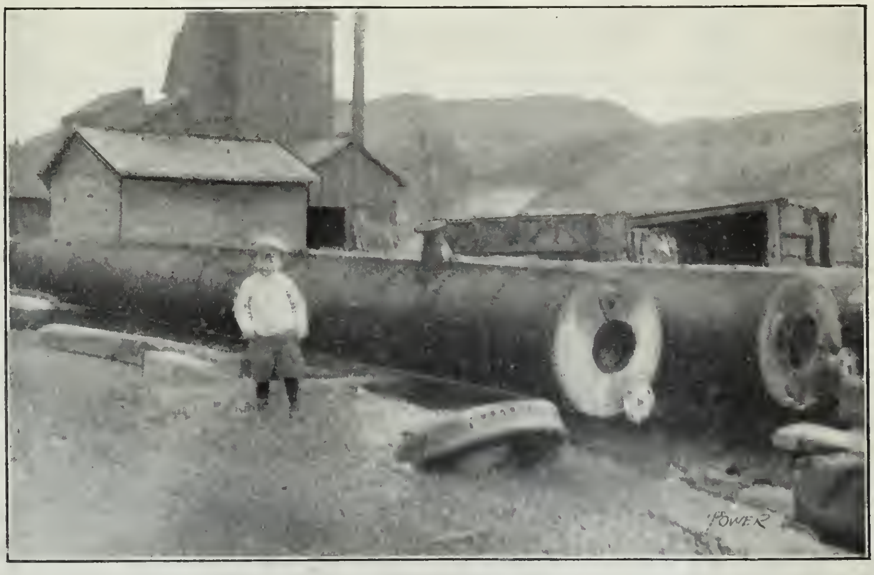
Forty Foot Cylindrical Boilers

By F. Webster Brady

This antiquated type of boiler, made up of many single-riveted sections with cast-iron heads, is still in use at a number of collieries in Pennsylvania, where its simplicity and adaptability to hard usage make its retention particularly desirable.

In the anthracite region of Pennsylvania there are still to be found many of the old type of cylindrical boilers, consisting of a long cylindrical wrought-iron shell with cast-iron heads. Some of these have been in service for many years, but are still fulfilling their purpose.

The illustration shows some of these old boilers, which were removed from a nearby colliery. They are 34 inches inside diameter and about 40 feet long, with the shells made up of eighteen sections of ¼-inch plate, single riveted at both the longitudinal and girth seams. The heads are of cast iron, having a 6-inch flange around the edge which fits into the shell, and are held in place by a single row of ¾-inch rivets. The front heads each have a manhole of regular size, and two flanged openings cast solid with it; the bottom, or feed opening, is 2¼ inches in diameter and the top, or steam connection, is 3 inches in diameter. The safety valve, which was of the ball-and-lever type, was mounted on the casting riveted to the top of the third section, as shown. A front head, separate from the shell, may be seen in the foreground of the illustration, also an extra rear head on the ground at the far end of the shell. The rear head is a solid casting without any openings. It is flanged and fitted to the shell in a manner similar to the front head. A wedge-shaped lug cast on the back head serves to attach a supporting sling, and two other supporting slings are spaced along the shell. There is a 6-inch handhole on top of the fourth section from the rear end.



40-foot Cylindrical Boilers with Cast-iron Heads

Boilers of this type are placed in banks of from two to twelve with a plain brick setting. The space between the shells is choked with bricks which usually become covered with a thick layer of flue dust. There is a continuous grate extending the whole width of the fire room and a single header connects all the steam flanges. The steam pressure usually varies from 50 to 100 pounds, according to the age and condition of the plant.

This type of boiler found favor among the collieries of this section because of its low cost, extreme simplicity and easy operation. Refuse, dirt and unsalable coal are burned under these boilers, and the boiler-room work is performed by the lowest grade of labor. The steam blower, usually consisting of a plain ⅜-inch pipe is in constant use. The ashpit, excepting a small space around the blower nozzle, and the dust chambers become filled with ashes, and it is not uncommon for the flame to burst from the firing doors or most any place around the shell. The metal stacks are about the same diameter as the boilers.

In a plant of this kind the equipment has to withstand much hard usage and abuse. Many of the happenings would end with serious results were it not that the equipment is so simple. These old boilers make good water tanks, stacks, etc., long after their steaming days are over.

Ottawa has a peat bog at her doors, estimated to contain over 5,000,000 tons of fuel, and three others within about 40 miles aggregating over 20,000,000 tons of peat.

Safety Stops for Steam Engines

By W. H. Wakeman

> *A safety stop on a steam engine is of as much importance as a safety valve on a steam boiler. In this article most of the safety devices used in connection with the engine governor are illustrated and their action described.*

A safety stop for a steam engine is any kind of a device that will stop an engine should the governing apparatus get out of order, and, although some types of stop device will shut down an engine when the load is greater than it can carry under existing conditions, they should also be classed with the safety stop.

Fig. 1 shows a simple type of safety stop. An idler travels on the top side of the belt, and, should the belt break or slip off, the idler will fall into a lower position. This disengages the speed pawl, releasing the valve, which drops, shutting off steam from the engine cylinder. The packing in the stuffing box round the valve stem should be kept in ood order, as it is possible to prevent rompt action of this stop should the tuffing-box nut be screwed down too hard on unsuitable packing, or on packing that has outlived its usefulness.

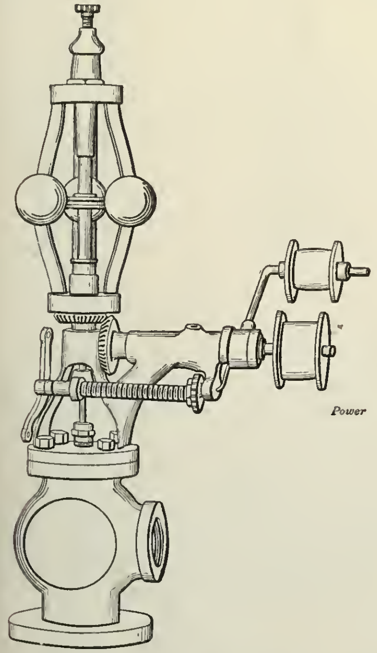


FIG. 1.

The governor shown in Fig. 2 is located on the engine frame about 4 feet from the cylinder, instead of directly over the steam chest. Should the governor belt break, or be thrown from the pulley, steam would be shut off from the cylinder. The device for opening the steam valves is similar to the Corliss gear. Steam is admitted and cut off automatically from zero to about one-half stroke, under normal conditions, and if more load is added the flyballs take a lower position and steam follows the piston the entire stroke, providing the balls and center weight do not fall to their lowest possible position.

If the governor belt should break, or from any other cause the flyballs should move slower than usual, or stop, the center weight will drop as low as possible, which will cause the tripping toes on the cutoff mechanism to prevent the steam valves from opening, hence steam is not admitted to the cylinder, and the engine will stop. When it is time to shut down, a block is inserted, as shown at *A*. While this allows the center weight to go low enough to give steam to the cylinder during full stroke of the piston, it does not allow the safety stop to operate. This makes it possible to handle the engine properly, and not allow it to stop on either center. When starting the engine, and the throttle valve is wide open, the block *A* is removed from the governor.

Another device for blocking a governor is shown in Fig. 3, which, to a certain extent, is automatic in operation. It is shown in running position. The sliding block *B* is held in an out position by a concealed spiral spring. Should the governor belt fail, the flyballs will stop and the center weight will fall to its lowest position. If it is desired to shut the engine down in regular service, close the throttle valve and as the speed slackens, the stop lever *B* is pushed inward against the spiral spring until it strikes the spindle, and held there until the weight and collar *C* rests upon it. When the engine is started and the flyballs gain sufficient speed to raise the center weight, *B* is liberated, the spring throws it outward and the stop is set automatically.

A homemade device for blocking a governor on a Corliss engine is shown in Fig. 4. The sliding sleeve *A* is in a position for an average load on the engine, but if the throttle valve is closed the projecting pin *B* falls until it rests on the thick washer *C*, carried by the stationary pin *D*. This allows steam to follow the piston full stroke, and enables the engineer to stop the engine in any desired position. When the engine is started again and brought up to regular speed, *B* rises from the washer *C* which is then removed from the pin *D*. This permits the collar *B* to drop low enough to prevent the steam valves from hooking on the catch blocks should the governor belt break.

An automatic device for releasing a governor after it has been blocked is shown in Fig. 5. As soon as the flyballs revolve fast enough to raise the center weight, the catch piece *A* turns one-half revolution to the right on the pin *B*. This automatically sets the safety stop ready for an emergency, because *A* will not stay in the position shown unless it is held there.

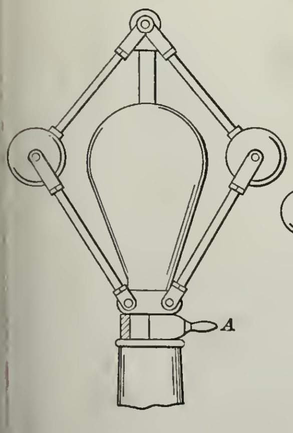


FIG. 2

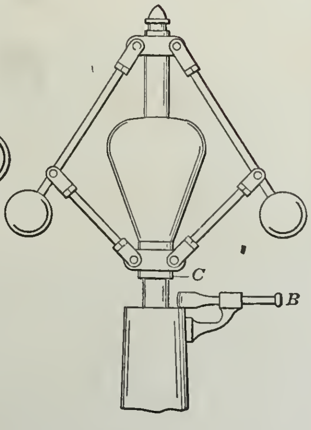


FIG. 3

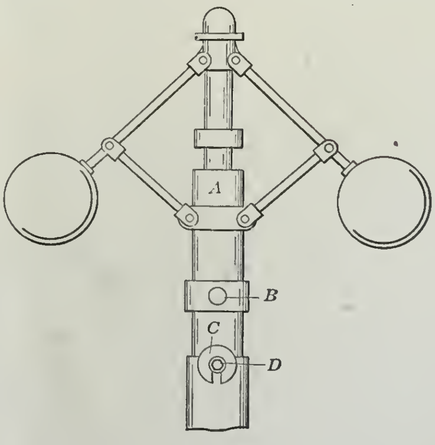


FIG. 4

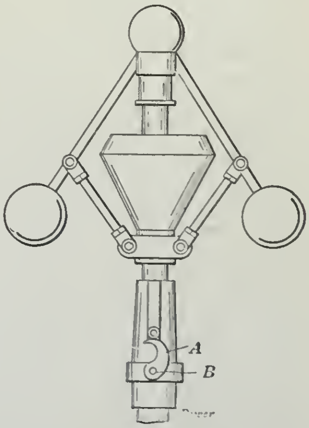


FIG. 5

Fig. 6 shows a safety stop in running position. When the governor is to be blocked while shutting down and starting again, the pin *P* is inserted in the hole *H* and prevents the flyballs from falling to their lowest position.

In case the engine should stop with the pin out of the hole *H*, first close the throttle valve, then lift up the flyballs and put the pin in the hole *H*. Then unhook the reach rod and rock the wristplate to its extreme travel in both directions so that both steam valves will catch into the catch blocks, and drop the reach-rod hook into place. Steam can then be turned on and the engine started.

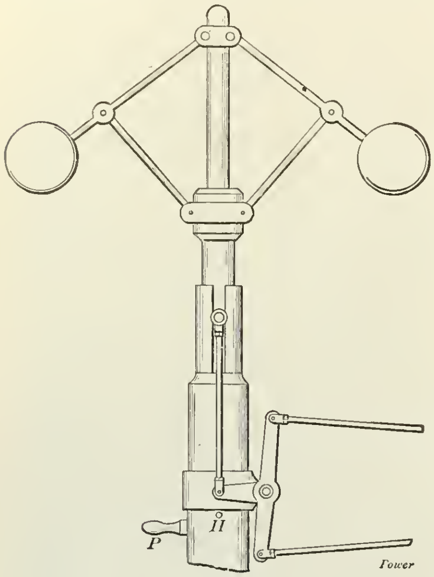


Fig. 6

The governor shown in Fig. 7 is in its running position with a heavy load on the engine. If more load is added the projecting pin *A* will rest on the movable collar *B* causing steam to follow the piston full stroke. If the collar *B* were moved until the vertical slot *C* is directly under pin *A*, and more load is put on the engine, the centerweight will fall to its lowest position, causing steam to be shut off from the cylinder.

A governor fitted with a centerweight appears to be much more sensitive than if the centerweight were omitted, hence it ought to operate a stop motion almost instantly. This form of governor is more sensitive while in full operation than the style shown in Fig. 6, but the former is always operated at a much higher speed, and it will run longer after the belt is removed. The effect of this action on the flyballs is partially offset by the centerweight. In either case great care should be taken to have the dashpot, which usually forms a part of such governors, properly adjusted, as otherwise it may hold the flyballs up for a few seconds, when time is very valuable, and thus cause much damage.

This warning may be needed in cases in which such need is least suspected, for the following reason. An engine may nearly always carry a light load, causing the flyballs to move on a high plane, and allowing the centerweight to travel through a limited vertical space. Long service under these conditions results in a very easy movement of these parts through their ordinary limits, but when moving below these limits they may not move so easily. This difference would only be noticed when the engine is started, and not then unless the engineer is especially careful to observe the details of operation, for the only difference is that the flyballs must be given a higher speed than if there were less friction. Even this may not exceed the highest speed attained under common conditions. After steam is shut off by closing the throttle valve, and the centerweight is moving downward, it may show irregular motion, but as this cannot affect any other part, it may not attract attention.

Fig. 8 shows two views of a governor equipped with a stop that is automatic in both directions. It is shown in running position. Steam from the bonnet of the throttle valve is carried through the small pipe *A* into the disk *B*, where it operates on a device similar to the Bourdon spring in a steam gage, which throws *C* out of a vertical position, as shown. Therefore, the bolt *D* cannot rest upon it, if the centerweight is falling to its lowest position should the governor belt break or come off the pulley. Consequently the stop operates and steam is shut off from the cylinder.

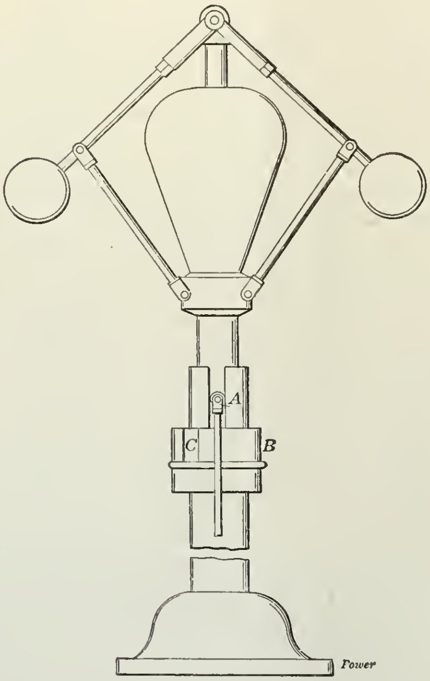


Fig. 7

When steam is shut off by closing the throttle valve, pressure falls in the pipe *A*, which is piped to the steam chest or to the throttle valve under the disk; hence, the spring inside of *B* carries *C* into an upright position just in time for *D* to rest upon it. When the throttle valve is opened again pressure begins to rise in the pipe *A*, but it is not sufficient to throw the lever *C* over until *D* is lifted from it by the action of the flyballs, thus requiring no attention from the engineer. *F* is a dashpot which prevents the centerweight from moving too rapidly.

The small pipe *A* carries pressure to *B* when the throttle valve is open, but when closed, pressure is shut off from the stop.

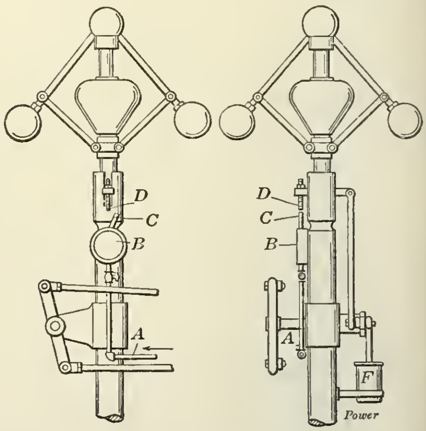


Fig. 8

A peculiar governor is illustrated in Fig. 9. Centrifugal force, caused by high speed, throws the governor balls outward, and the bell cranks, to which they are fastened, raise the sliding sleeve which operates the long bell crank pivoted at the left-hand side of the governor. This gives motion to the horizontal rod *A*, and as it is connected to the tripping devices on the valve gear, it gives a long or a short point of cutoff according to the load carried. The cam *C* is shown in mid-position. When the engine is about to be shut down the long arm *C* of this cam

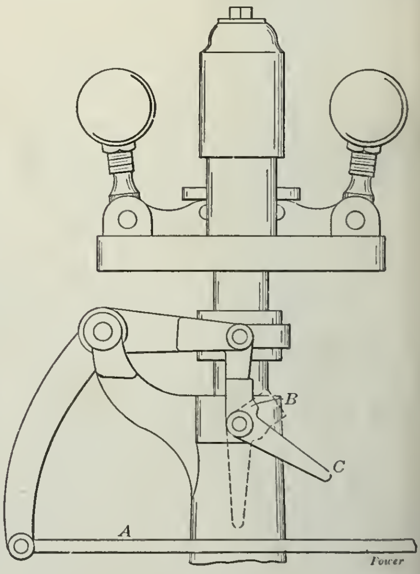


Fig. 9

s raised to a horizontal position by hand, ience the short arm *B* stands vertically ind receives the projecting part of the ;overnor. When the engine is started igain and centrifugal force throws the lyballs outward, and the cam, being lib-rated, drops to the position shown by ie dotted lines.

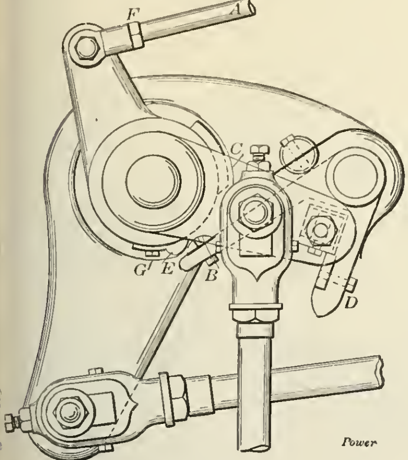


FIG. 10

Every engine stop ought to be given a ractical test at frequent intervals (say nce each week), to prove that it is in orking order. For illustration, take a orliss engine with a flyball governor.

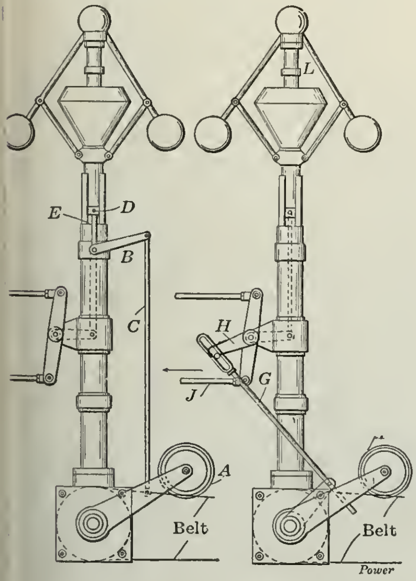


FIG. 11 FIG. 12

ose the throttle valve until there is st enough steam admitted to maintain ll speed. Suddenly force the flyballs wn to their lowest position and keep em there for a few seconds by holding convenient part of the governor. The gine will run faster for two or three volutions, and then gradually stop, ovided the adjustments are properly ide. If not, steam may be shut off om one end of the cylinder, and ad-tted to the other, but this will not give excessive speed. If it is not shut off at both ends, make an adjustment that will give the desired result, but do not change the comparative point of cutoff, for if they are even they should not be changed, and if the condition of the valve gear in this respect is not known, an intelligent change cannot be made. It is usually possible to set the stop motion without changing the point of cutoff, but on the other hand, there are ways whereby both can be changed together, and it is plain that these ought to be avoided.

Fig. 10 illustrates this point, and shows the gear for opening and closing a Corliss steam valve. The reach rod *A* is adjusted so that when *B* is carried upward by the lifting arm, it will strike the knock-off cam *C* and the latch *D* will be unhooked and the dashpot will close the valve. If the flyballs of the governor of this engine should fall to a lower plane than at present, the reach rod *A* would be carried to the left, and the safety cam *E* would move toward the right hand, striking *B* and preventing *D* from hooking on. Consequently the valve would not open and no steam could pass into the cylinder. It is possible for the valve gear to wear until *E* is no longer in proper position to prevent the valve from opening. A careless or thoughtless engineer will loosen the jam nut *F* and lengthen the reach rod until *E* will perform the duty for which it was designed. This action causes *C* to be moved from its proper position until it no longer disengages *D* at the right moment, resulting in an uneven cutoff for the engine, assuming that it was even before. The correct way is to let *F* remain as it is, and loosen nut *G* and slide *E* into proper adjustment and then screw *G* tightly down upon it.

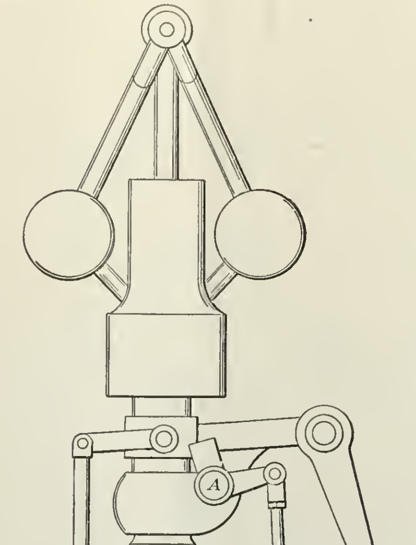


FIG. 13

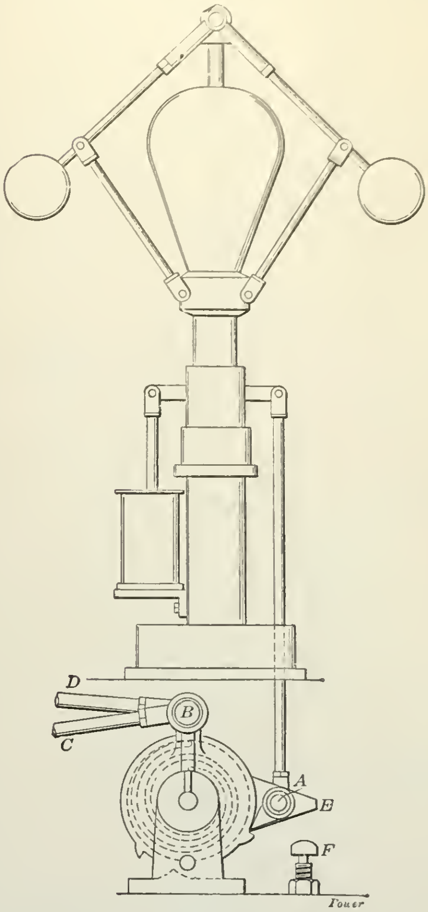


FIG. 14

An automatic stop which needs no attention from the engineer when starting or stopping his engine is shown in Fig. 11. The pulley *A* rides on the governor belt. The bell-crank lever *B* is kept in the position shown, by means of the long vertical rod *C*, as long as the governor belt remains in place. The flyballs and centerweight are in position to give the longest point of cutoff to the steam valves, as the projecting pin *D* rests upon *E*. When the load is reduced, *D* rises, but *E* remains in position. If the governor belt breaks, *A* falls, *B* is also lowered and *E* is carried toward the right, therefore *D* goes lower than shown, and steam is shut off because the steam valves do not hook on and the engine stops. An important feature of this device is that if an overload is put upon the engine it will not be shut down. On the other hand, if the governor ceases to revolve for lack of lubrication or any other cause, and the belt remains on the pulleys, steam will be admitted full stroke to the cylinder, causing excessive increase of speed.

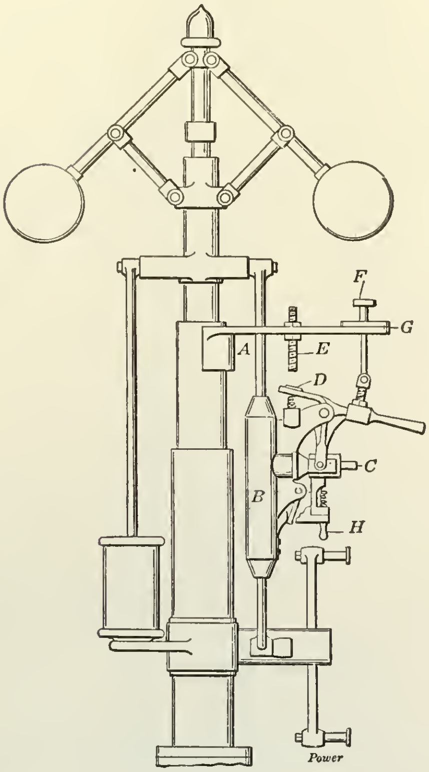


FIG. 15

Another stop is shown in Fig. 12. The governor is shown with a medium load on the engine. The pulley *F* rides on the belt and during ordinary service this pulley does not change its position, consequently the slotted rod *G* is practically stationary. If more load is added the centerweight is lowered, also the right-hand end of the lever *H*, the opposite end of which moves upward in the slot. The lower reach rod *J* moves as indicated by the arrow. If the load becomes lighter the centerweight rises and the upper reach rod is carried toward the left, shortening the cutoff. The same action takes place when the governor belt breaks, because *F* falls, and the slotted rod *G* is drawn down until the end strikes the pin in the bent lever *H*, the effect of which is to force the centerweight upward, shortening the point of cutoff until it is reduced to zero and the engine stops. The collar *L* must be set high enough to allow the centerweight to rise as described, but no higher. As the flyballs rise to operate the stop motion, the pulley *F* and its connecting levers must be heavy enough to overbalance the centerweight, flyballs, etc. With this gear the knock-off cams are utilized for safety cams, but only one set is required for both operations.

Fig. 13 represents another stop depending on an idler for its operation. As long as the belt supports this idler the engine cannot be shut down by an overload, but when the belt fails the cam *A* is reversed, and is no longer in a position to support the centerweight of the governor, consequently when it falls as low as possible, the valves remain closed and the engine stops. This device acts quickly because no load is put upon the idler and all of the mechanism connected to the idler tends to carry it downward when the belt breaks.

A flyball and centerweight governor attached to an ingenious device which operates the cutoff mechanism, also the stop, is shown in Fig. 14. The main parts of this device consists of two disks designated as rear and front, both of which are hung at the center. The rear disk *A* is attached directly to the governor by a vertical rod. The front disk *B* operates the two reach rods. Under normal conditions these disks move together, and transmit the motion of the governor to the cutoff mechanism.

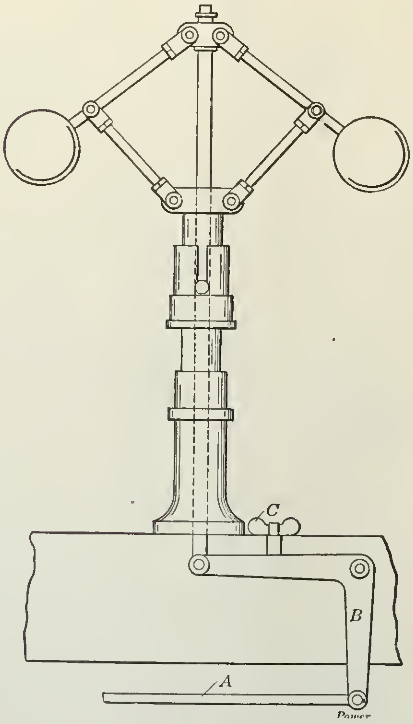


FIG. 16

The flyballs are shown in position for a medium load on the engine. If the load is reduced the balls and centerweight move upward, rotating the disks from right to left. This demonstrates that if *C* and *D* are carried to the left the cutoff is shortened, therefore if they are carried far enough the steam valves will not open. Increasing the load causes the centerweight to fall, carrying *A* downward until the pawl *E* rests on the adjustable screw *F*, causing *E* to release the fror disk which is quickly thrown to the lei by a strong coil spring. This action roll the cutoff cams until they prevent th steam valves from opening. But one se of cams is used to designate the point c cutoff in regular service, and to prever the valves from opening in case of acc dent to the governor, without raising th centerweight.

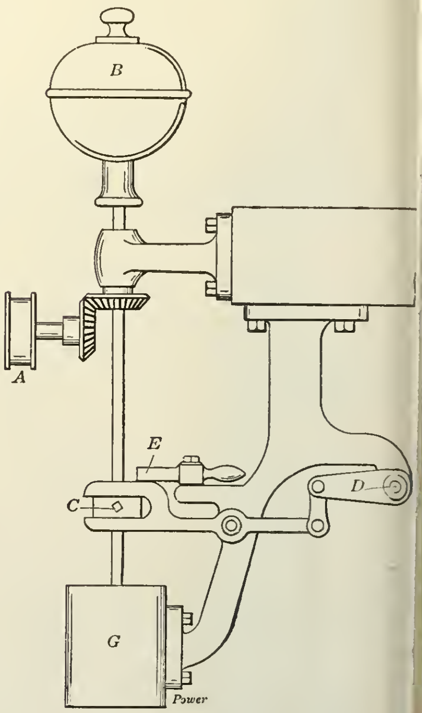


FIG. 17

To prevent the stop motion from op erating when the engine is shut dow under normal conditions, a pin is inserte into both disks, thus locking them t gether. When the engine is started aga this pin is thrown out automatically, th making it impossible for the engineer leave it in place, and render the st useless.

A more complicated form of stop shown in Fig. 15. The rod *A* correspon to the ordinary side rod on a governo except that its length is variable, and shorten this length is the sole object the device, because this section thro the ordinary reach rods to their extren position and steam cannot be admitted the cylinder. The enlarged part of t rod *B* is hollow and fitted with a stro spiral spring which tends to shorten t rod, but the pin *C* extends into this holl part and prevents the operation of t spring.

Suppose that the valve gear is deranged and the engine starts to "run away." The flyballs rise carrying the spring-supported lever *D* upward until it strikes the stationary stop *E*. As *D* is depressed by this action, *C* is withdrawn from *B*, and the inclosed spiral spring draws the reach-rod lever upward, shortening the point of cut-off to zero (without carrying the flyballs higher) and as steam cannot enter the cylinder, the engine stops.

During ordinary service the rod *F* slides freely in the plate without interfering with the action of the governor in regulating the engine speed. However, if the belt runs off, the flyballs drop to a lower plane, and the movable part of the device is lowered until *F* rests upon *G*, thus pulling *D* down as before, drawing *C* outward and allowing the concealed spring in *B* to draw the reach-rod lever up until the point of cutoff becomes shorter, and finally is reduced to zero, stopping the engine. When preparing to shut down the engine in regular service, the stop is rendered inoperative by adjusting the plug *H*, thus enabling the engineer to stop and start at pleasure. When the engine is started again the flyballs rise and liberate *D*, which action throws *H* back automatically into the position shown.

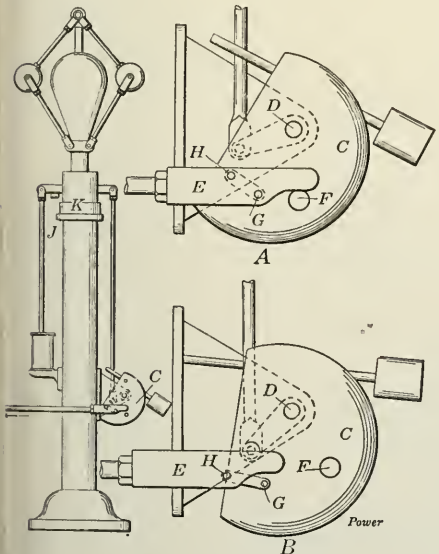


FIG. 18

A very simple device for operating a stop is shown in Fig. 16. As the flyballs drop the rod *A* is carried toward the right by means of the bell-crank lever *B*, and this prevents the steam valves from opening. A slot is provided in the frame for the thumb nut *C* allowing it to move freely vertically as long as it remains in the position shown, but if it is turned at right angles to this position it cannot pass through the slot. While turning the thumb screw prevents the device from operating in case the governor belt breaks, it is necessary to do so in order to operate the valves while starting and shutting down the engine. If the thumb nut is not set in the running position the device is no longer a safety attachment.

Fig. 17 illustrates a governor which is set low on the engine frame. The pulley *A* is driven from the crank shaft by a belt and bevel gears transmit motion to the vertical shaft on which is mounted a pair of flyballs inclosed in a iron case *B*. If this belt breaks, the flyballs come to rest, and the vertical shaft is drawn upward, the revolving collar *C* moving with it, thus turning the side shaft *D* through part of a revolution. This action prevents the steam valves from opening.

This stop will not operate while the hand lever *E* is in the position shown, because it prevents *C* from rising, but when given one-quarter revolution *C* is at liberty to rise. A dashpot *G* prevents *C* from rising and falling rapidly.

Another type of stop and its operating mechanism is shown in Fig. 18. Two views of the tripping device are shown at *A* and *B*; *A* shows the position taken with a light load on the engine. The half circular plate *C* is carried by a short shaft. The reach rod *E* rests upon the pin *F* and is held there by gravity, also by action of the link. The pin *G* is set into the reach rod and *H* is a similar pin set into the plate *C*. Therefore the stress upon the rod in the direction of the arrow will hold *E* more firmly on the pin *H*.

If the governor belt breaks, *J* is lowered into the slot in the movable collar *K*, and the action is further illustrated by *B*. The plate *C* has turned toward the left until the pin *H* is lower than pin *F*, causing the link to reverse its position, thus moving *E* toward the left twice the length of the link, which is sufficient to operate the stop.

A stop operated by electricity is shown in Fig. 19, used in connection with a Corliss governor, but it is suitable for several other types of valve gear. Steam at boiler pressure is admitted to the angle valve *A* which is closed under normal conditions. When the crank-shaft speed exceeds the limit for which this stop is set, a special governing device closes the electric circuit *B* and the armature *C* releases the lever *D*. This action allows steam to open *A*, when it passes through the check valve *E* into the vertical cylinder *F*, causing the piston *G* to rise and the rod *H* to lift *J* until the collar *K* prevents it from going higher, thus raising the flyballs to their highest position, and rolling the knock-off cams on the valve gear so far around that they prevent the steam valves from opening.

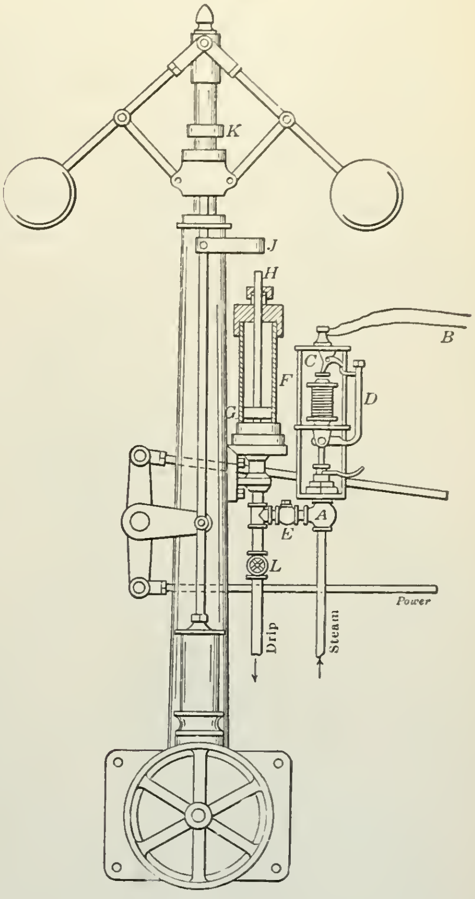


FIG. 19

When *D* is replaced, *A* is closed, and by opening the drip *L* all pressure is released from *F*, thus allowing the piston *G* to resume the position shown, when the stop motion is ready for service. Then the throttle valve is closed, the steam valves hooked up, and the engine is ready to start.

It has been stated that the exhaust from a gas engine will, if passed through suitable exhaust-gas boiler, evaporate from two to two and one-half pounds of steam per brake horsepower-hour. Where the steam can be used this marks a substantial economy, and one would expect such boilers to be popular. As a matter of fact, very few are in use. One reason is, undoubtedly, that these boilers are subject to serious corrosion in many cases, particularly where, as frequently occurs, cold water is used for feeding the boiler. If, in addition, the boiler is shut down at week ends, it may be very cold when starting and will sweat. The exhaust gases being corrosive—especially if producer gas is used—a dilute acid is formed which has serious effects on the plates. A method of partially overcoming this corrosion, which is not often used, consists in feeding the boiler with water from the jackets. This water is, perhaps, at 120 degrees, and its temperature prevents sweating. On the other hand, there is no hot water when the engine first starts after a prolonged stoppage, and unless the boiler can wait half an hour or so some artificial means of heating the boiler should then be adopted.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Modern Governing Mechanisms

By Warren H. Miller

Of the various governing systems used by European and American builders of gas engines the means for varying the explosion pressure, and consequently the mean effective pressure, may be classified as follows:

1. Constant volume of charge and variable ratio of gas to air.
2. Constant ratio of gas to air and variable volume of charge.
3. Variable ratio of gas to air and variable volume of charge.

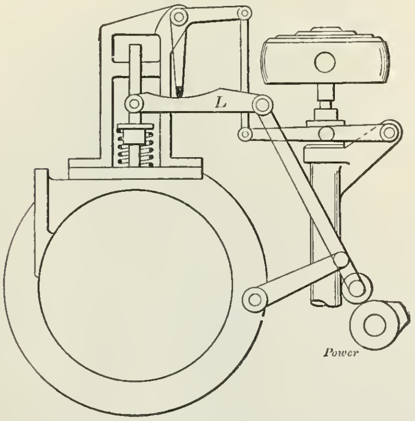


Fig. 1. Otto-Deutz Movable-fulcrum Gear

For small engines means for simply varying the stroke of the inlet valve will answer for regulation by varying the quantity of a constant-ratio mixture, and this is used in the original Deutz gears illustrated in Fig. 1. The governor alters the position of the fulcrum of the inlet-valve lever *L* and thereby changes the stroke of the inlet valve. In large engines, however, it is usually not practical to depend on the stroke of the inlet valve alone to determine the quantity of mixture admitted, without imposing too much work on the governor and robbing it of its sensitiveness, or else incurring the liability of the valve to reopen under the suction when only light charges are to be admitted. For this reason nearly all the large governor mechanisms include a mixing valve through which the governor controls the mixture, in addition to the main inlet valve for admitting the charge to the cylinder and keeping it there. Fig. 2 illustrates the Deutz mechanism used on large engines, the engraving being made from a drawing of the 2000-horsepower size. The governing is by throttling and the volume admitted is controlled, as in the small engine, by means of the movable fulcrum; shifting the fulcrum *f* alters the stroke of a pair of balanced gas and air valves, and the fulcrum at the left of the air-valve spindle is also shifted, still farther altering the opening of the gas valve. The ratio of gas and air can be altered to suit the quality of gas by turning a handwheel which moves a sliding cage on the inlet-valve stem.

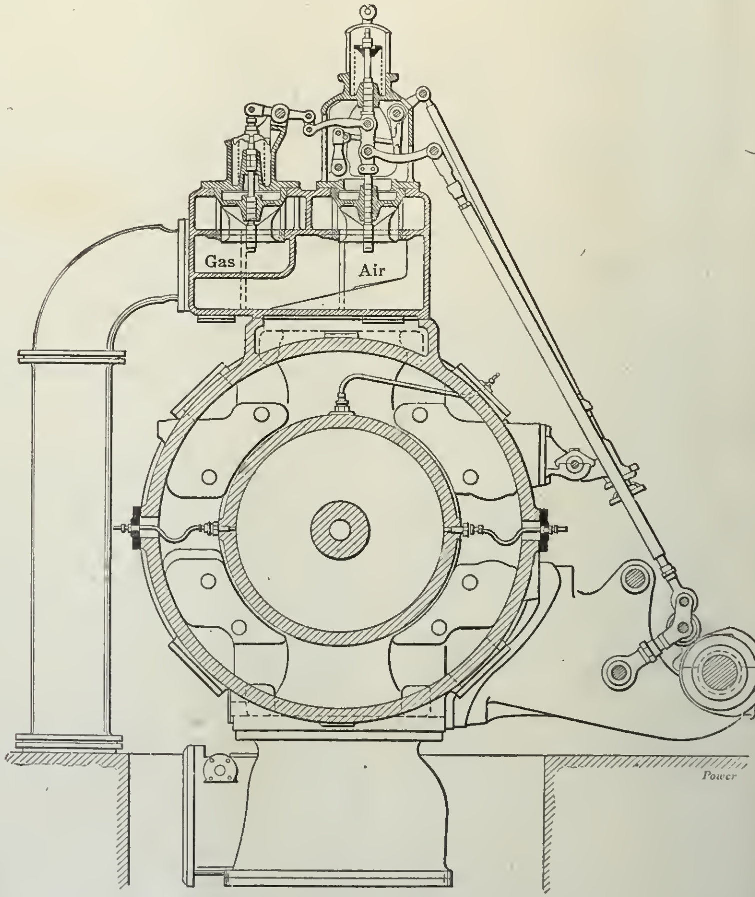


Fig. 2. Deutz Valve Gear for Large Engines

Fig. 3 shows the form of shifting fulcrum gear devised by the Nürnberg company for varying the opening of the inlet valve. The governor shifts the roller *R* and thereby varies the "leverage" of both the rocker arm *A* and the valve lever *L*.

Fig. 4 illustrates the cut-off governor gear of the Société Alsaciénne. It is somewhat similar to some of the American mechanisms in that it embodies a ported-cage mixing valve. The inlet valve has a constant lift. The mixing valve, shown in section in Fig. 5, is a sliding cylinder with ports registering with ports in the housing which open into the gas and air mains. It is divided circumferentially into air and gas ports in the correct ratio for the quality of gas used. This valve remains normally down, shutting the ports, as shown in Fig. 4. With each stroke of the eccentric rod operating the inlet valve, the valve is lifted until the ports register with those in the housing and latched in that position, allowing gas and air to flow through to the cylinder until the valve is released by a trip gear under the control of the governor; when the latch is tripped, the valv drops, shutting the ports.

Fig. 6 shows the mixing-valve gea used on the large size of the Nürnber engine, which is probably more famili to American engineers than any oth

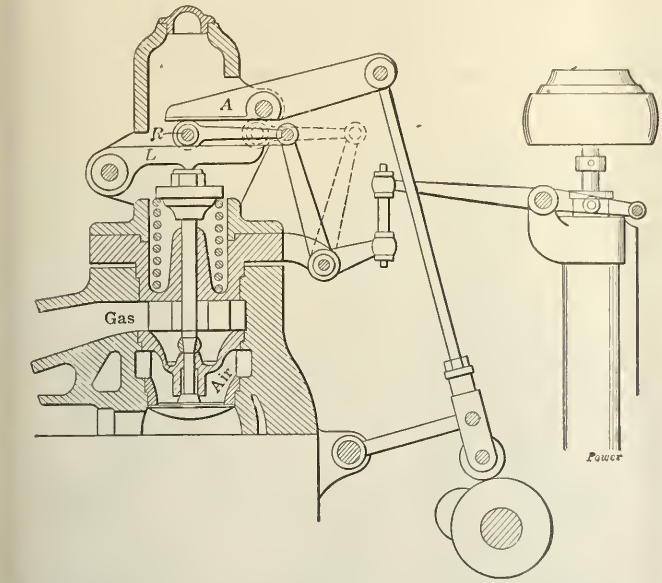


Fig. 3. Schmitz Movable-fulcrum Gear

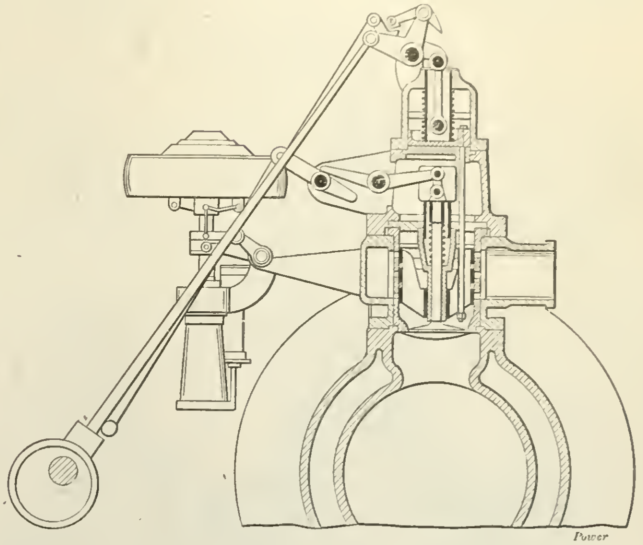


Fig. 4. Alsacienne Variable Cutoff Gear

Continental engine. Air is admitted throughout the suction stroke and the gas is admitted toward the end of the stroke, both being shut off simultaneously at the end of the stroke. The quantity of gas admitted is varied by opening the gas valve earlier or later in the stroke; regulation is effected, therefore, by varying the quality of the mixture, the volume being constant. Fig. 6 shows only the mechanism that operates the mixing valve, of which there is one on each cylinder, placed midway between the two inlet valves, which have constant lift and are opened by the usual cam-rod-lever combination. A bifurcated arm *A* is pivoted at its fork ends to the outer housing and linked at the other end to one end of the bell crank *R* which is controlled by the overnor. The valve-operating lever *B* is pivoted at its inner end to the cylindrical member *C*, which slides within another cylinder *E*, attached to the stem *S* of the valve; the lever *B* has a sort of rolling fulcrum on the arm *A*, and when the governor lowers the outer end of this arm, the "heel" of the lever *B* strikes the face of the arm *A* later in the stroke and therefore opens the valve later and reduces the explosive mixture. The lever *B* opens the valve by lifting the sleeve *E* through the medium of a dog *D*, pivoted to the plunger *C*, and this dog is tripped at the end of the suction stroke, no matter at what point the valve is opened. When the dog is tripped the valve is closed by the helical springs in the bonnet.

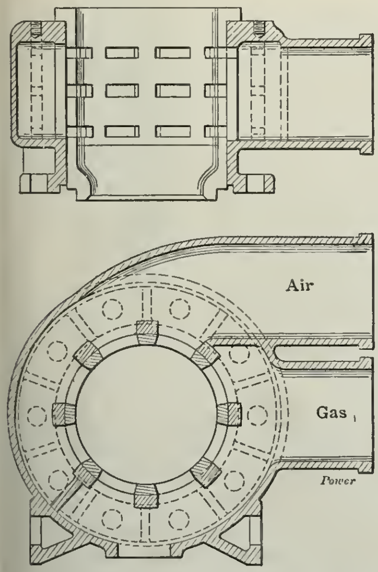


Fig. 5. Alsacienne Mixing Valve

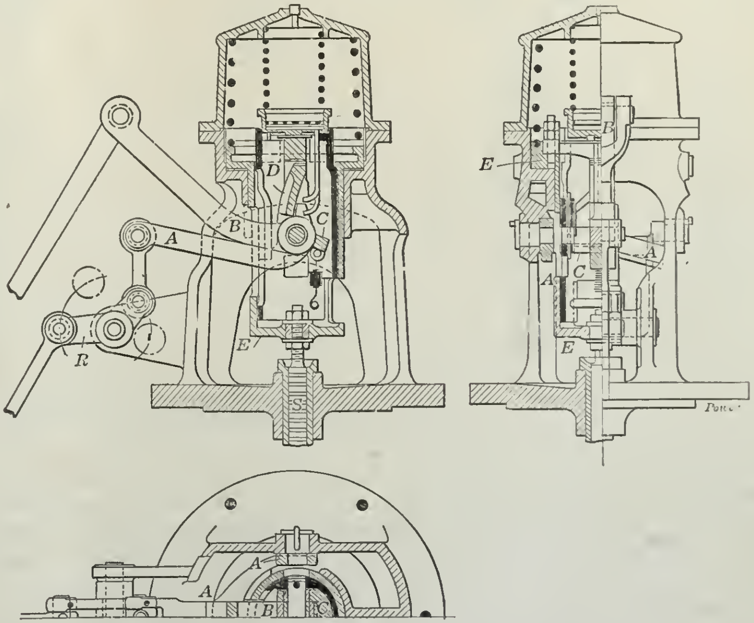


Fig. 6. Nurnberg Variable Cutoff Gear

The governor gear of the John Cockerill company, of Seraing, Belgium, also belongs to the variable-quality constant-volume class, but the mechanism is so designed that it can be used for either quantity or quality regulation. Fig. 7 shows longitudinal and transverse sections of the Cockerill engine cylinder and the principal features of the valve gear. The inlet valve has constant stroke and a sliding air valve moves with it, so as to admit air synchronously with the opening of the inlet valve. The gas valve is mounted above the air valve on a concentric sleeve, and its actuating lever *L* is normally latched at the outer end by the dog *D*. When the push rod *R* is forced upward by the inlet cam it opens the inlet valve and at the same time compresses a spring in the housing *S* which is attached to the lever *L*. This would open the gas valve but for the latch *D*, and this latch is tripped by the governor sooner or later in the stroke, allowing the spring at *S* to open the valve. The valve is closed at the end of the stroke by the downward motion of the housing *S*, due to the fall of the push rod *R* when released by the main cam.

The tripping rod *T* is moved back and forth by a small eccentric on the main shaft, and its engagement with the dog *D* is timed by the governor through a mechanism which shifts the pivot of the intermediate rocker arm.

By extending the hollow spindle of the gas valve downward, unpinning the air valve from the inlet-valve stem and attaching it to the gas-valve spindle, the gear can be changed to quantity regulation.

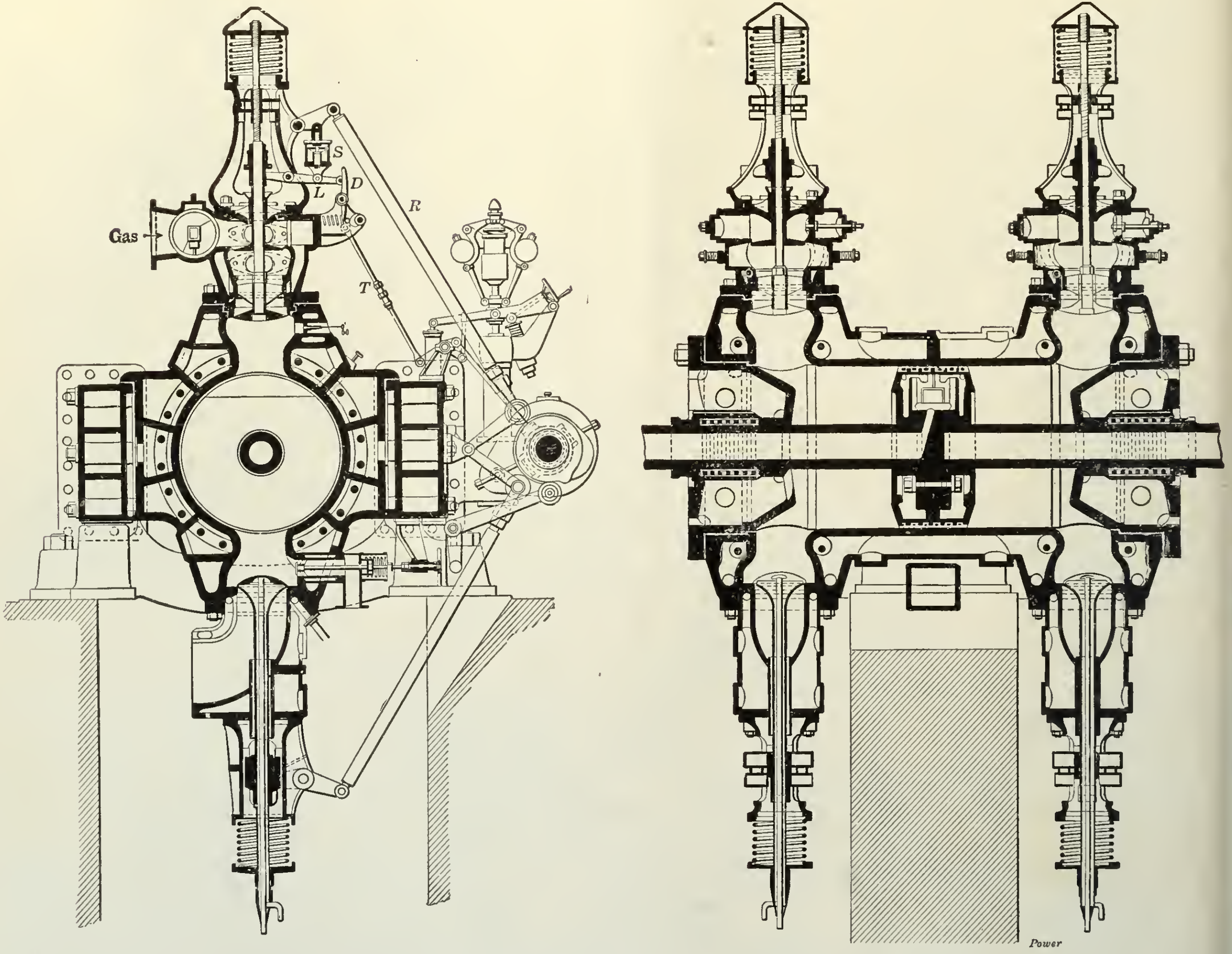


Fig. 7. Valve Mechanism of the Cockerill Engine

A very interesting English gear operating on the variable-quality constant-volume plan is that of Crossley Brothers, shown in Fig. 8. The usual cam-rod-lever system opens the inlet valve against a spring. The nut *M* on the inlet-valve stem tends to carry the gas valve down with the inlet valve by compressing the spring *K*. The gas-valve disk *G* carries an upward cylindrical extension *H* which terminates in a vacuum piston *P* and incloses the spring *K*. A protected opening *O* allows air to pass in and out of the space below the piston *P*, and a check valve *N* prevents the entrance of air above the piston but allows the escape of any that may be there. A "bleeder" valve *J*, controlled by the governor, allows more or less air to enter the vacuum cylinder and vary the upward pull on the piston *P*. When the spring *K* is compressed by the downward movement of the inlet valve, it presses the gas valve open more or less, according to the amount of air admitted by the valve *J* and the resulting degree of vacuum that a given downward movement of the piston *P* can form above it.

As all of the American gears excepting that of the Bethlehem Steel Company's engine have been fully described in Power, there is no need to discuss any but the Bethlehem gear in this article. Fig. 9 is a sectional elevation of the mixing-valve cage and shows also the inlet and mixing valves and the immediate operating mechanism. In this gear the ported cylinder attached to the inlet-valve stem which was first used in this country on the Westinghouse engine has been applied, but the ports are cut diagonally. This makes it possible for the governor to control both the area of the port openings and the length of time that they are open, without using trips or cutoffs. The sleeve *S* is twisted about the vertical axis by the governor rod, and the relative angular positions of its ports and those in the mixing-valve plunger are thereby varied. The mixing-valve plunger is attached to the upper end of the inlet-valve stem and moves up and down with invariable stroke, under the pressure of the cam *C* and the usual seating spring, which is not shown here. At full load the ports in the housing sleeve register completely with those in the plunger, and since the total time that they connect is one-half of that for continuously-registering ports, the area of the ports must be twice as great to pass the same amounts of gas and air. When the load lightens, the governor shifts the sleeve *S* around so that at each stroke the ports in the sleeve and those in the plunger overlap less and less, until at very light loads they register only during the latter part of the stroke. As the ports are very wide and slant at a considerable angle, the governor has a very long "selective" range, the total movement in an 800-horsepower engine being about 3 inches on the surface of the sleeve and 7 inches at the in on the outer end of the arm.

This gear is the latest one that has een developed in American practice, and oth this and the Westinghouse gears onstitute an advance over English gears p to the present time in that they pro- de a variable cutoff without the use of ny complicated trip arrangements or ultiplicity of levers, links, cams, etc.

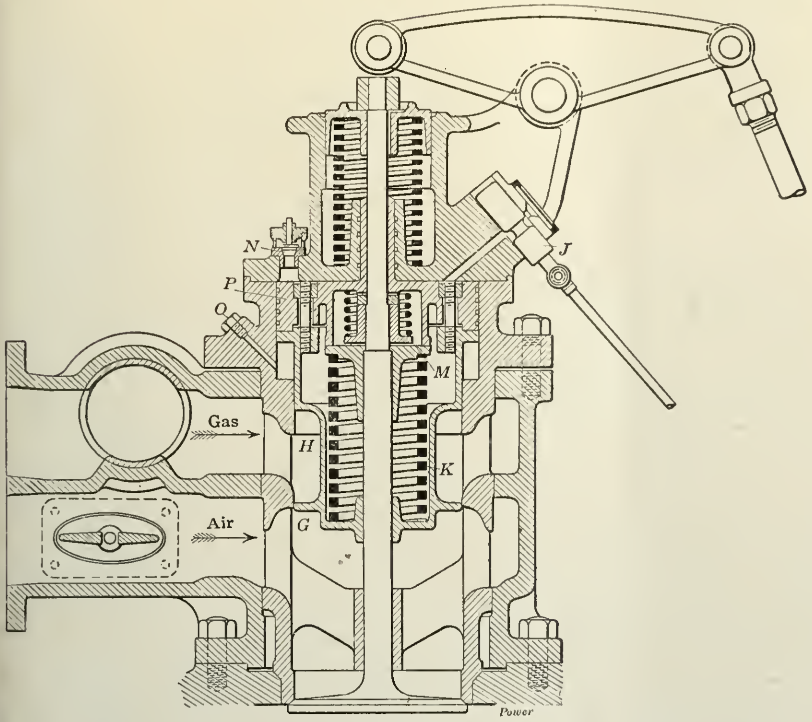


Fig. 8. Crossley Regulating Mechanism for Large Engines

ir William White's View of the Oil Driven War Ship

A recent issue of the *New York Herald* ntained what purports to be an inter- ew with Sir William H. White, who dur- g the period from 1895 to 1902 was rector of naval construction and as- stant controller of the British navy. ne alleged interview attributes to Sir Villiam some statements which must ive been either garbled beyond recogni- on or else supplied entirely from the porter's imagination. Most of the re- rted statements, however, have a rather thentic ring. Following is the com- ete article published by the *Herald*:

"Yes," said Sir William H. White, "we ive heard a good deal lately from riters in the press—not the scientific ess—about the wonderful advance that coming in the immediate future from e use of internal-combustion engineer- g in ship propulsion. I do wish that ese gentlemen who spread such stories impending violent changes in existing nditions would read the reports of the Gaseous Explosions Committee, which has investigated the subject carefully, and learn what the real difficulties are in the problem involved. It is really shocking, when one thinks of it, to read some of their maunderings.

"But perhaps all this piling of rumor upon rumor is accounted for by the fact that the 'silly season' is at hand. It has taken the place of stories about the gigantic gooseberry, of which we have not heard this year, or of the sudden and repeated appearances of the sea serpent, another old friend whose head has not been raised. We have had the funnelless battle ship rushing about again. It always will be so and does not matter much, I suppose, only it is to be regretted that when such a mass of scientific and experimental information is available for anyone who will take the trouble to read, these sensational and absolutely incorrect views of the present situation regarding internal combustion should be put before the public.

"I admit that if it were possible to concentrate the great amount of power now provided by huge engines which is required for the propulsion of a battleship into smaller engines such as one might imagine a motor-driven engine [!] would be, we should be getting to a position of tremendous possibility, but we are far from that position yet.

"Now, consider that point along with the fact that at the present time the most powerful petrol motors in serviceable use are from 100 to 120 horsepower. [Diesel engines of 1000 horsepower are built in Germany.—Ed.] In those motors the amount of energy wasted is very considerable. [!] What would be the amount of waste if you were dealing with motors of 30,000 horsepower, and if you had the same proportion of waste? Where would you be? Work and experimentation are proceeding on scientific lines, and it is from that that our friends the physicists and chemists will be able to help us so much by showing the direction upon which such investigations may well be conducted.

"But the public should not be so deceived as it has been by such nonsense as the probabilities of getting in the immediate future a success which we hope and believe may ultimately be achieved, but in order to obtain which there must be much more experimenting, and in which there must be the most cautious advance.

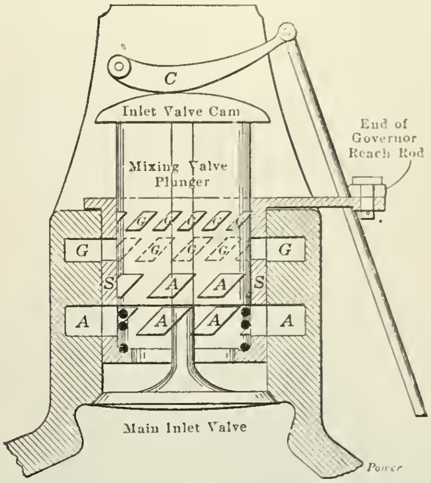


Fig. 9. Bethlehem Mixing and Inlet Valve and Cage

"Of course, the report that the government was about to construct a motor-driven battleship represented an advance the importance of which, had the statement been correct, could not possibly have been overrated. But all this talk of the immediate running of petrol-driven battleships is sheer nonsense."

Another Battleship Prophecy

According to cable despatches printed in the daily newspapers, German designers are at work on a small type of battleship which is expected to put the great "Dreadnoughts" into the obsolete class. The new vessels will be lightly armored and very speedy. In general design they will be similar to the old monitor. Their length will be 280 feet, breadth 46 feet and draft 19⅔ feet, while the hull will extend 59 inches above the water line.

The armaments will consist of two 16.8-inch guns, each capable of throwing a projectile of 5940 pounds from one great turret amidship. Four explosive gas motors, each of 6000 horsepower, will give the vessels a normal speed of 19 knots an hour and *under forced draft* [?] 27 knots. Ordinarily heavy plate will be placed on the decks, turrets and sides of the new vessel.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Transformer and Switch Insulating Oil

By J. Vincent Hunter

There is hardly an electric-power station in the country that does not contain apparatus which depends on oil for some part of its insulation, and the investment involved in this apparatus is too great to permit the usual vague or limited knowledge of the proper care and treatment of this very important portion of a plant's material equipment. Oil is used for insulating and cooling of transformers, for interrupting the arc at the contacts of oil switches and circuit-breakers and for a number of minor purposes, and the same grade of oil can usually be employed with almost equal success in every case, provided proper care is taken in the selection of the oil.

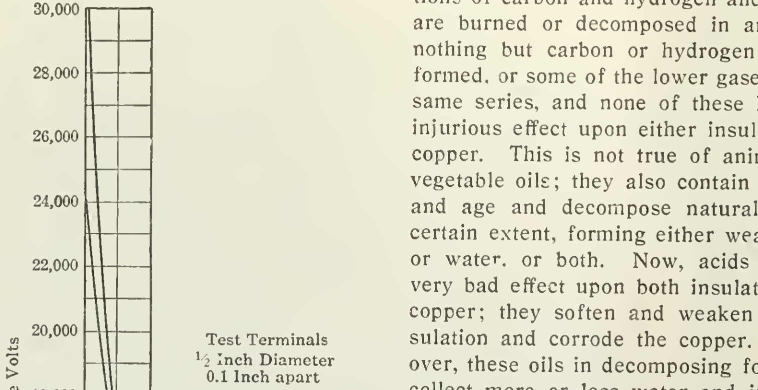


Chart Showing Effect of Moisture Upon Insulating Quality of Oil

The oil used for insulating purposes is an entirely mineral oil, distilled from petroleum and refined. No oils of either animal or vegetable origin can be used and their presence as an adulterant in transformer oil renders it absolutely unfit for use. Mineral oils are combinations of carbon and hydrogen and if they are burned or decomposed in any way, nothing but carbon or hydrogen can be formed, or some of the lower gases of the same series, and none of these has any injurious effect upon either insulation or copper. This is not true of animal and vegetable oils; they also contain oxygen, and age and decompose naturally to a certain extent, forming either weak acids or water, or both. Now, acids have a very bad effect upon both insulation and copper; they soften and weaken the insulation and corrode the copper. Moreover, these oils in decomposing form and collect more or less water and its presence has the effect of very greatly impairing the insulating qualities of the oil.

One of the valuable properties of transformer oil is its ability to carry heat from the body of the transformer to the shell, whence it may be radiated. This is probably the principal value of oil for use in transformers. In the old air-cooled transformers there was always a portion of the transformer which it was very difficult to cool, and there was considerable overheating in such spots. When a transformer is immersed in oil there is much less chance of this occurring. The oil acts in much the same manner that water does in a teakettle, and everyone knows it is impossible to burn the bottom out of a teakettle as long as sufficient water remains in it to cover the bottom.

Temperature Effects

In order that the heat may be carried away from the transformer body to the transformer case, it is necessary to have some circulation of the oil, but this comes naturally in the same way that circulation of water occurs in a tank when heat is applied at the bottom and the hot water rises to the top, while the cold water descends; in the same way the hot oil rises and the cold oil descends, keeping up a constant flow. From this it is evident that there must be circulation in order to keep the transformer cooled properly, and it therefore follows that an oil which would solidify at ordinary weather temperatures would not be suitable for this work. It is, therefore, customary to use an oil which will not solidify at a temperature above 20 degrees below zero. This is a lower temperature than usually occurs even in many of our Northern localities, but should the oil in a transformer freeze up, at so low a temperature, the heat generated in the core when the transformer is in service would tend to warm the oil to a flowing point.

As already stated, transformer oil is distilled from petroleum; it is of the same series as gasolene, kerosene, naphtha, etc., though of a considerably higher boiling point. Its specific gravity ranges from about 0.82 to 0.825 for a very light oil to 0.88 for a heavy oil. These oils are not inflammable at ordinary temperatures and in order to get them to burn it is necessary to heat them until a temperature is reached which is known as the "flashing point." At this temperature a flame applied to the surface will produce a momentary flash over the top o the oil, indicating that considerable vapo is passing off which mixes with the ai above the oil to burn, but there is no enough vapor liberated at this temperature to make the flame continuous. Stil further heating of the oil finally brings i to a temperature at which it will take fire and burn of itself. The flash point o transformer oils ranges from 265 to 37 degrees Fahrenheit and the burning temperature ranges from 285 to 425 degree Fahrenheit.

It may be readily understood that if a oil has a very low flashing point it i apt to evaporate readily at a temperatur produced by an ordinary heavy workin load on the transformer, and will therefore require constant replacement. Fo this reason it is more desirable to use a oil of a flashing temperature high enoug to prevent the loss by evaporation from running very high. To show what wi occur with an oil of low flashing poin the writer made a test of an oil with flashing point of 278 degrees Fahrenheit; the loss by evaporation was 3.15 per cent. in 24 hours, when the temperature was maintained at 205 to 210 degrees Fahrenheit. As this is but little above the full-load temperature of a large transformer, the test indicates the possibility for loss of oil when the flashing point is too low.

Insulating Qualities

The insulating or dielectric property of an oil is a very important one and as it is the one which comes most prominently to the attention of an operating engineer it deserves a great deal of consideration. A good quality of oil is a very strong insulator; its insulation value will run somewhere in the neighborhood of fifteen to twenty times that of air, which is generally considered one of the best insulators available. There are factors, however, which may quickly reduce the insulating quality of an oil to less than one-tenth of normal without any visible change taking place; the most effective, and therefore the most objectionable, of these is the presence of moisture in an oil. It is a remarkable fact that the addition of so small an amount of water as one one-hundredth of one per cent. of the quantity of oil will reduce the dielectric strength of the oil to less than one-half, while the addition of one twenty-fifth of one per cent. will reduce it to one-tenth or less of its original dielectric strength. From this it will be obvious that it only requires a few drops of water to a gallon of oil to produce very serious reduction in the insulating value of the oil.

The danger due to the presence of moisture makes it important that a transformer when in operation or in construction should never be left standing open out of doors if there is any likelihood of rain, snow or dew, or any chance of any moisture getting into either the transformer casing or the oil, nor should transformer oil be stored in a damp place unless hermetically sealed, as sweating may occur which would cause moisture to collect in the oil.

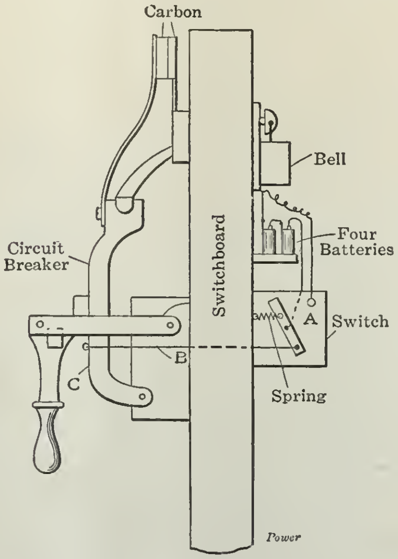
A good grade of oil will have a dielectric strength or electrical resistance to puncture sufficient to withstand a difference of potential of from 25,000 to 35,000 volts between terminals one-tenth of an inch apart, or, stated for a distance of two-tenths of an inch, which is customary, the value for this distance will range from 35,000 to 50,000 volts. The test for dielectric strength is usually made with ½-inch flat disks adjusted exactly one-tenth or two-tenths of an inch apart and covered with at least one inch of oil in the testing apparatus. Alternating current is always used for testing and it is usually obtained from a high-potential step-up transformer. The breakdown voltage is usually measured and recorded as the effective voltage indicated by the voltmeter at the primary terminals of the transformer, but the actual breakdown is probably caused by the instantaneous maximum voltage, and it is therefore desirable for such tests that a wave form as near as possible to a sine wave should be used. When the oil breaks down (is punctured), it does so with a sharp red flash or snap and this should not be confused with the slight static discharge which may take place between the terminals before the actual breakdown of the oil occurs.

To show the effect of moisture upon the dielectric strength of oils, the accompanying chart has been prepared, in which the curves indicate the breakdown voltages with different percentages of moisture contained in the oil, the test terminals being located one-tenth of an inch apart. The two curves here shown have been taken from different sources, and give practically the same results for moisture percentages below 0.01.

A Simple Circuit Breaker Alarm

By R. O. Warren

In electric-power plants of considerable size, a switchboard attendant is always on the watch to detect and correct anything that goes wrong on the switchboard, as well as to adjust voltages and put machines in and out of circuit. In small power plants, however, especially street-railway power houses, the engineer is switchboard attendant as well, and disturbances sometimes occur which interrupt the service by the opening of a circuit-breaker, while the engineer goes about his work in blissful ignorance that anything is wrong. This sort of thing happened so frequently in the power house of the North Carolina Public Service Company, Greensboro, N. C., that the engineer in charge installed the simple device illustrated herewith.



Circuit-breaker Alarm

On the back of the switchboard is placed a small switch *A*, the two terminals of which are properly connected to batteries and a good, loud, electric bell. A cord *B* is attached to the lower end of the switch lever, and passes through a hole in the switchboard to the movable contact arm *C* of the circuit-breaker. The cord is of such length that when the circuit-breaker arm goes wide open it closes the switch *A* and the bell rings until the circuit-breaker is reset. With this arrangement the engineer does not need to watch the circuit-breakers at all, because the bell is loud enough to be heard all over the plant. A spring pulls the switch *A* open when the circuit-breaker is closed.

LETTERS

Three Wire and Two Wire Generators

I have read with much interest the discussion of three-wire versus two-wire generators resulting from Mr. Smith's letter in the issue of August 23. I once operated a plant with a balancing set and to me it was not entirely satisfactory. The load was constantly changing from one side to the other with a consequent fluctuation in voltage. This made it necessary to change the rheostats at frequent intervals in order to approximate equal voltages on both sides of the neutral. Sometimes one side would be 115 and the other 105, and by the time they were equalized, the unbalanced load flopped over to the other side and the equalization had to be done all over again.

At present I am handling three-wire generators with one elevator, twenty or thirty motors and a lighting load. I have watched the voltmeters when the load is unbalanced and have never noticed over one or two volts difference on the two sides. At times the balancing coils are carrying about 50 per cent. of their full-load rating. From my experience I should prefer the three-wire machine to the two-wire machine with balancer.

G. F. Barker.

Dorchester Centre, Mass.

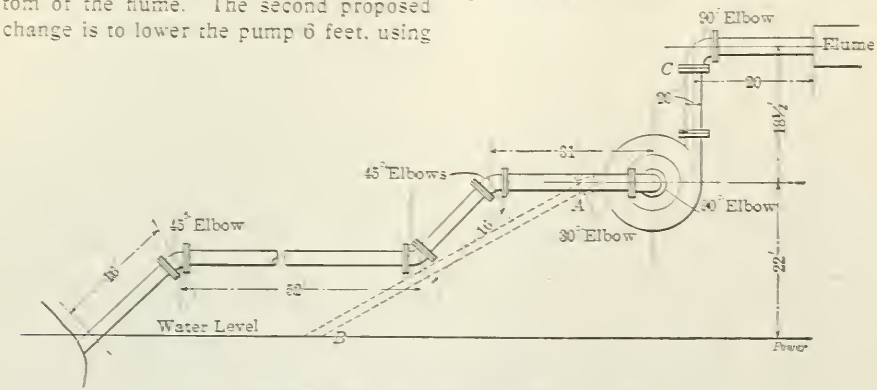
Japan is an ideal country for hydro-electric development because of its many rapidly flowing rivers. There are difficulties in the way of such development, however, not the least of which is the danger of earthquakes. Recently, it was proposed to build a large dam to furnish power for a generating plant of huge proportions, but fearing lest there might be a break in the dam caused by one of the earthquakes so frequently experienced in that region, it was decided to abandon the project.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Irrigation Problem

The accompanying sketch shows a pump connected up for rice irrigation and handling 16,500 gallons per minute. I would like to have the opinions of some of the readers of POWER as to the losses due to friction in the discharge and suction pipes with the pump in this position, also, what would be gained by the following proposed changes: First, if the pump were left in its present position and a ditch cut from the bayou to the point *B*, reducing the length of the suction from 115 feet to 60 feet, as shown by the dotted line; putting in a 30-inch elbow at *A*, and discharging directly into the bottom of the flume. The second proposed change is to lower the pump 6 feet, using the same construction of suction but reducing the length to 46 feet.



PRESENT AND PROPOSED PIPING LAYOUT

L. J. ROBERTSON.

Iola, La.

Specifications for an "All-round" Engineer

Some time ago, being out of a job, I decided to advertise for one, to which I received the following reply:

"We are in search of an engineer and electrician to take the place of our present man, who expects to leave soon. We have a plant containing two boilers, one high-speed engine, one Corliss engine, one 35-kilowatt alternator and one 65-kilowatt alternator. The 65-kilowatt alternator and the Corliss engine are operated regularly, the other unit being kept in reserve.

We want a man who throughly understands the care and management of boilers, engines and dynamos, and who can do all ordinary repair work on them. He must also be able and expect to do such house wiring as may be called for, and keep our street lights in repair. We have no arc lights, the town being lighted with incandescent lamps. The engineer has a fireman to help him, and they, being all the employees we have, are expected to do everything that is necessary to operate and keep the plant and system in operation and repair.

Our fireman is a negro and the engineer is expected to see that he performs his duties, and in the event that the fireman should be missing, the engineer would be expected to perform his duties, till someone else could be procured.

The plant runs from dark till daylight and the engineer is required to be at the plant from starting time till 1 a.m., the fireman remaining the rest of the night. We require the engineer to make daily reports to the office on printed forms, provided for that purpose, and we want someone who is willing to keep up with our report system. We want a man who will feel an interest in the plant and the success of the company, and who will devote his best energies to that end. He must be able to climb a pole, or do any other work that may be necessary around such a plant or system.

Our company has not a large income and for that reason we cannot afford to pay a large salary, but we are always willing to recognize good services. We are paying our present man $60 per month. If you feel that our conditions are not too burdensome and that you would be willing to undertake them, we would be glad to have you write us, giving responsible references."

G. J. PLACE.

Bridgeport, Conn.

Feeding a Boiler

The usual method of feeding a boiler is to have the feed pipe enter the front head just above the top row of tubes and extend to within about 3 feet of the rear head, where it crosses over and discharges through an elbow between the tubes and the shell. A much better way, however, and one that has been in successful operation at our plant for some time, is to have the feed pipe enter the front head in the usual place but extend into the boiler only about 2 feet, where it crosses over, discharging through a pipe capped at the end and into which has been cut a ½-inch slot which will allow a flat file to be used for cleaning out the pipe. This is the coolest part of the boiler, and scale trouble is considerably lessened.

A. RATHMAN.

Chicago, Ill.

Check Valves in Cylinder Drains

To drain the steam chest, cylinder, etc., of a steam engine, it has been the practice to run a pipe from each chamber to one main drain which leads outside of the building or into the sewer. This requires a globe or an angle valve in each pipe as close as possible to the cylinder or chest to be drained, often making them almost inaccessible. This frequently results in careless engineers opening just one or two drains which are within reach, and often these are the ones that do the least good. In a compound engine there are usually six or more globe or angle valves in the drains, and in a duplex pump there are as many as twelve in the steam and water ends combined.

A much better plan is to use check valves instead of the globe or angle valves and place them close to the chest or cylinder, only a short nipple being used. Then pipe all the independent drains into one main drain, placing the check valves with the lift away from the cylinder. Increase the size of the main drain to accommodate the number of drains entering it and place a large globe or angle valve so that it can be opened and closed by the engineer while standing at the throttle. An extension handle will come in handy for this. The check valves employed should not have over one-third the lift of the pipe size in inches, that is, a ⅜-inch check should have ⅛-inch lift.

By closing the main drain valve all the drains are closed simultaneously, and I have had no trouble in using this system on tandem or compound duplex pumps where 20 drains led into the main.

There might be fewer piston rods bent or scored if the simple drainage system were universally installed.

LEON K. PULLIAM.

Pensacola, Fla.

What Size of Reducing Valve is Required?

We are installing a steam-heating system in connection with our electric-light plant and naturally a pressure-reducing valve will have to be installed to supply live steam at times when the exhaust steam is not sufficient. I find a great difference of opinion among engineers as to the proper size of regulating valve to install. Our main steam header is 4½ inches in diameter and the regular boiler pressure ranges from 100 to 125 pounds.

The capacity of the heating main is 7000 pounds of steam per hour at 5 pounds back pressure and there will be times when we will have more than this amount of exhaust steam. To meet these conditions, pressure-regulating valves have been recommended, ranging from 1½ to 4½ inches.

We have one 14x36-inch Corliss engine running at 100 revolutions per minute and one 14x15-inch four-valve Atlas engine running at 200 revolutions per minute. The minimum amount of exhaust steam available will be when the four-valve engine is running alone at a light load of about 30 horsepower, at which it will require about 50 pounds of steam per horsepower-hour. This will make 1500 pounds of steam per hour plus the amount used by the duplex boiler-feed pump. Generally it will not be as low as this, as we do the city pumping and nearly always have one pump running which would make the exhaust more than double this amount.

It is my opinion that a 2-inch reducing valve would be about the proper size. In figuring the capacity of a 1½-inch pipe at 50 pounds boiler pressure I find that it will supply 7000 pounds of steam per hour at a little less than 6000 feet per minute. A 2-inch pipe having nearly twice the area will deliver about double this amount. Of course, there will be considerable friction loss in passing through the valve and for that reason I have selected the 2-inch size. On the other hand, there may be times when we will increase our back pressure and consume more than 7000 pounds of steam per hour, but that is a question quite remote and need not be given much consideration. A 4-inch valve would become badly wiredrawn after a short time due to the small opening which would be necessary in handling the average supply. I would be pleased to hear from the readers of POWER as to the proper size of valve to install and also their experiences with the piston and diaphragm types of pressure-regulating valves.

G. S. SPRAGUE.

Geneva, Neb.

Firing with Natural Gas

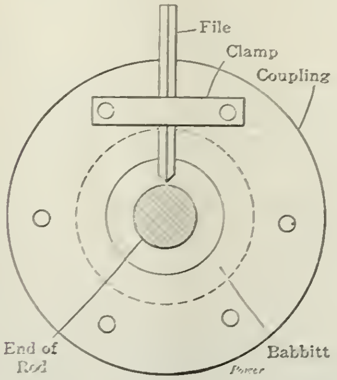
I would like to hear from some of the readers of POWER as to their experiences in firing with natural gas. I have three 150-horsepower return-tubular boilers, carrying 100 pounds steam pressure. What kind of burners gives the best results, how should the draft be handled and how should the firebrick be arranged in the furnace? What amount of gas is required per horsepower and how much water will 100 cubic feet of gas evaporate from and at 212 degrees Fahrenheit?

F. P. WILSON.

Grafton, W. Va.

A Homemade Lathe

Recently I was called upon to put in a new section of piston rod in a 7-inch steam feed in a sawmill, it being desired to lengthen the feed by 6 feet in order to afford a longer travel. The rod was $2\frac{13}{16}$ inches in diameter and hollow; one end was bored out and at the other end a pin, made of a piece of threaded shafting, was shrunk on. When we attempted to screw the new section onto the pin we found that the latter was too large. Some way had to be devised to chase the threads, as filing did not seem to help matters any. The nearest machine shop was about 20 miles distant and it was imperative that the mill be in operation the next day.



COUPLING AND FILE USED AS A LATHE

While passing through the blacksmith shop I noticed half of an old flanged coupling and instantly devised a way out of the difficulty.

I took the coupling and drilled three or four holes in the hub to serve as anchor holes for babbitt. Next, I made a threading tool out of an old three-cornered file, and clamped this against the flange of the coupling. Then I put the coupling over the pin at the end of the rod, filled the bore with babbitt and gripped the device with a pipe wrench at the hub of the coupling and the lathe was ready for use. The sketch shows an end view of the lathe. Three cuts did the work. Of course, I had to cut just half of the pin at a time, and then take the coupling off and turn it around, and cut the other half, but the work was accomplished satisfactorily and the thread made to fit the new section.

N. HOWSE.

Bossier City, La.

Hot Discharge Pipe

A few days ago upon entering the compressor room I detected the odor of burning oil and began to look around and to feel the bearings, but found them all cool. The discharge pipe from the air cylinder is in an obscure place but I finally discovered that this pipe was red hot for a distance of about 4 feet from the cylinder. My first thought was that the cooling water had been stopped in some way and I began to think of what would happen if it should suddenly start again. I closed the throttle, then shut the valves of the cooling-water line and began to carefully look things over. I was surprised to find the air cylinder cool, so lifted the relief valve on the receiver and started the engine, running it without any air pressure until everything was cool.

I now closed the relief valve and ran the engine against 90 pounds pressure, and in about five minutes the pipe began to get hot again. I then shut down the engine and took off the cylinder head, and found the discharge side of the cylinder full of burned oil and ashes. Some dirt and rubbish must have gotten in regardless of the fact that there was a strainer in the intake.

It is my opinion that the area was so small that the excessive friction heated the air hot enough to make the pipe red. After removing two buckets of burned crust, I was bothered no more with the pipe heating.

M. G. FLETT.

Amherst, N. S.

Valves and Packing to Withstand Acid

We have about a dozen small pumps each pumping hot acids into large wooden tanks, and have tried hard-rubber valves with the result that after a few days some cracked and others broke into pieces, and at times the broken parts could not be found. Next, valves cut out of thick, soft rubber were tried; these would have lasted longer if they had not worn out so fast at the middle, around the valve-seat screw. We used flax packing for the pistons in the acid ends of the pumps, but this rotted out in about a week. Rings of thick, soft rubber were then tried but these wore off all way around and stuck to the cylinder walls.

Will some of the readers of POWER kindly inform me of anything that would make more durable valves and better piston packing?

Frank Demarle.

Barnards, N. Y.

Flywheel Trouble

A friend called upon me one morning to find out what was wrong with his engine, as the rim bolts of the flywheel were breaking and were liable to injure someone. Upon arriving at his plant I found an 18x42-inch Corliss engine running at 85 revolutions per minute, and pounding badly. We shut down the engine and found that the head-end steam valve had 3/8-inch lead and the crank end was set on a line with the edge of the port. This was the cause of the pounding and the breaking of the rim bolts. He claimed that the valves were set in this way to save steam.

This same engineer had a new feed-water heater but was too lazy to connect it up, preferring to take warm water from the building returns in winter and cold water in the summer.

Walter W. Smith.

Chicago, Ill.

Knockoff Cam Positions

Assuming that both knockoff cams on a Corliss engine are of the proper thickness to knock the hook clear off the block and are properly adjusted to give the right cutoff, if the governor is blocked in such a position that the long rod is at right angles to the head-end cam lever, then the short rod should bear the same relation to the crank-end lever. This leads to a question which may be satisfactorily answered by some experienced engineers who are readers of Power.

Does the design of any Corliss engine allow shortening or lengthening of one of the cam levers? If not, then why could not one of them be adjusted in this way, so that the crank-end lever might be the shorter? In this way the engineer might be able to secure approximately an even distribution of steam either with a very light or a very heavy load. This shortening of the crank-end cam lever would balance the angular advance of the connecting rod.

Leroy Wheat.

Emmetsburg, Iowa.

Should Water Be Used in Ashpits?

In the plant where I am employed as chief engineer, there are three Babcock & Wilcox boilers connected in one battery, each boiler rated at 150 horsepower. Every morning after the ashes are removed from the ashpits, I have the latter filled with water. The superintendent came through the other day and wanted to know why the water was put into the ashpits. I explained that the coal would burn better and would not clinker on the grates as badly with the water present as it would without the water. He replied that he had been a superintendent for fifteen years and had never heard of such a thing before; he believed that the water would injure the walls of the ashpits by softening the bricks and ordered me to discontinue the practice.

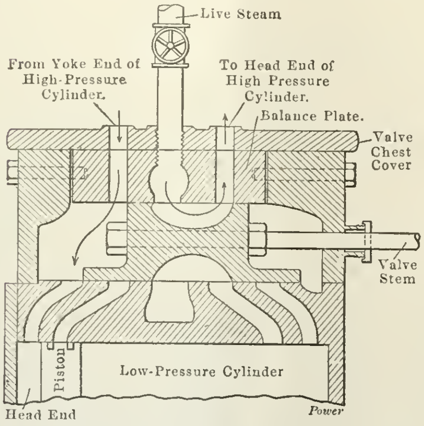
I would like to have the opinions of the readers of Power regarding this. If the superintendent's statement is true, there is much money spent annually all over the country for replacing brick that has crumbled by the absorption of water from the ashpits.

Henry Perlman.

Philadelphia, Penn.

One Valve Performing the Function of Two

I should like the opinions of some of the readers of Power as to a proposed scheme for making one valve, in a compound pump, perform the functions which ordinarily require two valves.



Section through Proposed Valve

This valve is to be located in either the high-pressure or the low-pressure steam chests and will be operated by the usual valve motion employed on the ordinary compound direct-acting pumps. There is only a pipe connection between the one valve chest and the other cylinder, no valve nor chest being required on this cylinder.

The accompanying sketch shows the valve as designed for use on the low-pressure cylinder. This design makes a neat and simple valve which can be made much cheaper than those of the present regular construction. It also produces a valve which is nearly balanced.

Bernard U. Potter.

Holyoke, Mass.

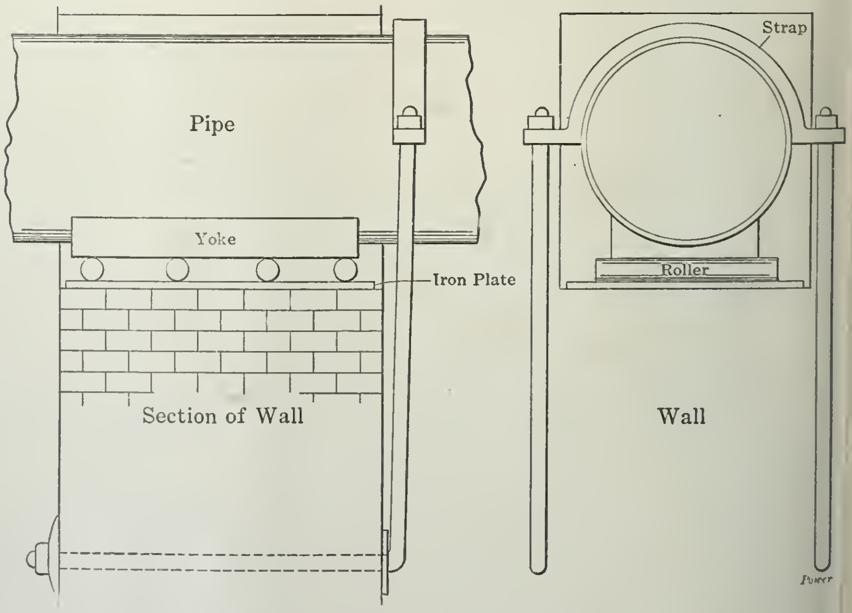
Flexible Pipe Brace

A 10-inch steam pipe about 40 feet long and connected to a 24x48-inch Corliss engine gave trouble by continuously vibrating in step with the strokes of the engine. A device was needed which would be sufficiently rigid to stop the vibration and at the same time flexible enough to allow the pipe to assume its normal position relative to its engine and boiler connections when either cold or hot.

A 16-inch wall separated the engine and boiler rooms through which the pipe passed. The hole around the pipe was made square and an iron plate was embedded at the bottom, as shown in the sketch. A yoke and rollers were then snugly fitted between this plate and the pipe. The wall was drilled and the bolts and strap placed as shown. It will be seen that there is sufficient clearance to allow the pipe to move in either direction under contraction or expansion while only enough tension is put on the bol to stop the pipe from vibrating.

F. C. Holly.

Yazoo City, Miss.



Means of Supporting Pipf to Prevent Vibration

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

What Causes the Vibrations

Referring to E. S. Davis' inquiry in the August 9 issue under the above heading, I would say that probably the vibrations are caused by the impact and recoil of the steam due to the pulsations set up by the frequent starting and stopping of the flow of steam in the pipe and header, caused by the opening and closing of the valve on an automatic engine.

I have met with and overcome quite a number of cases of serious shaking of boiler headers and piping from this cause. It is not always easy to predict in advance from the piping layout of a plant when to expect vibrations severe enough to be troublesome or dangerous, the length and the size of the piping, the boiler pressure, and the speed of the engine, all have an influence on the result I remember one instance in which the vibrations of the boiler and piping were quite severe, and it was considered that the piping was a little too small and that a larger pipe line would correct the difficulty, whereupon the line was replaced by one having 2¾ times the area of the old line. The vibrations were found to be more severe than with the smaller line, owing to the starting of the greater weight of steam contained in the larger line.

In the present case it is not so easy to understand why the vibrations should be much greater when using the No. 1 boiler than when using the No. 2, unless the dead pipe in the header when No. 2 is idle, acts as a compression chamber, or spring, to intensify the pulsations. If that is the case, the placing of the gate valve now over No. 2, against the end of the tee over No. 1 (where it should be to avoid dead pipe, if no more than two boilers are contemplated), would probably correct the difficulty.

I have never found a case that could not be cured by putting a sufficiently large receiver or drum at the throttle, thus allowing a comparatively constant flow of steam from the boiler to the receiver and permitting the violent starting and stopping action only through the throttle and a very short piece of pipe.

It is difficult to determine the exact size of the receiver; the speed of the engine, the character of the load, etc., each have an influence. It should be somewhere from four to fifteen times the volume of the cylinder. In the present case, I would recommend one having eight or ten times the volume of the cylinder.

G. A. Reichard.

Los Angeles, Cal.

Conservation

A great deal is being said about the conservation of our natural resources and rightly so. also about the various acts of Congress regarding conservation. What can any commission do, if those directly interested do not lend their aid in every possible way?

In one of the large cities, a certain railroad, when it wishes to replace damaged or worn-out cars, removes the glass from the sashes, dumps the body into a lot and sets it afire. This applies not only to cars, but to all packing cases and refuse from their wood-working machinery. No doubt that company profits in some way, as it is known to be very efficiently managed in every department, but the fact that they do burn the wood means that quite a quantity of this material is utterly wasted. Could not this be utilized in some way? The above is only a specific case. I believe that it is the custom of most all of the companies to burn their discarded wood, amounting in the aggregate to quite a forest which is annually disposed of in this manner.

On the other hand, the same corporations are very careful as to how their coal is burned, and keep careful comparison records of coal consumption and output. The loss of a fraction of a pound per square foot of grate per hour would amount to a large annual loss unless such records were kept.

While a complete system of records is kept by practically all large concerns, a considerable number of engineers who operate smaller plants keep records, probably not on a very extensive scale, but sufficient to enable the engineer to compare costs from week to week and determine where improvement can be made. Such engineers are progressive and keep uptodate, by careful observation and by a thorough reading of the many valuable articles which appear in the technical press. Many go a step further and keep others posted on matters pertaining to their work. When they have accomplished anything they believe beneficial and of interest, they give the benefit of their experience to others. Such men have been responsible for the great strides made and wonderful results obtained in power production. While due credit must be given to the technical engineer, the actual work must be carried forward by the practical man. Each has his own sphere.

Getting back to conservation and coal economy, there is a large number of plants where the engineer has lost interest or gotten into a rut; where economy is not particularly the watchword; where boiler settings are not taken care of; where scale and soot are allowed to collect on the tubes with consequent high flue temperature; where comparatively low-temperature feed water is supplied to the boiler; where leaks in pipe joints are neglected as well as the engine pistons and valves; where possible advantage is not taken of the opportunity to use exhaust steam for heating, and where a very fair amount of power and coal is directly wasted. That engineer is a conservator who sees to it that full and proper use is made of all apparatus on hand and who trains his subordinates in such a manner as to insure maximum efficiency and economy.

W. E. Rector.

Brooklyn, N. Y.

Installing Globe Valves

There seems to be a difference among the engineers in general as to which way globe valves should be installed. I have had considerable experience along that line but I do not pretend to know it all. I prefer to have the pressure come on the bottom of the disk when possible, as this gives two chances to pack the valve stem, when the valve is shut and when it is wide open, provided the stem has a shoulder and makes a tight joint with the bonnet when opened as far as it will go. If the stem has no shoulder the valve can still be packed when tight shut, whereas, if the pressure is on top of the disk, the only time when it can be packed is when the valve is wide open, provided there is a shoulder on the stem, or when the pressure is off of the valve entirely.

In regard to the stripping of the threads from the stem, I have never had much trouble in that direction. What trouble I have had has been caused by applying a monkey wrench to the handwheel. I also believe that the opening of a valve can be set with a greater degree of accuracy, when it is wanted, say, one-quarter, one-third or one-half open, by having the pressure below the disk than when the pressure is on top.

For a boiler blowoff valve I prefer a

reversible, renewable disk, and a renewable seat. I do not care much for asbestos-packed cocks on boiler blowoff pipes, but do not doubt they are giving excellent service in some plants. I am trying an asbestos-packed cock on the blowoff from the water column and it has given good service so far.

GEORGE DREYER.

Gibsonburg, O.

Advantages of Compression

It would be hard to say anything new on this subject, but in the article under the above title in the August 30 number, Mr. Clarke gives a refreshing and sensible phase when he describes his regard for compression by calling it a "faithful servant."

It is probable that the natural dissatisfaction of the mechanical mind with anything short of exact knowledge has been the cause responsible for keeping this subject so persistently before the engineering public and it cannot be denied that the motive has been a laudable one. Although every phase of the matter has been gone over repeatedly, no universally accepted conclusion has yet been reached, and the diversity of opinion as to its economical status is still so great as to make it appear that we have been traveling in a circle.

I believe in the use of compression when it is needed. In practice, I use as little as possible, and would dispense with it altogether were it possible to do so, believing that the advantages for and against it are about equal, with the balance in favor of compression for its purely mechanical effect upon the journals and connections. By the practical operating engineer it should be regarded as simply one of the very useful and handy resources at his disposal to use or not, as taste or occasion indicates. This is a sort of neutral position between the opposing factions that should be taken by all practical men who aim to keep an open mind and who are wide awake to the best things developed on either side of the discussion of any subject, and it would seem that it is about time we all were getting into that position about this much mooted matter of compression, leaving the economical phase of the subject to the theorists and experts, who really seem to be the most concerned about it.

Perhaps no class of men appreciates the usefulness of compression any more fully than do those who are engaged in erecting engines. At the trial start of a reciprocating unit before the bearings are worn enough to be safely keyed to their final position for smooth running, compression is a valuable aid and often means the difference between a favorable and an unfavorable expression of opinion by the buyer and others interested who throng the engine room at such times.

In Corliss-engine work, especially where the compression can be varied to a considerable extent without affecting the economy, or, temporarily, for a greater extent to meet an emergency, by the simple manipulation of the exhaust rods, the convenience of this medium to help the engine quietly over the centers is greatly appreciated by the erector.

I once received hurry orders to babbitt the main journal of a 14x36-inch Corliss engine in a municipal electric plant. I found the superintendent anxiously awaiting and learned that the town had been in darkness the preceding night and part of the night before. By dint of hard labor the box was rebabbitted and a start made at about 5 o'clock. When the load came on a little later, my first impulse was to seek safety in flight, for the variety of pounding shocks and noises emanating from various parts of the engine would have turned a foundryman's rattle-box green with envy. However, a glance toward the superintendent and the fireman (the latter was also engineer), neither of whom seemed in any way disturbed, reassured me and I decided to stay and die game and when the fireman said, "She's running fine!" my courage rose to normal, although I could not help wondering what kind of a rolling mill "she" had been when not running fine.

Shying around the cylinder head I attached an indicator and obtained a diagram which had perfectly square corners. Slacking the jam nuts on the exhaust rods I lengthened both rods at once. The change made the engine room seem like a cemetery at midnight compared with its former state.

Although expecting the improvement in the engine, I was not prepared for the astonishment of the superintendent, who hurried to the switchboard, thinking his line was down.

This is only one instance in which compression proved to be timely and useful, but there are other instances that could be mentioned, less spectacular perhaps, but equally as interesting and illustrative of the fact that compression is a useful aid to invoke when needed.

The following day the valves were properly readjusted and other things corrected about the engine, obviating the necessity for as much compression as was used in the emergency.

Another side of the question is illustrated in numerous instances where the range of cutoff and, therefore, available horsepower, is seriously curtailed by too much advance of the eccentric to obtain an excessive compression where, if the engine were properly alined and adjusted, it would run better on much less. Frequently, the engineer in such cases is found to be so wedded to the high-compression idea that he thinks he cannot run without an excessive amount.

Mr. Clarke's letter with its account of the difficulties of steam engineering in the 'fifties is exceedingly interesting. It may be of interest to know that the millers of that period were not the only ones to hold the opinion that grain products were superior when ground by water power. In fact, at the present day there are many sections of this country where corn meal ground by steam power is only purchased when "water-mill meal" is not obtainable.

F. C. HOLLY.

Yazoo City, Miss.

Packing Condenser Tubes

Mr. Simpson's letter in a recent issue of POWER on the subject of packing condenser tubes suggests a method for doing this rather tedious job which, while undoubtedly faster than the old method, is very liable to cause trouble. The most essential point in packing a condenser tube is that the joint be absolutely air tight and at the same time allow sufficient movement of the tube to take up the contraction and expansion of the metal under changes in temperature. In the method used by the Wheeler Condenser and Engineering Company, of Carteret, N. J., and criticized by Mr. Simpson, the packing is inserted by using a hand tool and a light mallet which serves to tap in each portion of the corset lacing as it is wound into position. By this method the operator can feel his way and make sure that he is not winding the packing too tightly, causing it to twist or bind. The finished job is then an even, fairly tight coil of corset lacing which exerts sufficient pressure against the tube surface and tube plate to prevent the leakage of air and water and at the same time is not too tight to cause the tube to bind.

If, instead of this method, the method proposed by Mr. Simpson is used, namely, using a carpenter's brace to turn the packing tool, the corset lacing is liable to be wound in more unevenly and hurriedly, resulting in a poorly wound packing that twists and has uneven spots in it. The trouble when using a carpenter's brace is that the operator, unless very well acquainted with his work, will try to go too fast and do careless work.

The matter of proper filling for condenser packing is of much importance. Mr. Simpson proposes boiled oil as the most suitable filling or lubricant for the packing. Twenty years of study on the part of the Wheeler Condenser and Engineering Company has resulted in their determining that paraffin properly applied gives the best results and preserves the tightness of the joint for the longest period of time. Oil is liable to become rancid and break down into its organic acids which may cause corrosion of the tube plates and the tubes.

PAUL A. BANCEL.

New York City.

Cracked Cylinder

In Power for September 20, C. G. Harrison asks about a cracked cylinder. I suggest that he read the various advertisements relating to welding and brazing that appear in the technical papers, especially the *American Machinist*. Then, write to one or two of the firms and explain fully the nature and size of the crack, giving a sketch if possible. They would, no doubt, advise brazing, and would give a figure on what the job would cost. The job could be done on a Sunday without the necessity of taking the cylinder off.

However, if Mr. Harrison wishes to repair the crack himself, I suggest that he uncover the crack and expose it on the outside, drill a ¼-inch hole at each end of the crack, then fit a patch, say about 3/16 of an inch thick and at least 2½ inches wide, over the entire crack. Drill this patch for 5/16-inch machine screws and, using it as a templet, drill and tap the cylinder to suit. Then mix some steel cement and work it well into the crack; apply the patch with cement and screw up tightly. I am sure that this will remedy the trouble.

Regarding the question, "Is it safe to keep on running with the cracked cylinder?" my answer is no. There is a possibility of the crack extending further and reaching a stage where repairs would be costly. Then, again, there is the constant worry. If the engine must be kept going, I would certainly take the time to offset the chance of the crack extending by drilling small holes at the ends and plugging them up.

George H. Handley.

Newburgh, N. Y.

Trouble with a Pumping System

In the issue of September 20, the question is raised as to why a certain pump operated better when placed near a river than it did when at a distance from it.

Two suggestions are made as to the reason for the failure of the pump when placed at a distance from the stream, but can we not consider that both are practically alike and that one is merely the effect of the other?

As the writer says, there should be no trouble in raising the water 19 feet if the pump, valves and piping are all in good condition. But when the suction pipe is 900 feet long there must be many more joints in it than when it is only 50 feet long, and each joint makes the pipe a little less likely to be air tight. There is not only the long suction pipe, through the joints of which the air may leak, destroying the vacuum, but the vacuum must be higher to raise the water 19 feet than when it must be raised only 14 feet as in the second case.

There is also the probability that the piping and pump received some attention when the change in location was made so that they are now in better condition than at first.

G. H. McKelway.

Brooklyn, N. Y.

In regard to the letter of inquiry by Mr. Kropidlowski in the September 20 issue, I think that the trouble must have been in the foot valve, otherwise the water could not have left the pump. If there had been any escape in the suction pipe, priming would have done no good. If the pump worked all right for several hours, it should have worked all right continuously if the power was kept up and the water level did not lower enough to uncover the end of the suction pipe. Of course, we all know that the shorter the suction pipe is and the fewer elbows there are the easier is the lift.

In all cases where the suction pipe is long, or the lift high, it is advisable to use long bends on the pipe instead of square elbows, as shown in the layout. If the strainer, shown in Fig. 3, got blocked with weeds or other material, it would stop the flow of water, but priming the pump would not do any good in starting the pump in this case, so the trouble could not have been due to this cause.

J. E. Noble.

Toronto, Ont., Can.

Centrifugal Pump Experience

In the September 20 issue an account was given of two centrifugal pumps that refused with criminal persistency to deliver water. The conclusion drawn seems to be that the only reason why they finally started was that they had made up their minds to do so. While familiar with the sometimes rather puzzling tricks of this style of pump, I believe that the solution in this case was a simple one. Probably the impeller was air bound. Owing chiefly to bad design, air will sometimes collect around the center of the impeller, allowing the water to be drawn up round the outside by the primer; and even when the primer is throwing a solid stream of water no amount of running the pump even at several times its normal speed will bring water to the discharge. The primer, however, will still continue to throw water. Shutting down the pump and letting the trapped air flow out, and (if it is a small enough pump) turning it backward, will cure this in an ordinary case. But in a severe one like that mentioned, the best way is to tap another primer connection in the pump casing as close to the center of the impeller as possible.

The fact that the pumps under discussion were always hard to start tends to bear out this theory, as evidently all of the piping was tight and every possible detail was carefully looked after.

Of course, it is taken for granted that the suction of the primer entered the highest part of the pump.

However, as these pumps finally worked, it could not have been a similar case to one that I once was called to remedy. A pump in a small plant just erected could on no account be made to throw water. A local machinist who installed it was evidently entirely unfamiliar with centrifugals. At first glance it was seen that the pump was running the wrong way. Upon being questioned as to why it had been installed in that fashion, he removed the suction elbow so as to expose the impeller and pointed out that the curved vanes seized the water like the buckets of a dredger and hurled it out. After trying to explain for half an hour the reason why the vanes were curved, which was listened to very doubtfully, I persuaded him to cross the belt, after which the pump threw the water at once.

O. Monthan.

Tucson, Ariz.

In the September 20 issue, W. N. Wing describes his experience with two centrifugal pumps in a very entertaining manner, and I certainly would like to assist him in his praiseworthy efforts to get the best of those temper-destroying pumps. In justice to these same pumps, I cannot help but state that, in my opinion, it was through no fault of theirs that they did not pick up the water. A centrifugal pump is not designed to pump air, and when it comes to pumping air and water mixed, when the lift is anything to speak of, it will surely balk. In regard to the particular plant where Mr. Wing experienced all of that trouble, I believe that the layout of the suction piping was faulty, that there were some places in this piping where air pockets could form. Consequently, when the pumps were started, there would be some air mixed with the water. The water, being much heavier, acted with greater centrifugal force and therefore took its place at the periphery of the pump, while the air remained at the center and destroyed the suction. The logical remedy was to move the siphon suction from the top to the center of the pump, or as near to the center as it is possible to have it, and on the side opposite to the suction. I have no doubt but that this would have started the pumps in good shape.

Charles L. Andersen.

East Chicago, Ind.

Siphon Heating System

I was much interested in reading the description of a siphon-heating system in Power for September 6. Some time ago I assisted in the installation of a similar system in a hotel at Niagara Falls. The system worked perfectly.

In this system the exhaust of several pumps led into one exhaust pipe, and the jet was coupled in the same way, as indicated in the article referred to. The returns led back to a closed heater and at times we were much annoyed by a considerable back pressure forming in the heater.

This occasionally got so bad that it backed the returns into the lower tier of radiators. We found that a great deal of the trouble was caused by an irresponsible person fooling with a steam trap and leaving the bypass valve open. It seemed impossible to stop this, so the following plan was adopted.

A 2-inch pipe was coupled to the top of the heater and to the exhaust line just back of the siphon. This kept the heater free from pressure and tended to cause a vacuum in it.

The trouble with the returns stopped and we could heat the building with but the slightest pressure in the mains.

William Watt.

Lambton Mills, Ont.

Engineers' Wages

In the September 13 number, C. R. McGahey has an interesting article on "Engineers' Wages."

After pointing out the difference between 'good" and "bad" engineers, he makes this statement, "Wages count for nothing. We cannot all be the president, but if a man amounts to something and puts his ability on the market in the proper way, he will receive a fair price for it." Will Mr. McGahey please tell us the proper way in which to market one's ability; what is a fair price, and how is it determined?

Further on, he writes, "Almost anyone can get into the National Association of Stationary Engineers. What is the result? Some of those who should wear a button in the lapel of their coat are somewhat ashamed to be a member." He objects to this purely educational organization because it admits to membership anyone who is willing to learn.

He next "takes a fall" out of the unions. He writes, "Some men are shortsighted enough to say, let us have a union, . . . this would mean that a good man would have to work for the wages of a poor man, in other words, equal wages." That statement is a gem. He tells us in so many words that unions tend to lower wages. If such is the case, why do the employers of labor and their retainers oppose the unions?

Again, he writes, "Let us have an examining board which cannot be bought, to see how much a so called engineer knows. Determine whether he is a two-dollar man or a four-dollar man, then classify him." There is one serious objection to that proposition, we cannot all be on the examining board.

Further, he writes, "Many so called engineers working for two dollars per day are not worth it." He makes no argument in support of his statement and I ask him what standard he uses in arriving at the worth of a man?

In conclusion, Mr. McGahey writes, "I am 'from Missouri' when it comes to a union for education; I must be shown."

In order to aid him, I submit the following: The ability, both mental and physical, of the engineer is bought and sold on the market in much the same way as is sugar or steel. The sugar trust and steel trust have been organized to control the prices of sugar and steel and have been very successful from the trust standpoint; therefore, I propose that workers, including engineers, firemen, oilers, helpers and all others engaged in the generation of power, organize into one compact body, trust if you please, that they may better control the price of their labor. They should have an effective educational system to raise their efficiency and a compact organization to secure the results due a higher efficiency. Also, an employment bureau, and an insurance department to care for sick and disabled members.

If the trust method works well for our employers, why not for the workers? I hope others also will try to show Mr. McGahey.

C. C. Harris.

Springfield, Mass.

In the September 13 issue I read with interest Mr. McGahey's excellent article on "Engineers' Wages." I think he is about right. To abolish the merit system means not only to equalize wages, but to stop progress, for no man will stay up nights studying engineering to qualify for a position at the top which will not net him a better salary than that received by those lower down.

Some men have risen to the position of chief and have held it for many years simply through influence. Many chiefs have with them men of merit as assistants without whom they could not hope to retain their places. Every year some of them are forced out of their places on account of not being able to run an uptodate engine or ice machine which had been recently installed. Such men are constantly complaining, through jealousy, of others. They can rely only on the "oldest-man" system to keep up in the line of promotion which is sure to lead them to the chopping block when it becomes necessary to rebuild the plant.

Steam engineering today is more of a profession than a trade, and the men most successful are the ones who are the best informed on all subjects relating thereto. For instance, the uptodate office building or factory plant requires an engineer who possesses a general knowledge of the various trades in order to determine whether or not the repair work, done under his supervision, is completed in the shortest possible time and in a workmanlike manner. He is not a master machinist, steamfitter, plumber, electrician or gasfitter, millwright or boilermaker, yet he must possess a general knowledge of all of these trades. The license examination touches on all of them in so far as he is supposed to deal with them.

Trade unions are continually encroaching on the rights of the engineer's license, hoping thereby to make their own business good. The plumbers endeavored to pass laws prohibiting an engineer from putting in washers in sink hydrants and drain pipes. The steamfitters have tried to prohibit them from doing any piping. The elevator repairmen's union has endeavored to pass laws to stop the engineers from doing such work as putting new leathers into valves and packing plungers, thereby forcing the employers to dispense with the use of the machine and pay these men for the minimum of an hour to travel from the shop and a like period of time to travel back, simply to do a job which may require but a half hour.

The question may be asked, what are the engineer's duties? Does he not have all of the various kinds of work just mentioned to look after in the large building and factory plants? And is he not paid in accordance with the size of the plant and his ability to maintain constant service? The man in charge of a 350-horsepower plant cannot expect the same salary as the chief of a 20,000- to 50,000-horsepower public-service plant and no union could ever force the owner to pay him as much. The union might perhaps succeed in forcing the owner of the small plant to buy his power from the public-service company and thus dispense with the services of the engineer in his plant.

I read in a daily paper of the intention of the boilermakers' union to use their influence in securing a law compelling boilermakers on repair work to be licensed, on the grounds that engineers and firemen, machinists and other "inexperienced" men were constantly engaged to put in thimbles, tubes and to calk seams and rivets, and that work done by such men was frequently unsatisfactory and unsafe.

R. A. Cultra.

Boston, Mass.

One or Two Boilers

In Power for September 13, O. Lindblade asks for opinions as to whether it is better to use one or two boilers for a given service. I will not give an opinion but will state my experience and let him draw his own conclusions.

In a certain plant, two boilers were in use and between them they used 6800 pounds of coal per day. In order to do some work on one of them the load was carried with the other and the coal consumption fell about 1600 to 1800 pounds per day.

R. McLaren.

Berlin, Ont.

Horsepower on Nine Pounds of Steam

The development of an ultra high-speed, high-pressure and high-expansion engine was Charles T. Porter's latest engineering activity. An attempt is now being made to bring his work to the surface.

The recent death of Charles T. Porter recalls the fact that his latest engineering activity was directed to the development of an engine which should produce a horsepower on nine pounds of steam per hour. The essential features were 300 pounds initial pressure; superheat to a degree required to prevent initial condensation and furnish the heat converted nto work in the first cylinder; jacketing vith steam at the boiler pressure but not superheated, on account of the more ac-ive interchange between saturated steam ind the containing surface; 30 expan-iions in a quadruple-expansion engine cept within reasonable dimensions by eason of the high mean effective pres-ure and a piston speed of 1250 or 1500 eet per minute; and fixed cutoff at the oint of maximum efficiency. This en-ine was first made known to the pub-ic in a discussion by Mr. Porter of Dr. R. H. Thurston's paper upon "The Steam Engine at the End of the Nineteenth Century," presented to the American So-iety of Mechanical Engineers and to be ound in Volume XXI of the transactions f that body, although some of its ele-ients were described in a number of apers presented by Mr. Porter to the ame society in 1894 and to be found in olume XVI of its transactions. In still nother paper in 1893 upon "The Limita-on of Engine Speed," Mr. Porter, whose ery utterance upon the high-speed engine, of which he was the father, has special significance, had recommended that 600 feet per minute be taken as the limit of piston speed; but this same paper demonstrates that speed is not limited by centrifugal stress in the flywheel, nor a tendency to knock on centers, nor vibration, but simply by the fact that an engine ought not to be run as fast as it can be run, and principally by the prejudicial effect of the large clearance required by engines of high speed. His new cam-operated valve, permitting him to give port openings sufficient to fill and clear a cylinder in the short time available at high speed without large clearance space, removed the main objection, and allowed him consistently to double his limitation in his own practice.

We understand that an engine embodying the ideas of Mr. Porter was built and tested by Prof. J. E. Denton, of Stevens Institute of Technology, with results so satisfactory that the Charles T. Porter Steam Engine Company was organized for its development and manufacture under the patents which Mr. Porter had taken out. The advent of the steam turbine, however, which was at that time being exploited with great energy by large financial interests, led Mr. Porter to advise the cessation of efforts to develop his project then, and he laid it away and, owing to his advanced age, never took it up again.

Looking through his engineering papers his sons have brought the matter again to the surface, and they, thinking that possibly the time is ripe, since the turbine has found its special field, for a revival of the steam engine for its legitimate sphere, have placed the papers in the hands of H. F. J. Porter, consulting engineer, of New York, who, with George H. Barrus, M. E., of Boston, was associated with Mr. Porter in the development of his ideal engine. Mr. Porter has renewed Mr. Barrus' association in the matter and has secured the active interest and assistance of Admiral Melville, now consulting engineer, of Philadelphia, to whom Charles T. Porter outlined his plans when the former was chief engineer in the navy. It is hoped that some way may be found of making available the work which the master devoted to the subject for so many years.

Coal Briquetting in the Year 1909

Due to the large supply of cheap fuel and the high cost of labor the briquetting industry in this country has not thrived. Last year, however, the briquetted output was more than 50 per cent. greater than in 1908.

According to Edward W. Parker, in the rthcoming volume on the "Mineral Re-urces of the United States," for the lendar year of 1909, issued by the nited States Geological Survey, there as a marked increase in the produc-on of manufactured or briquetted fuel the United States in 1909, compared th the previous year.

"The quantity of compressed or iquetted fuel made in the United States 1909 amounted to 139,661 short tons, lued at the works at $452,697. This as an increase over 1908 of 49,303 short ns, or 54.56 per cent. in quantity and of 29,640 or 40.13 per cent. in value. hen this domestic output of briquets, wever, is compared with the 17,000,000 18,000,000 tons manufactured in Ger-ny, it appears insignificant; but it does ow that the industry after many years trial and discouragement is getting rted.

'The successful operation of a briquet-g plant must depend upon its ability utilize one or more of several classes low-grade fuel and to produce an arti- which will compete in efficiency and price with raw or unmanufactured coal or with coke. When comparisons are drawn between the extensive development of the briquetting industry in Europe and the small beginning in this country, it must be remembered that in foreign countries the raw fuel is relatively high priced. In Germany, where the briquetting industry has been most highly developed, the raw fuel is not only more expensive, but also of lower grade than that of the United States. Labor is less expensive in European countries, and, moreover, the wasteful type of coke oven known as the "bee-hive" is now practically unknown. Coal is coked in retort ovens, and nearly all of these are supplied with byproduct recovery equipment, which makes available a large supply of coal-tar pitch for binding material. These conditions are all favorable to the successful operation of briquetting plants.

"In the United States the industry is held back (1) by the large supply of cheap fuel, (2) by the higher cost of labor, which necessitates a higher price for the manufactured fuel, (3) by attempts, which have not yet ceased, to exploit secret processes for which extraordinary claims are made but which have not proved successful in commercial operation. If the byproduct recovery coke process had made more advances in the United States there would doubtless be available for binders a large supply of coal-tar pitch at prices which would induce the investment of more capital in the manufacture of briquetted fuel. There is an abundant supply of raw material which can be used for briquets, and in the utilization of which one of the greatest steps in the practical application of conservation principles could be made.

This consists of materials which unless used in this way are wasted. The classes of low-grade fuel available for this purpose are: (1) anthracite culm; (2) slack coal from semi-anthracite, bituminous and subbituminous-coal mines, which does not possess fusing or coking qualities, and is therefore not available for the manufacture of coke; (3) lignite, which disintegrates on exposure to air, will not stand transportation to distant points, and cannot be stored for any length of time; (4) coke breeze, which possesses high fuel efficiency, but which, because of its small size, cannot be used as such either for domestic or other fuel; and (5) peat, which usually occurs at long distances from the coal supplies, and which, if properly prepared, makes an excellent fuel. The utilization of slack from bituminous coal which possesses coking qualities does not enter into the problem, as it can be and is used for making coke and can be used satisfactorily under boilers. A large quantity of powdered fuel is consumed each year in the manufacture of cement, for which even the lowest-grade fuels can be satisfactorily used. This consumption is growing rapidly, with the phenomenal advance in the manufacture and use of cement and concrete, but it is not enough to take care of the enormous quantities of slack produced in the noncoking-coal regions, which is in many cases burned to keep it from 'cumbering the ground.'

"The utilization of coke breeze for briquetting purposes presents a problem distinctly its own; that is, the destructive abrasive effect which it has on the crushing apparatus and bearings of the machinery. It has been truly said that this material 'cuts like the diamond,' and the repair and replacement of parts of the machinery makes an expensive item in the attempts which have been made to use coke breeze in the manufacture of briquets. Two of the briquetting plants built in the United States have been for the purpose, primarily, of utilizing coke breeze. It is the custom to mix the breeze with some bituminous slack or anthracite culm, and it is found that such a mixture makes a more satisfactory briquet. The quantity of this coke breeze produced in the United States each year and practically wasted at the present time is from 2,000,000 to 3,000,000 tons."

In Mr. Parker's opinion:

"There is absolutely no reason for secrecy with regard to the constituents of patented binders. The field to be developed is so large that there is room for everyone, provided the expense is not inhibitive and the resultant briquet is a satisfactory fuel. The better educated the public becomes in the use of briquetted fuel the more rapidly will the industry develop, the most pronounced retarding element being the tendency to exploit secret methods of questionable merit instead of progressing along conservative lines in paths laid out by the experience of European countries.

"The production of briquets in the United States during the last three years has been as follows:

Year.	Quantity.	Value.
1907	66,524	$258,426
1908	90,358	323,057
1909	139,661	452,697
Total	296,543	$1,034,180

"In order to meet with popular favor in this country briquets must be of a convenient shape for shoveling and for the circulation of air in the firebox. They must be of suitable sizes for the purposes they are intended to serve, and must possess sufficient cohesion to resist fracture and abrasion under rough handling. The very large briquets made in European countries, particularly in Germany, the chief reason for which is the advantage for stowage in bunkers of steamships and tenders of locomotives, are not adapted for use in this country. They must be handled and stowed by hand, and must be broken up before being shoveled into the fire. Labor conditions in this country make the cost of such handling prohibitive.

BINDERS

"Experience in European countries and the investigations which have been carried on in the fuel-testing plant of the United States Geological Survey have demonstrated clearly that the successful briquetting of lignite may be accomplished without the use of any additional binding material, and that for the anthracite, semi-anthracite, bituminous and subbituminous coals the most satisfactory binders are coal-tar pitch, gas-tar pitch and asphaltic pitch, or inexpensive mixtures of which one or more of these are the principal constituents. The plants in successful operation in this country at the present time have added their experience to the evidence on this point. One plant began operating in 1910, using a binder of which oil obtained from the distillation of wood is one of the constituents, with apparently satisfactory results. Inorganic binders, though efficient in cementing quality, have the serious objection of adding to the ash but of adding nothing to the combustible character of the fuel. Pitch binders, on the other hand, contribute combustible material and do not increase the ash.

"Considerable attention has been given recently in Europe to the possibilities of sulphite pitch as a binder in the manufacture of briquets."

Separating Water from Steam

By Prof. C. H. Peabody

Though the distinction is not always made, it is convenient to consider that a boiler may prime or it may foam. When working under normal conditions the steam from a boiler will contain about one and one-half per cent. of priming or moisture; under somewhat unfavorable conditions the priming may be double that amount. This, then, may be called the normal condition of a boiler with proper steam space and free water surface.

When a boiler in regular working, in consequence either of condition of water supply or of peculiarities of construction, is liable to foam, the remedy is to put in a separator. Almost any good separator will do, and usually some stock pattern offered for sale on the market will be chosen because it can be easily obtained and will be cheaper than one made to order, even if made at home.

There are two things to be looked after in a separator; there should be a good chance for the steam to drop the water and no chance for it to pick it up again. To illustrate by aid of a certain separator designed at home and made by the neighboring boilermaker, a horizontal steam drum was made 3 feet in diameter and 6 feet long. Steam entered near the top at one end with an interior extension 2 feet long; the exit was at the top over this extension. Here the water simply fell to the bottom and the escaping steam had no chance to get hold of it again.

In their design most separators depend on the fact that water is heavier than steam and when moving at the same velocity has a much greater momentum. When the steam is given a whirling motion the greater inertia of the water carries it to the outside while the steam remains in the center. A sudden chang in the direction of flow by means of baffl plates will also separate the water fron the steam. Either method calls for separator carefully designed to avoid pro ducing a spray which may be easil picked up again by the steam.

A good separator will extract almos all the water from the steam, leavin about one per cent. of priming. This the normal condition of so called dr steam from a boiler, and consequentl the separator may always be considere as a remedy, as aforesaid. The only ex ception is found in the separator of ce tain water-tube boilers which have litt or no steam space and instead have separator. It is to be noted that suc boilers are habitually run at fifty to hundred pounds higher pressure than found in the engine cylinder and cons quently loss of pressure in the separat is of no importance.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.
Our circulation for September, 1910, *was* 134,000.

October 4.............. 33,000
October 11.............. 33,000
October 18.............. 33,000

None sent free regularly, no returns from ews companies, no back numbers. Figures re live, net circulation.

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Power
and The Engineer

Editorial

A Fig for Human Life

Three men were killed recently in Philadelphia by the explosion of a large cast-iron cylinder which was being tested with compressed air under high pressure. In the newspaper account of the affair the closing paragraph which is suggestive reads:

"The night superintendent, who could give no cause for the explosion, said that he had never before known of the explosion of a similar piece of apparatus. The casting is nearly two inches thick, he said, and could not have burst under the pressure put upon it unless defective. It had been inspected and was supposed to be sound."

Superintendents and others who should be held responsible in such exhibitions of the lack of mechanical common sense are always at a loss to account for the cause of anything which is due either to their carelessness or indifference. If a boiler explodes from overpressure, the failure is gravely charged to waterhammer, to some mysterious cause or to something done by one of the attendants who was providentially killed, and thus incapacitated for refuting the libel. Boilers, receivers and pressure tanks fail from overpressure and in a large percentage of cases the failures would not have occurred if there had been periodic intelligent inspection.

Boilers are provided with safety valves which the careful fireman blows every day or by other means ascertains that they are in working order. Steam-engine receivers are also provided with safety valves which are seldom or never tested. Both boilers and receivers fail and from the same cause, overpressure. But how seldom is the condition which makes overpressure possible investigated otherwise than in a farcical manner or for whitewash purposes.

The cast-iron cylinder which exploded in Philadelphia was evidently defective for it had been repaired and it was being tested with compressed air. Presumably the air pressure was applied for the purpose of finding leaks or other defects. If defective, air under pressure was certain to cause a destructive explosion and should not have been used. Air was used and what might logically be expected happened, notwithstanding the fact that the superintendent had never heard of any one of the hundreds of similar cases which are matters of history.

Had water been used, it would have shown the defects, if any, as readily as air, and if rupture occurred, the damage would have been confined to the cylinder alone, for water, having no elasticity, loses its pressure immediately on the occurrence of a rupture. With steam or air the case is entirely different and when a crack starts the whole expansive energy of the confined medium is expended in producing what usually proves to be a disastrous explosion. Applying the pressure test to any closed receptacle with air or any other elastic medium is a practice that cannot be too strongly condemned.

This cylinder "had been inspected and was supposed to be sound." How easily these all-excusing words, whether true or otherwise, are uttered, as though a supposition of safety on someone's part excused such utter disregard of the rights of the worker to be provided with reasonably safe working conditions. As report after report of disastrous accidents is received, and as the death roll is stretched by each additional victim of mismanagement or carelessness, the belief that human life is the cheapest thing on earth grows more widespread every day.

Water Hammer

It is an awful moment to the man in charge of a steam plant when mains begin to shake and bang as though a giant were pounding them, when each succeeding concussion seems more prodigious than the last, when it appears as though the snapping, swaying system must certainly let go at the next impact, and he has no idea whether the opening or shutting of this or that valve will stop the disturbance or incite to greater riot.

Water hammer is one of the most mysterious and unmanageable occurrences about a steam plant. It will start in an ordinarily well behaved system without apparent reason, and the attendant, ignorant of the cause, often can only await helplessly for it to play itself out, fortunate if in the combat between force and material the metal is the victor; killed or mained and discredited if it is not.

But there is no effect without a cause and engineering needs more practical knowledge of the causes and methods of water hammer. A slug of water traveling along a pipe has all the energy in it (friction neglected) which has been expended in getting it into motion. Suppose for easy figuring, a pipe to have a cross-section of ten square inches. A plug of water which would fill a foot of it would weigh something over four pounds. Suppose the pressure on one end of this plug to be twenty pounds greater than it is on the other. The unbalanced pressure on the ten square inches of one end will be two hundred pounds and if that pressure pushes the plug one hundred feet it will expend twenty thousand foot-pounds of energy upon it, enough to get it into motion at a velocity of nearly five hundred and fifty feet per second.

This energy is a compound quantity comprising space and force—foot-pounds. To get the supposed plug of water into motion, a force of two hundred pounds has been exerted through one hundred feet. To bring it to rest a force must be exerted upon it so great that the product of that force and the distance through which it acts under the impact or in resisting the impact will amount to twenty thousand.

Suppose that the dead end or the valve or elbow upon which the plug impacts can move one-tenth of a foot, considerably over an inch; the average force opposed by the impacted surface to bring the plug to rest would be two hundred thousand pounds.

Is it any wonder that when forces and energies like these are allowed to manifest themselves pipes writhe, fittings break and heavy machines skate on their foundations?

It is easy to say that the way to avoid it is not to get water into the pipes, but so long as the pipes are used to carry steam there will be water in them under some conditions. All that one can do is to get it out as soon as possible, keep it out as constantly as possible and not let it form plugs, between the opposite ends of which there may be a difference in pressure which will get them into motion with harmful velocity. All pipes should be pitched in the direction of the normal steam flow and properly drained. Pockets and loops are dangerous. Care should be taken when connecting anything into the system not to reverse the direction of flow, perhaps pulling back flowing or standing water into a plug, nor to admit steam into a cold system in such a way that it can form plugs to be driven by the pressure behind them into the vacuum which the cold produces in front. When a system of piping has the water-hammer habit go at it intelligently to find out how plugs of water can form between the source of pressure and the place where the hammering occurs, and then fix it or run it so that they cannot form.

There is much to be learned upon this subject, and we should welcome discussion especially with concrete examples. If any case has been mastered, tell how you did it; if not, give all the particulars and perhaps Power readers will show you how to vanquish it.

Drawing Fires

About the first instruction a new man in the boiler room receives is on handling the fires. Among other things, he is told to put either wet ashes or green coal on the fire in case of low water, a badly leaking pipe or burst tube. In other instances, he is told to pull the live coals from the grate should the blowoff pipe burst or any other accident occur that would lower the water to a dangerous level.

Such accidents frequently happen, so that pulling a fire is no uncommon occurrence, and easily accomplished when the grates are on a level with the fire door or they are of the dumping type. Pulling a hot fire from the sloping grates of some types of mechanical stoker in which the hottest bed of live coals is a foot or more below the furnace doors presents a more difficult problem. With such a stoker it would be almost an impossibility to draw the fires quickly, and should it become necessary, there would be no time to stop and think out a scheme for doing the work.

One engineer recently admitted that if such a demand should arise, he would not know what to do. He had considered the matter, but had not settled on any particular method of procedure.

How many more are in the same position, not only in regard to drawing a fire, but in the matter of handling other power-plant apparatus in case of emergency? Such matters should be firmly settled in mind in advance of the emergency call. Every engineer and fireman should have a clear idea as to what to do in case it becomes necessary to protect a boiler from the heat of the furnace. Would it be better to draw a fire or cover it with green coal or wet ashes? In the case of a furnace where the fuel bed is considerably below the furnace door, or where there are no fire doors in the boiler front, there is but small chance of drawing a fire, and it is a question as to whether it is even advisable to do so.

Blanketing a fire is the quickest way of reducing the furnace temperature when there is a pile of coal in front of the furnace. But what would be the better way if the boilers were equipped with dutch-oven furnaces in which the stokers are fed from the top of the setting and there is no coal on the fire-room floor? What would be the best thing to do in case the ashpits were so arranged that the ashes fell to the basement out of reach of the men in the fire room?

These conditions may not exist in all steam plants, but they do in one instance. The furnaces are of the dutch-oven type, the grates are inclined toward both sides of the furnace, and the coal is delivered from overhead coal bunkers. Most of the incandescent fuel in the furnace is some distance below the furnace door and the ashes fall into an ashpit in the basement. There is no coal on the boiler-room floor, and the ashes are not within reach.

What should be done if it became necessary to reduce the furnace temperature instantly in case of serious accident to any of the boiler fittings?

Corn Cobs as Fuel for Steam Boilers

A Western correspondent wants to know if corn cobs mixed with slack coal and used as fuel are in any way injurious to a boiler. He is engineer of a roller mill where the cobs are given away, and when he proposed to burn them the objection was made, and generally sustained, that they would ruin the boiler. He tried them, however, mixing them with from 20 to 25 per cent. of slack coal, with the result that the coal burned better, he did not have to use a slice bar, got along with less draft and the fireman after a day or two's experience with them was much pleased as it made his work easier.

We do not see why there should be any injurious results from the use of this kind of fuel. It is pure cellulose, the same as soft wood, and contains nothing that would make the products of combustion deleterious to the boiler. If there is any general conviction, however, on the part of people who live in sections where cobs abound that their use for fuel in steam boilers is not advisable, there is likely to be some basis for it and we should be glad to have some information founded upon actual experience. There must be thousands of tons of this material produced every year and with the exception of forming the basis of the Missouri meerschaum, we know of n other application for them.

Do you know all that you ought t know about the proposed Institute o Operating Engineers? Inquiries addresse to the Institute of Operating Engineer 29 West Thirty-ninth street, New Yor City, will secure full particulars concer ing the purpose of this organization an reasons why it will pay you to becom a member. It may pay you to look th matter up, right now.

Zinc is often used to prevent the co rosion of boilers. Its efficiency is due the fact that any metal with a great tendency than iron to dissolve, on whic hydrogen is liberated with more difficul than on iron, will protect the latter at t expense of the more soluble metal. T protection, however, becomes ineffecti in pure water at a distance greater th an inch. Salt or some other electrol will extend the zone to 20 inches or mo will extend the zone to 20 inches more.

Inquiries of General Interest

Influence of Reverse Lever

With a link-motion valve gear, what effect does "hooking up" have on the steam distribution?

I. R. L.

"Hooking up" causes a valve travel which is a resultant of the motion of both eccentrics, and produces the effect of a single eccentric with a shorter throw and more angular advance hastening each event. Lead is increased, and cutoff and compression occur earlier in the stroke.

Pressure on Flat Surfaces

How is the allowable pressure on flat unstayed surfaces calculated?

P. F. S.

One rule says: Multiply the thickness of the plate in inches by ⅙ of its tensile strength, divide this product by the area in square inches and multiply the quotient by 0.09 for the working pressure.

Sun and Planet Gears

With two gear wheels of equal diameter, one stationary and the other revolving around it, how many turns on its axis will the revolving gear make in going once around the stationary one?

F. H. R.

The moving wheel will make two complete revolutions on its axis for each circuit.

Mean Effective Pressure Rule

What is the rule for finding the mean effective pressure in an engine cylinder when the steam pressure and point of cutoff are known?

E. P. R.

The mean pressure of steam expanding in an engine cylinder, neglecting condensation and clearance, is calculated by the formula,

$$Pm = p\,\frac{1 + hyp.\ log.\ R}{R}$$

in which

Pm = Mean pressure;
p = Initial pressure;
R = Ratio of expansion.

The mean effective pressure is the mean pressure minus the back pressure.

Specific Gravity

What is meant by the term specific gravity, and how is it found?

S. G.

It is the weight of a substance compared with the weight of an equal volume of water. The specific gravity of lead is 11.33, which means that a cubic inch of lead weighs 11.33 times as much as a cubic inch of water.

The specific gravity of a substance is found by weighing it first in the air and then in water and dividing the weight in the air by the difference between the weight in the water and in the air. For instance, a body weighs 4.25 pounds in the air and 2.5 pounds in water; its specific gravity is,

$$\frac{4.25}{4.25 - 2.5} = 2.428$$

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Manometer

What is a manometer?

T. F. B.

Any instrument used for the purpose of measuring the pressure of elastic fluids. The term is usually applied to draft gages, mercury columns and similar apparatus, though it is equally applicable to steam gages and steam-engine indicators.

Benefit of Vacuum

How does a vacuum increase the power of an engine?

B. O. V.

When an engine is run noncondensing the expanded steam must be pushed out of the cylinder through the exhaust port against the pressure of the atmosphere, which is approximately 15 pounds per square inch. But when running condensing the steam is pushed out of the cylinder against the pressure in the condenser, which is below that of the atmosphere. The difference between the pressures on opposite sides of the piston is the measure of the work done by the engine. Removing a part of the resistance by exhausting into a condenser increases this difference in pressure and consequently the power of the engine.

Size of Engine Needed

Will a 20x42-inch Corliss engine, with 90 pounds of steam, running at 75 revolutions per minute, consume more coal to generate 100 horsepower than an 18x36-inch engine under the same conditions, and if so, how much? I have an 18x36-inch engine practically worn out and am considering replacing it with one 20x42-inch and desire to know whether there will be any difference in the coal consumption.

S. E. N.

Either engine at the speed and steam pressure mentioned would be much too large for a load of 100 horsepower. The larger engine will take more steam for the work than the smaller one and this more than one of the right proportions, but just how much can only be guessed at. An engine 14x42 inches with 90 pounds pressure, running 75 revolutions per minute, will develop 100 horsepower economically.

Release and Compression

What is the meaning of release and compression?

R. A. C.

Release refers to that point in the stroke at which the valve opens the exhaust port and releases the steam in the cylinder. Compression refers to that portion of the piston stroke which is accomplished between the closure of the exhaust port in front of the piston and the end of the stroke.

Horsepower of Boilers

I have two horizontal return-tubular boilers, 5x16 feet. What is their horsepower?

H. O. B.

A boiler should develop one horsepower for each ten square feet of heating surface. As the heating surface is not given, an approximation may be had by using the thumb rule

$$\frac{Diameter^2 \times length}{5} = horsepower$$

Substituting

$$\frac{5^2 \times 16}{5} = 80$$

the approximate horsepower of each boiler.

Receiver Pressure

What is the rule for finding the proper pressure to be carried in the receiver of a compound engine?

P. R. P.

There is no rule. For ordinary cylinder ratios of 3 and 4 to 1, operating condensing with a steam pressure of 100 to 125 pounds and a fair load, the receiver pressure should be between 4 and 10 pounds per square inch, and from 10 to 20 pounds for steam pressures from 150 to 175 pounds. The best results are usually obtained where the load is equally divided between the high- and low-pressure cylinders. That receiver pressure which will cause the governor to revolve in the highest possible plane for any given load is the right pressure.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Lagonda Multiple Strainer

The Lagonda Manufacturing Company, Springfield, O., has recently perfected a multiple strainer for removing impurities in feed water, circulating water, etc. These strainers can be installed either outside or inside of the buildings and in any position, horizontal, vertical or inclined. They are made up of different sections which can be cleaned, one at a time, without interrupting the flow of water.

The strainer consists of a cast-iron body having several removable strainer baskets, the number depending upon the size of the line. These baskets can be removed for cleaning, without shutting down the pumps, and as the mesh is made very fine, they eliminate a large per cent. of foreign matter contained in the water.

Referring to Figs. 1 and 2, it will be seen that after the water enters the inlet pipe, it passes up to the top of the valve chamber, where it divides, going down through all of the baskets in multiple. When it is desired to clean the strainer, one of the baskets is drawn to the upper part of the chamber by turning the handwheel; at the same time the valve collar is forced tightly against the valve seat and the valve disk on the bottom of the basket seats on the valve collar. This is all done automatically and entirely shuts off the water supply from the section containing the basket to be cleaned. By means of a small bypass the pressure in this chamber is relieved and the bolts at the top loosened and the flange cap tilted over, exposing the basket.

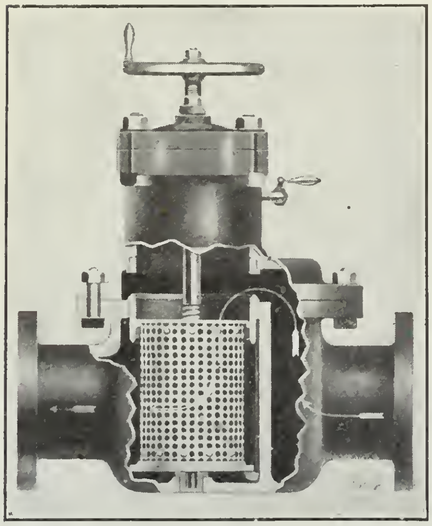


Fig. 1. Side View of Multiple Strainer

The basket may then be taken out and cleaned. In replacing the basket the operations are exactly the reverse. The next basket can be removed in the same way and the whole strainer cleaned without shutting off the water. Should an obstruction rest upon the top valve collar, the basket, when raised to a cleaning position, will have a tendency to tilt the obstruction either into the basket or out of the way of the valve. This is due to the fact that the basket travels nearly its entire distance before the lower valve disk comes into contact with the lower seat of the valve collar.

The effective straining area of the strainer is from two and one-quarter to four times the area of the pipe line and in removing one basket for cleaning the straining area is not reduced more than 30 per cent., it is claimed; thus the pipe line is never throttled in the process of cleaning.

All of the internal parts subject to the action of water are made of bronze, thus eliminating all troubles from corrosion. No leather or rubber washers are used. These strainers are built in sizes from 2 to 48 inches, having two to six baskets, the number depending upon the size of the strainer. They are also suitable for use in either suction or pressure lines and are built to stand working pressures up to 200 pounds per square inch.

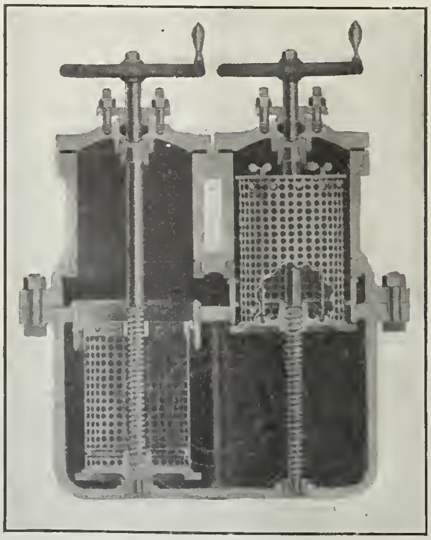


Fig. 2. Sectional View of Two-basket Strainer

In practice, one of the baskets is left out of service so that in case the baskets in use should become clogged the clean basket can be quickly lowered into service.

Weber Coniform Reinforced Concrete Chimney

Details of the new coniform type of chimney now being built by the Weber Chimney Company, of Chicago, Ill., are illustrated in Fig. 1. This chimney is so named for the reason that it is built in the form of a true frustum of a cone, and having a uniform thickness of walls at any given cross-section throughout. The walls and the diameter of the chimney decrease in a uniform ratio from the base of the chimney to the top. Where conditions require, the structure is lined with a suitable brick or reinforced-concrete lining to whatever hight may be necessary, but usually about one-third the total hight.

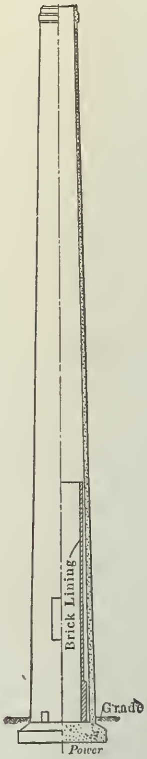


Fig. 1. Details of Weber Coniform Chimney

The reinforcement of this type of chimney consists of vertical bars encircled at regular intervals by steel rings properly bent to the desired circle. The weight and number of these bars are dependent upon the dimensions of the chimney.

Steel bars in lengths of from 16 to 30 feet are used, the joints being made irregular, preventing so far as possible many joints at one hight. In the design of the chimney all the bending forces caused by wind pressure, etc., are taken up by the vertical steel reinforcement, and the resistance of the concrete itself against tension is not considered in the calculations.

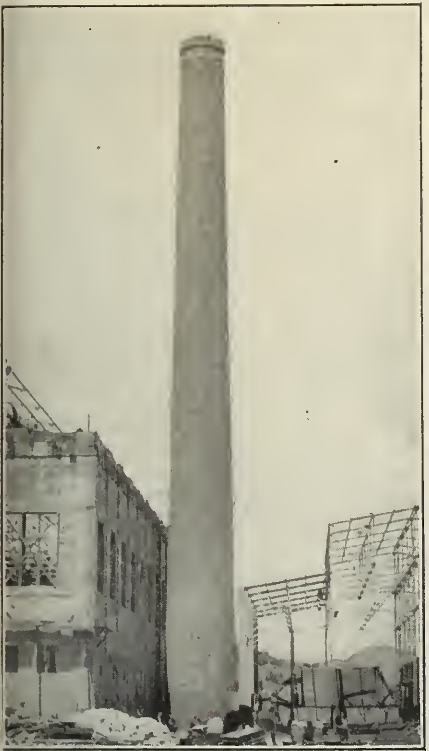


Fig. 2. Chimney Built for Miami Copper Company

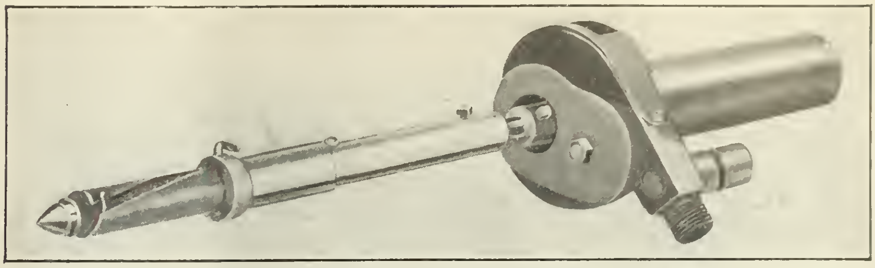
In constructing the coniform type of chimney a special form consisting of a series of wooden units is employed. These wooden units are held together in a rigid manner, forming a circle, at the same time retaining the principle of lexibility, so that the necessary units can be removed from the form at regular ntervals in order to properly reduce the diameter of the chimney and the thickness of the walls in uniform ratio as the hight increases.

Fig. 2 gives a view of a finished chimney and is a good example of the new ype of stack. The chimney and foundation constitute one single solid piece, he reinforcing steel bars from the chimney shaft going down into and being embedded beneath the steel bars in the foundation. The latter, which is thoroughly reinforced, is 27 feet 6 inches square and its greatest thickness is 6 feet 3 inches.

The hight of the chimney is 200 feet and it is 12 feet in inside diameter at the top; the largest outside diameter at the base of the shaft is 20 feet 4 inches, gradually tapering to the top where the outside diameter is 12 feet 10 inches, the thickness of the wall at the top of the chimney being 10 inches. This chimney has a firebrick lining to a hight of 68 feet above the ground level.

Stirling Packing Tool

This tool is an automatic machine for inserting and packing the lacings or strings in the stuffing boxes for the tube ends of surface condensers. It places and starts the end of the lace at the bottom of the stuffing box and automatically winds the lacing about the tube and hammers in each coil; all at the one operation. Each coil of lace is perfectly flattened and wound tightly against the tube, thus keeping it out of the threads of the ferrule.



Stirling Packing Tool

The number of boxes that can be packed within a given time is variable, depending upon the condition of the tubes and stuffing boxes, the position of the work and the ability of the operator. For ordinary work, the manufacturer claims that six to nine boxes a minute throughout the day is the average. Approximately 3000 may be packed in a day.

All sizes of boxes can be packed with one machine, the hammer tubes being interchangeable and fitting ⅝-inch and ¾-inch stuffing boxes; other sizes may be obtained.

The machine is operated with compressed air. When through packing with the air tool, the hammer can be pushed back and a pin inserted to hold it in place. The ferrule attachment can then be put on after taking off the packing tool. This attachment enables the ferrule to be run on tight in a second. This tool can be detached and used with an auger brace or breast drill.

The machine has already been placed in the navy yards of Boston, Brooklyn and Philadelphia, and from all accounts is giving the best of satisfaction. It is manufactured by Robert R. Stirling, 847 Avenue C, Bayonne, N. J.

Dean Dumping Grate

The use of the Dean dumping grate described and illustrated herewith, enables the fireman to thoroughly and

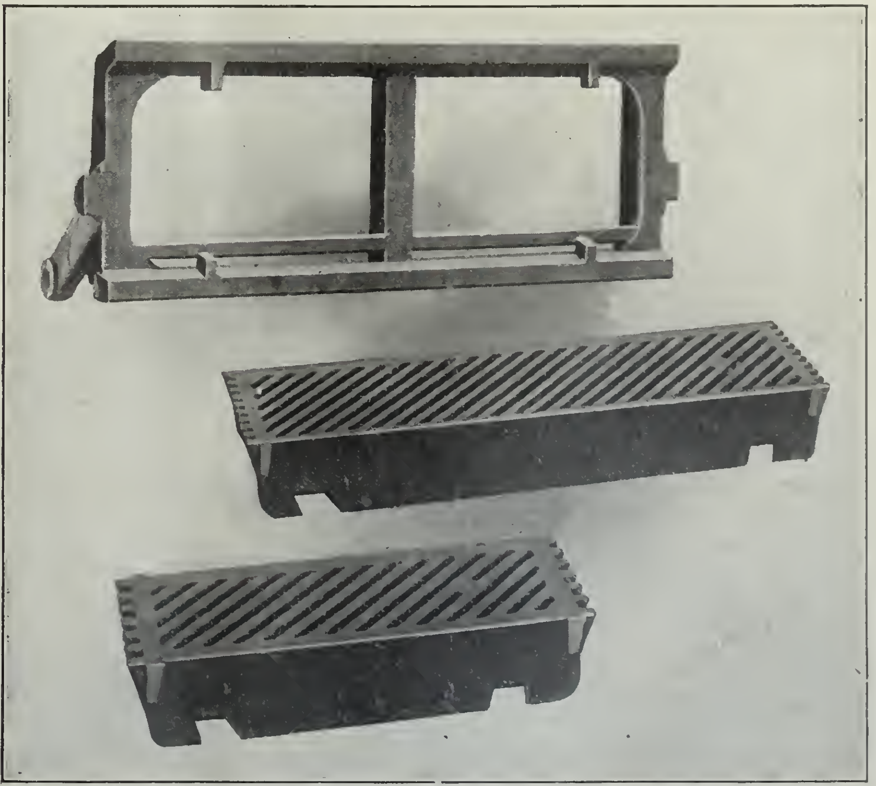


Fig. 1. Grate Sections and Cradle

quickly clean his fire and maintain steam pressure during the cleaning period, which varies from two to seven minutes per boiler according to the size of the grate surface.

The grate sections are each held in a cradle, as shown in Fig. 1. The cradle is designed to give two points of support for the bar. The bar is held in position on the cradle by an angular slot in the rear end, which engages a corresponding rib on the cradle. There is no tendency of the dumping bars to warp downward and break in the middle, it is claimed.

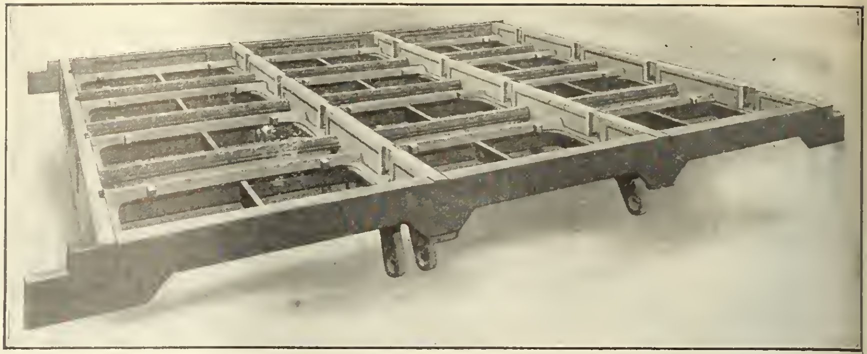


Fig. 2. Grate Frame and Cradles

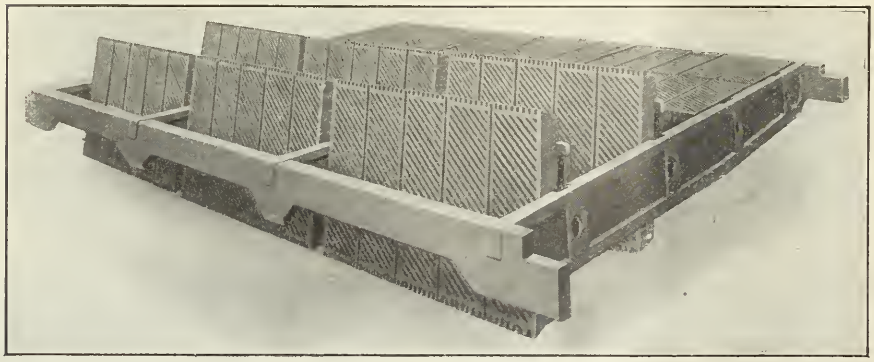


Fig. 3. Position of Bars When Tipped

The grate frame and the cradles on which the bars rest are shown in Fig. 2. Beveled slots are provided in the side and center frame castings, the functions of which are to admit the journals of the cradles. By this method of construction an open slot is avoided, the original strength of the casting is maintained and the use of caps and tap bolts avoided.

Fig. 3 shows the position of the bars when tipped. To clean the front half of the grate, the live coals are pushed onto the rear bars, when by tipping the cradle to an angle of 65 degrees the grate is freed of all ash and clinkers.

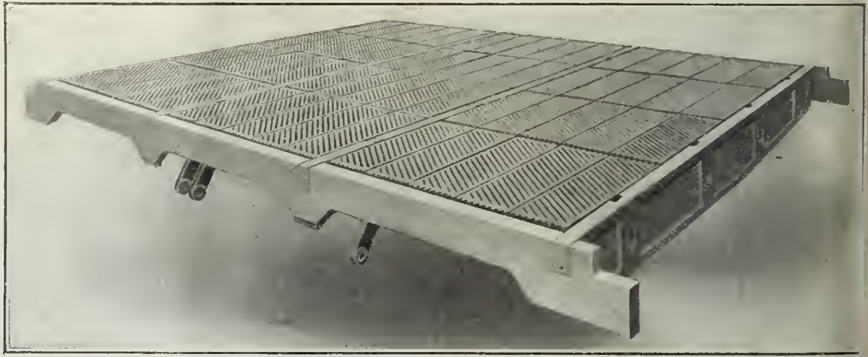


Fig. 4. A Twelve-cradle Grate

The bars are then returned to normal position, and the live coals raked forward onto the two front sections, when the two rear bars are dumped free from ashes and clinker. The entire grate is thus cleaned thoroughly in a very few moments.

In Fig. 4 is shown a twelve-cradle grate designed for a furnace 10 feet 4 inches wide by 7 feet deep, with the bars in a normal position. It is claimed that with this size of furnace the individual grate bar remains perfectly flat owing to the principle of support embodied in the cradle.

The method of installation as applied to brick-set furnaces is shown in Fig. The cradles tip in tandem, two front ar two rear, which greatly increases tl clearance between the floor line and tl bars when the grates are tipped. Th clearance is necessary to permit the a: to drop clear. This design also permi the rods to be brought out so as not come in contact with the ash, and specially advantageous on installatio having shallow ashpits. The space r quired for the operating levers is evide in Fig. 5.

All operating rods are made of f steel. The handles are made of malleat iron and the levers and links of forg steel. This grate is made by Washbu and Granger, 50 Church street, N York City.

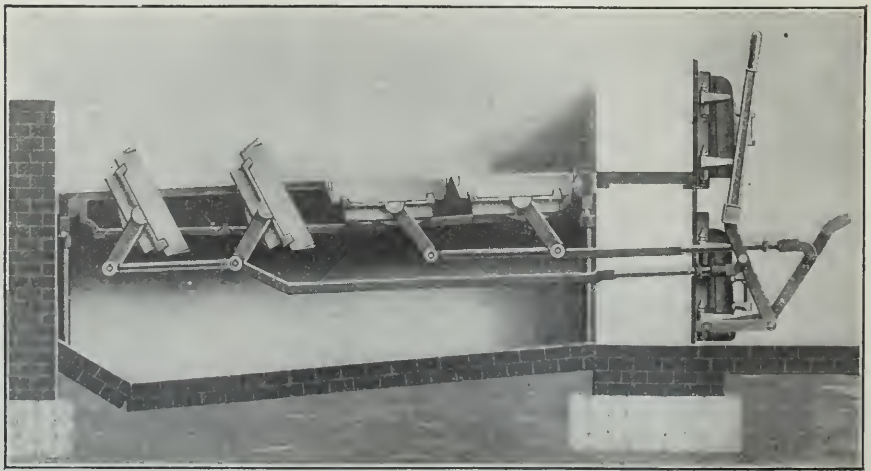


Fig. 5. Furnace Installed in Brick Setting

Stumpf's Engine Turbine Combination

In reciprocating steam engines where the steam is expanded up to the end of the piston stroke, it is impossible to carry out the expansion to the limit. Thus there is always a rounded toe on the diagram showing a loss, as indicated by the section lines in the diagram, Fig. 1. To render this lost energy, or a considerable portion of it, available for use without impairing the efficiency of the engine, Prof. Johann Stumpf, in a patent recently described in *The Mechanical Engineer,* proposes to combine a unidirectional-flow steam engine with a suitable turbine in such a manner that the turbine is operated by the exhaust steam from the engine. When the piston uncovers the exhaust ports in the cylinder wall near the end of its stroke, the expanded steam in the cylinder passes through the exhaust ports, and thence into a low-pressure or exhaust-steam turbine, where the energy of the steam is extracted. In such a plant the piston-controlled exhaust ports offer a very considerable area of passage for taking off the expanded steam from the cylinder, and further, they enable the application of simple means for combining a turbine with a steam engine. Such a combination is known in the case of ordinary reciprocating steam engines with valve-controlled exhaust. It has also been proposed to lead the exhaust fluid from an internal-combustion engine directly from a single exhaust opening into a turbine.

Fig. 2 is a cross-sectional elevation through the center of the cylinder and the turbine casing of the arrangement, and Fig. 3 is a side elevation. Steam is admitted to the cylinder *A* of the unidirectional-flow steam engine through inlet valves *B*. The piston working in the cylinder *A* is adapted, in its travel, to control a plurality of exhaust ports *C* located in the walls of the cylinder and about the center thereof. Shortly before reaching its end position, namely, the dead point in the crank position, the piston uncovers the exhaust ports *C*, and the steam then passes through pipes *E*, which are directly connected to the exhaust ports *C* and lead to or themselves form the nozzles of a turbine *D*. A few of the blades on the rotor of the turbine are indicated in Fig. 2. Each port of the ring or row is shown as connected to a nozzle pipe *E*, which has its narrowest point at or in the immediate neighborhood of the cylinder wall. The long, narrow part of the nozzle pipe allows a gradual increase in the velocity of the steam, which may reach as high as 1300 to 1600 feet per minute. It will be seen that the turbine receives puffs of steam just at or about the dead point of the engine crank, and therefore the maximum energy is obtained in the turbine just at the time when the minimum turning moment is being applied by the engine.

The exhaust from the ports *C* may, if desired, be led into a receiver, consisting of an annular chamber arranged as a belt around the cylinder, and the receiver may be connected to the turbine. The exhaust steam from the engine cylinder may be mixed with steam at the pressure of the condenser, and this mixture, at the pressure corresponding to the proportions of the mixture, may then be led to the nozzles of a turbine, where the pressure rapidly falls. In all cases the full advantages of the condenser pressure are obtained, as shown by the diagram from the reciprocating engine, which is just the same as though no turbine were employed. A number of parallel rows of ports may be provided in the cylinder, and one or more of these rows connected to the turbine and the remaining row or rows to the condenser.

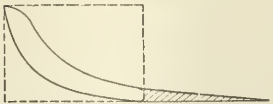


Fig. 1

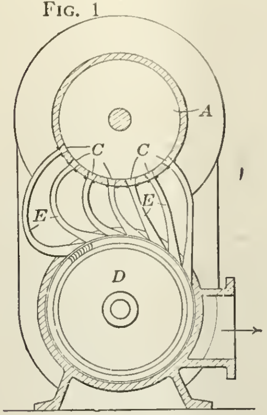


Fig. 2

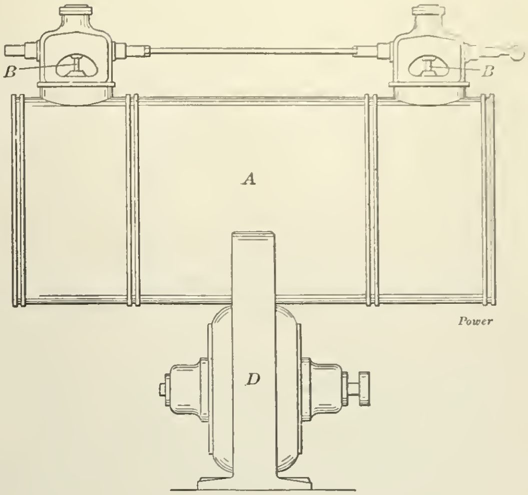


Fig. 3

A Practical Evening Course in Heating and Ventilating

The course has been especially provided by the Harlem branch of the Young Men's Christian Association, 3-15 West One Hundred and Twenty-fifth street, New York City, for engineers, draftsmen, mechanics and salesmen of heating appliances. It will consist of lectures, demonstrations of systems and appliances, discussions and the answering of pertinent questions.

During the first part of the year, gravity, steam and hot-water heating and ventilating apparatus for small buildings will be taken up. This will include the proper proportioning of boilers, mains, radiators, chimneys and flues. Also all special methods of steam and water circulation applicable to small buildings.

During the second part of the year the fan systems of heating and ventilating, and the heating and ventilating of office buildings, hotels, factories, etc., will be covered; this will include such parts of the power plant as are usually in the care of the heating engineer, and will cover mechanically circulated and thermostatically controlled steam and hot-water systems. Individual needs will be given special attention.

The fee for the complete course of two terms is $12; or $7 for half the course of one term. Membership in the Young Men's Christian Association is a preliminary requirement. Membership fee costs either $6 for limited privileges or $12 for full privileges, which include gymnasium, etc.

BOOK REVIEW

Bulletin 425, "The Explosibility of Coal Dust," issued by the United States Geological Survey, traces the growth of the belief in the explosibility of coal dust. summarizes the experiments and mine investigations that have established this belief, and gives the present status of preventive measures. It contains references to and descriptions of experiments made at the testing station of the United States Geological Survey, at Pittsburg, Penn. This station was established and equipped for the purpose of carrying on investigations relating to mine explosions, fuels and structural materials.

"Only within comparatively few years has the dry dust of bituminous and lignitic coals been generally recognized as an explosive agent more insidious, threatening and deadly to the miner than firedamp. Firedamp carries its own flag of warning—the 'cap' in the safety lamp—but coal dust, though visible, does not attract attention until present in large quantities. Firedamp is of local occurrence, and except in notable and very exceptional cases is controllable by careful manipulation of ventilating currents. If by mischance a body of firedamp is ignited in a mine, the force of the explosion is terrific, but the effect is localized unless dry coal dust is present, or unless, as it very rarely happens, an explosible mixture of methane gas and air extends through large areas of the mine. In a dry mine, dust accumulates everywhere and the blast from the ignition and combustion of bituminous dust may traverse miles of rooms and entries and wreck structures at the entrance to the mine. The comparative potential destructiveness of gas and of bituminous dust is strikingly shown by the history of the Pennsylvania anthracite mines. These mines not infrequently have large inflows of gas and the resulting mixtures of gas and air have sometimes been ignited, yet no such wide-sweeping explosions have taken place, despite the presence of dry anthracite dust, as have happened in excellently ventilated bituminous mines."

BOOKS RECEIVED

RULES AND FORMULAE. Indorsed and adopted by the International Master Boilermakers' Association, New York. Cloth; 65 pages, 4x6 inches; 1 illustration. Price, $1.

SWINGLE'S PRACTICAL HANDBOOK FOR MILLWRIGHTS. By Calvin F. Swingle. Frederick J. Drake & Co., Chicago, Ill. Cloth; 411 pages, 5½x7¾ inches; 226 illustrations; indexed. Price, $2.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

FLUID MOTOR. Carlos F. Benitez, Guadalajara, Mexico. 971,517.

STEAM TURBINE. Johann Friedrich Max Patitz, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 971,555.

ROTARY ENGINE. George T. Geer, Denver, Colo. 971,671.

REVERSIBLE TURBINE. Richard Cramp, East Ham, England, assignor to Cramp's Patent Reversible Turbine, Limited, London, England. 971,730.

ENGINE. Christopher John Lake (name changed from John Christopher Lake by judicial order), Bridgeport, Conn. 971,760.

INTERNAL COMBUSTION ENGINE. Theodor Reuter, Winterthur, Switzerland. 971,793.

MOTOR. Daniel F. Lawrence, Brandon, Vt. 971,854.

STEAM TURBINE. Thomas R. Morgan, Archibald L. Atwater and Benjamin A. Crowther, Pittston, and William M. Ash, Williamsport, Penn. 971,918.

GAS ENGINE. George R. Morris, Chicago, Ill. 971,919.

LIQUID FUEL INTERNAL COMBUSTION ENGINE. Leon Wygodsky, New York, N. Y. 971,954.

WATER MOTOR. Elijah R. McCreary, Bush, La. 972,010.

INTERNAL COMBUSTION ENGINE. David Roberts and Charles James, Grantham, England. 972,024.

EXPLOSIVE ENGINE. John H. Fitch, Ludington, Mich. 972,071.

BOILERS, FURNACES AND GAS PRODUCERS

FURNACE. Wliam H. McGuire, Hobart, Okla. 971,548.

GRATE. Horace H. Ruggles, Boston, Mass. 971,565.

BOILER. Luther C. Rhea, Columbus, Ohio. 971,561.

STEAM GENERATOR WITH WATER IMMERSER FLAME. Oscar Heinrich Ulrich Brunler, Brussels, Belgium. 971,724.

OIL-BURNING FURNACE. Ludwig Koczwara, Prague-Kgl. Weinberge, Austria-Hungary. 971,902.

STEAM BOILER FURNACE. August H. Hansler, South Chicago, Ill. 971,996.

HYDROCARBON BURNER. George J. McPherson, Salt Lake City, Utah, assignor to International Oil Gas Producer Company, a Corporation of Utah. 972,014.

NATURAL GAS BURNER. Charles E. Bunner, Grafton, W. Va. 972,056.

POWER PLANT AUXILIARIES AND APPLIANCES

SUPERHEATER. William F. Buck, Chicago, Ill., and Henry W. Jacobs, Topeka, Kas. 971,522.

QUICK-OPENING VALVE. William L. Osborne, Chicago, Ill. 971,552.

FEED-WATER HEATER. William S. Ferguson, Leesville, La. 971,599.

CARBURETER. George M. Schebler, Indianapolis, Ind., assignor to Wheeler & Schebler, Indianapolis, Ind., a Copartnership. 971,689.

STEAM TRAP. Julius R. Tanner, Pittsburg, Penn., assignor to the Pittsburg Valve, Foundry and Construction Company, Pittsburg, Penn., a Corporation of Pennsylvania. 971,801.

NON-RETURN VALVE. Julius R. Tanner, Pittsburg, Penn., assignor to the Pittsburg Valve, Foundry and Construction Company, a Corporation of Pennsylvania. 971,802.

STEAM TRAP BOILER FEED AND WATER METER. George W. Cole, Buffalo, N. Y. 971,831.

CENTRIFUGAL PUMP. Ferdinand W. Krogh and Carl A. Krogh, San Francisco, Cal. 971,849.

CARBURETER. George M. Schebler, Indianapolis, Ind., assignor to Wheeler & Schebler, Indianapolis, Ind., a Copartnership. 971,862.

GREASE GUN. Samuel P. Townsend, Orange, N. J. 971,939.

PRESSURE-REDUCING VALVE. Thomas P. Ford, New York, N. Y. 971,982.

PIPE JOINT. Charles R. McKibben, Pittsburg, Penn. 972,119.

CROSSHEAD. Frederick H. Smith, Pittsburg, Kas., assignor of one-half to Clarence R. Widby, Pittsburg, Kas. 972,133.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC SWITCH. Walter S. Mayer, Philadelphia, Penn., assignor to the Machen & Mayer Electrical Manufacturing Company, Philadelphia, Penn., a Corporation of New Jersey. 971,623.

ELECTRICAL CONDENSER. William W. Dean, Elyria, Ohio, assignor to the Dean Electric Company, Slyria, Ohio, a Corporation of Ohio. 971,667.

ELECTRIC-CURRENT GENERATOR. Herman A. Schmidt, Chicago, Ill. 971,690.

MAGNETIC SEPARATOR. Friedrich Oscar Schnelle, Frankfort-on-the-Main, Germany, assignor to Wetherill Separating Company, New York, N. Y., a Corporation of New Jersey. 971,692.

THERMO ELECTRIC COUPLE. Albert L. Marsh, Detroit, Mich., assignor to Hoskins Manufacturing Company, Detroit, Mich., a Corporation of Michigan. 971,767.

ELECTRIC IGNITION APPLIANCE. Archibald Montgomery Low, Brook Green, London, England. 971,908.

SPARK GAP FOR RADIOTONE WIRELESS TELEGRAPH SYSTEMS. Emil J. Simon, New York, N. Y., assignor to the Radio Telephone Company, New York, N. Y., a Corporation of New Jersey. 971,935.

METHOD OF AND APPARATUS FOR GENERATING ELECTRICITY. Botho Schwerin, Frankfort-on-the-Main, Germany. 972,029.

INDICATING SWITCH BUTTON. John A. York, Franklin, Conn. 972,048.

MAGNETIC ORE SEPARATOR. Anders Gustaf Holmberg, Langgrufvan, Sweden. 972,109.

ELECTRICAL HEATER. Milton H. Schoenberg and George T. Marsh, San Francisco, Cal., assignors to Appliance and Electric Device Company, a Corporation of California. 972,131.

INDUCTION MOTOR. John E. Webster, Pittsburg, Penn., assignor, by mesne assignments, to Westinghouse Electric and Manufacturing Company, East Pittsburg, Penn., a Corporation of Pennsylvania. 971,649.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

ol. 32 NEW YORK, OCTOBER 25, 1910 No. 43

BEING in business for the sole purpose of generating and selling power, the central-station people go about their work in the most aproved manner. They get the best men that they can btain to direct the running of their plant, which they o locate and design as to yield the highest results.

Further, they hire clever solicitors to get business—nen who can so present facts that central-station ervice seems highly to be desired to all who accept heir logic without investigation on their own account.

As a consequence, one after another of the isolated lants is taken over by the central station. And once n isolated plant "gets out of the game," like a noted ugilist, it "never comes back."

A time was when the central-station solicitor was andicapped in his "roundup" of isolated plants by he heating problem.

But this time has passed.

Central-station steam heating is the solution.

Just run a steam main of the required size into the asement at the same time that the conduits are put n and the boilers and the *engineer* as well as the engines an be dispensed with.

So many private plants are going "by the board" hat the matter is becoming one of real concern to the ngineers of the cities.

Except in a few special ases the cost of heat nd power from the cenral station is much highr than that of a private lant properly installed nd operated.

Why, then, are so many rivate plants being bandoned for central tation service?

Is it because central tation service is more eliable?

Or because it is cleaner?

Or because it is safer?

Is it because the owner of the private plant hears only the central-station side of the case?

Or is it because many private plants actually are run so inefficiently that their operation does cost more than central-station service?

It is dangerously easy to neglect the little bits of things—to delay attending to them. Individually, they do not amount to enough to talk about. Collectively, they often mean the success or failure of a plant.

Eternal vigilance is the price of success in any business; in none more so than in steam engineering.

Any engineer who by "falling to sleep" on his job makes it possible for the central-station steam and electric mains to force their way into his plant, not only sleeps himself out of a position but makes the central-station argument against the private plant just so much stronger, and consequently the task of other private-plant engineers so much harder.

On Saturday night, October 29, an open meeting will be held in the Auditorium at the United Engineering Societies building, 29 West Thirty-ninth Street, New York City, under the auspices of the Blue Room Engineering Society.

Two interesting papers will be presented: one on "Isolated Plants versus Public Vendors of Electricity" and the other on "Heat, Light and Power in Buildings."

All who are in any way interested in these subjects are earnestly urged to be present.

The gist of one of these addresses is hereinafter presented. An abstract of the other will appear in a later issue.

The Works of Escher Wyss & Co.

By F. R. Low

A general description of the works at Zurich where Zoelly turbines as well as rotary air compressors and water wheels are manufactured. Some turbine details and a general description of the air compressors.

This works had its beginning in the establishment at Zürich of a small plant for the manufacture of textile machinery by Hans Kaspar Escher in 1805, about the time that John Cockerill was developing the textile-machine works which has grown into the great establishment at Seraing. With the development of the factory system this led naturally into the manufacture of transmission machinery, steam and hot-water heating systems, paper-mill machinery, engines, boilers and water turbines, in the manufacture of the latter of which the works attained especial renown, perhaps by reason of the numerous water powers in Switzerland.

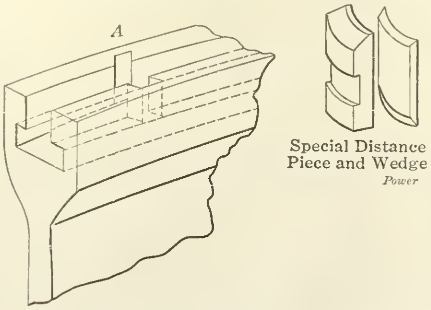


Fig. 1. Section of Runner and Details of Distance Piece and Wedge

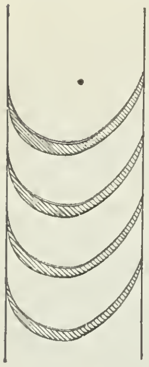


Fig. 2.

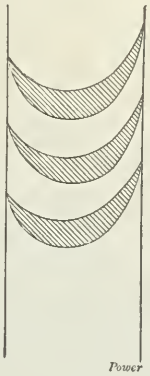


Fig. 3.

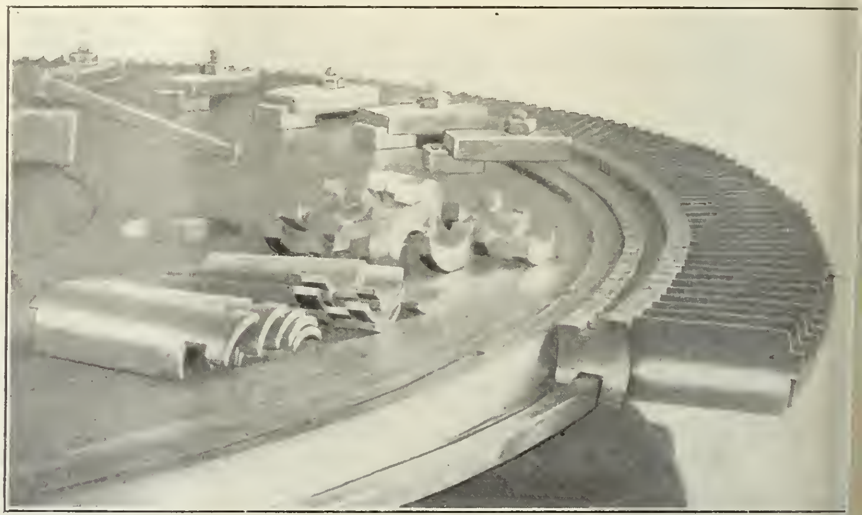


Fig. 5. Old Method of Blading the Runner

The present works includes an area of 100,000 square meters, 42,000 of which are covered with buildings. The compan has also a plant at Ravensburg, Germany All of these large Swiss companies hav German establishments in order to kee their German patents alive, to qualify fo German government work and to avoi duties. Escher Wyss & Co. employ som 1600 hands at Zürich and 600 at Ravensburg.

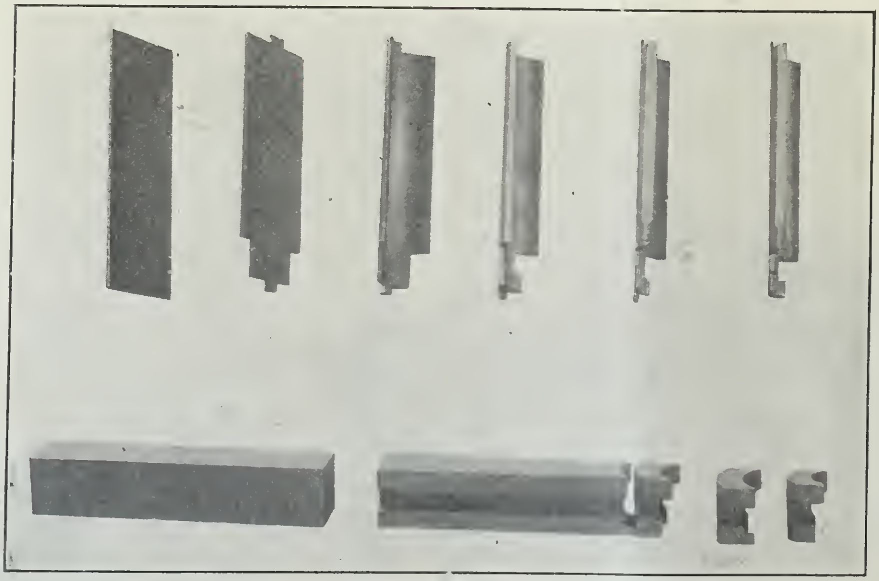


Fig. 4. Successive Steps in the Manufacture of Blades and Distance Pieces

We were welcomed to the Zürich works by Managing Director Zoelly, the inventor of the turbine which bears hi name and which is made not only b Escher Wyss & Co., but by numerou licensees, including the Cramp Ship Building Company in "the States." Th Société Alsacienne, at Belfort, Fran another licensee, is just building thr generating sets, consisting of 22,0 horsepower turbines attached to ge erators of 10,000 kilowatts capacity a capable of carrying a 50 per cent. ov load, for the Compagnie Parisienne Distribution Electrique, at Paris; a Schneider & Co., at Creusot, other censees, are building two sets of the sa size for the same station.

The Zoelly Steam Turbine

Ingenieur Albert Huguenin conduc us through the works and showed us turbines in process of construction. T Zoelly is a multistage impulse turb with a single wheel in a stage and admission. Such a turbine was pate by Réal and Pichon in France in 1827, has been made familiar in America Rateau, from whose turbine it differs in constructional details and in the

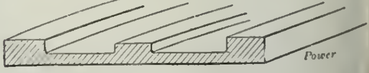


Fig. 6. Special Shrouding for Mar Turbines

that fewer stages are used. The suit for infringement has, however, recently been decided in favor of the defendants. The runners are made of disks of Siemens-Martin steel with a T-slot cut in the circumference, as in Fig. 1.

"Sheet" blades are used in land turbines, i.e., blades of uniform thickness all the way across, as in Fig. 2, while those thickened in the center to make the passages parallel, as in Fig. 3, are used in marine turbines. The former are made of nickel steel on account of the higher steam velocities, and the latter of bronze, which is allowable with the smaller steam velocities. The longer blades are thicker at the base than at the outer end, and the base of each blade, as well as of the distance piece between them, is fashioned into a T to fit the groove in the disk, as shown in Fig. 4, which shows the successive stages of their manufacture. These were formerly set into a built-up disk, as shown in Fig. 5, slotted segments being bolted on to hold the blades in place. In their newer practice, however, a slot is cut in the rim of the disk, as at *A*, Fig. 1, into which the blades and distance pieces are alternately slipped and pushed along. The gap is closed by a special distance piece with a side projection to close the gap, cut away so that at one end it projects into the T-slot and is thus held in place. Between this block and the blade is driven a circular wedge, fitting the face of the blade, which packs the circle of blades and distance pieces closely together, and which is held in by having the face of the contiguous distance piece staked down upon it.

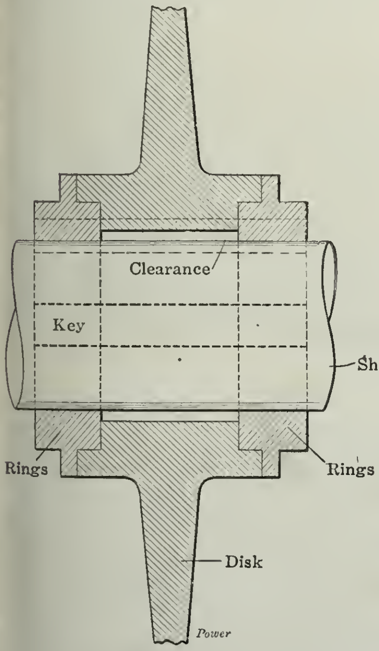


Fig. 8. Method of Attaching Disks to Shaft

The special shrouding for marine turbines has the section shown in Fig. 6 and is punched through the central rib to take the projections upon the tops of the blades which are riveted over. In case of rubbing it is these raised portions which first come into contact. There is, however, from 3 to 6 millimeters (0.12 to 0.24 of an inch) radial clearance so that the danger of contact between the blades and casing is very remote and has never occurred up to now. The axial clearance is about the same. Fig. 7 shows one of the two marine turbines for the German destroyer "G 173," which have given the following results: Load, 7080 brake horsepower; speed, 768 revolutions per minute; pressure, 202 pounds absolute; temperature at inlet, 360 degrees Fahrenheit; back pressure, 4 inches mercury; thermal efficiency, 63.7 per cent.; steam consumption per brake horsepower per hour, 12.8 pounds.

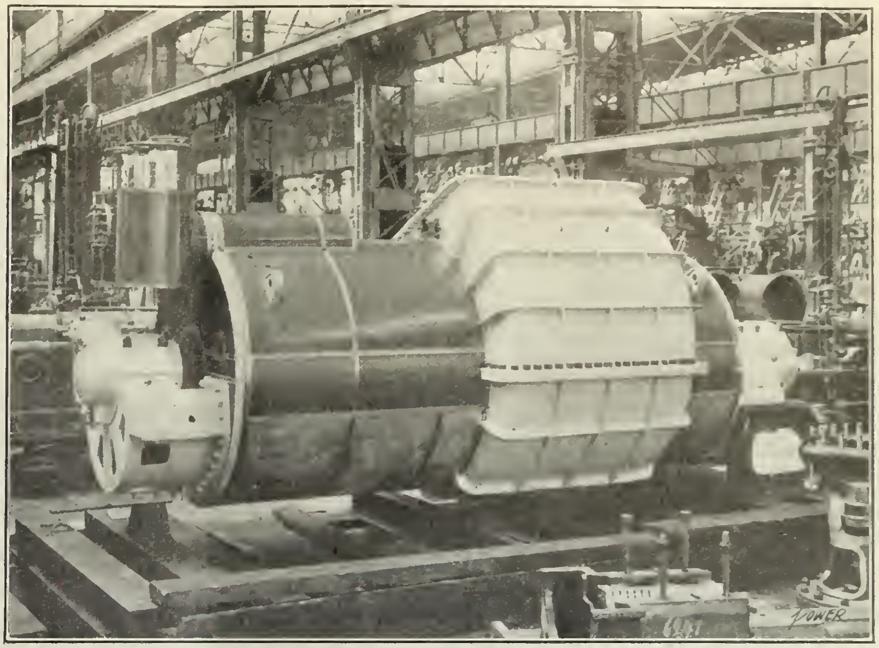


Fig. 7. Zoelly Turbine for German Destroyer "G 173"

The disks are finished all over and carefully balanced and are carried upon rings keyed to the shaft, as shown in Fig. 8. The object of this construction is to avoid the stiffening effect upon the shaft and the raising of its critical speed which would be produced by a continuous line of disk bases keyed thereto. As many of the shafts are run above their critical speeds, it is desirable to keep the latter down and to have as large a working leeway as possible.

As the fall of pressure between the stages is always less than 40 per cent. of the initial, the nozzles formed by the guide vanes must be converging. These are made by casting pieces of sheet steel bent to the right contour, as shown in the sections of the stationary guides in Fig. 9, into the cast-iron disks which separate the chambers. This foundry work is admirably done. Fig. 10 shows the mold containing the steel guide vanes set in the sand. The casting is like the two front sections in Fig. 11, which are placed between the rotary disks in the manner shown in Fig. 12, where the solid black sections are those of the runners and the cross-hatched dividing walls between them those containing the guide vanes, in such a way that the steam which they deliver impinges upon and drives the rotor wheels. Furthermore, the entering edges of the guide vanes are so fashioned as to receive the steam as it leaves the rotary vanes and preserve its residual velocity. See Fig. 9.

The high-pressure steam is received in the circular chamber *A*, Fig. 12, and the greatest allowable ratio of expansion is used in the first stage in order to reduce the pressure as much as possible before it reaches the stuffing box. Carbon rings are used for packing between the first two stationary disks and the shaft, the remaining disks being packed with a number of copper rings let into the disk, forming a labyrinth. Carbon rings are also used in the stuffing boxes.

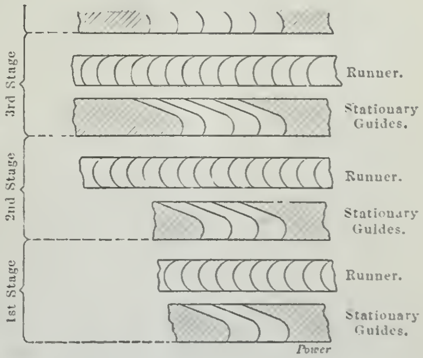


Fig. 9. Diagram of Blades and Nozzles

The fall in pressure between the stages is so divided as to produce veloc-

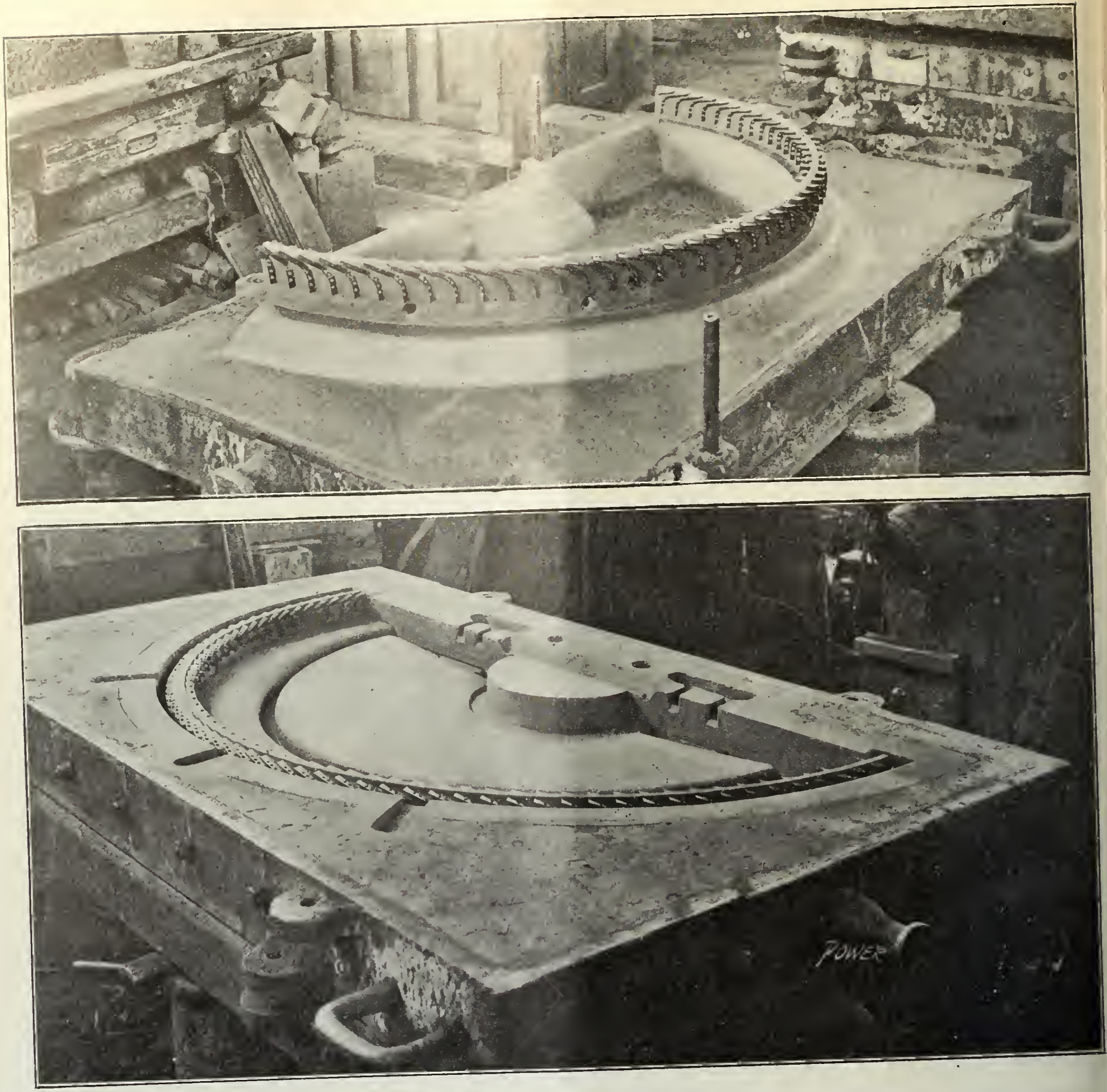


Fig. 10. Guide-wheel Castings in the Foundry

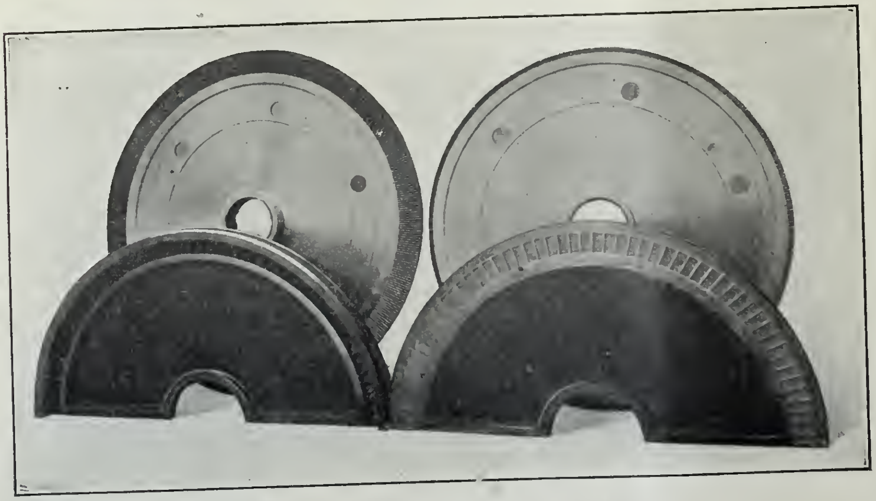


Fig. 11. Runners and Guide Wheels

ities of from 300 to 350 meters (965 to 1125 feet) per second and the blade velocity is from 140 to 145 meters per second, or 460 to 475 feet per second, giving ratios of blade to jet velocities of 0.415 to 0.465. The largest disks are 2200 millimeters (7.2 feet) in diameter at the centers of the blades. These high velocities, the result of considerable expansion ratios between the stages, allow the total expansion to be completed in a limited number of stages. Turbines running at 3000 revolutions, for 200 to 8000 horsepower, are generally built with 8 wheels. Those which run at 1500 turns have 10 to 12, and those for 5000 to 8000 horsepower at 1000 turns 16 stages with modifications, of course, for special cases of initial and terminal pressure. The ratio of expansion never exceeds that which will take place in a conveying nozzle, and the nozzles are made as previously described with converging walls, parallel at the point of issue.

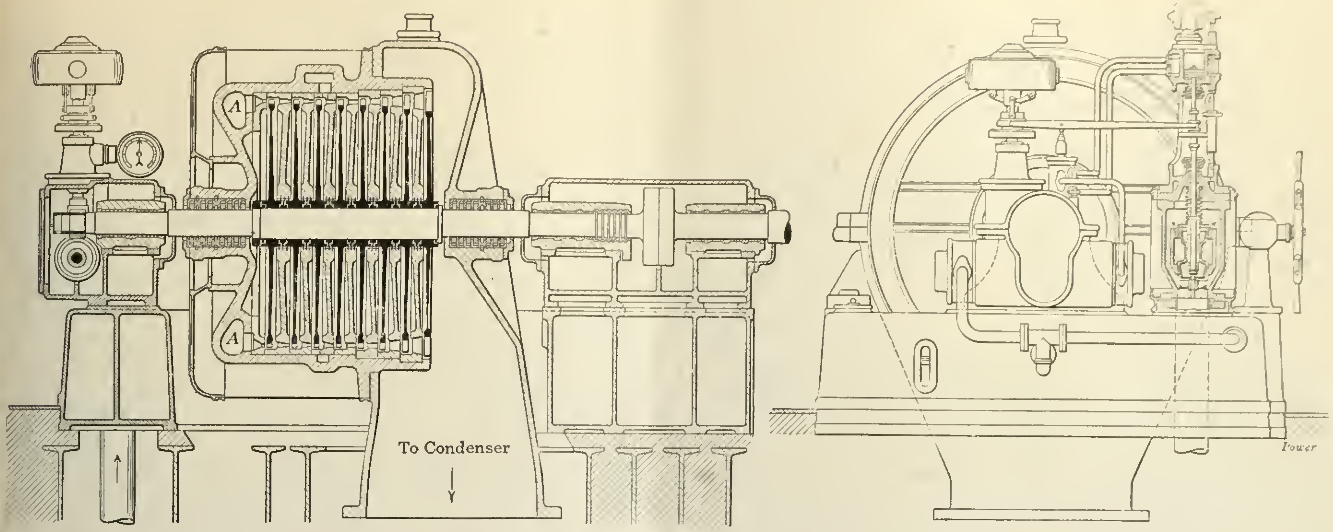


Fig. 12. Sectional and End Views of Zoelly Turbine

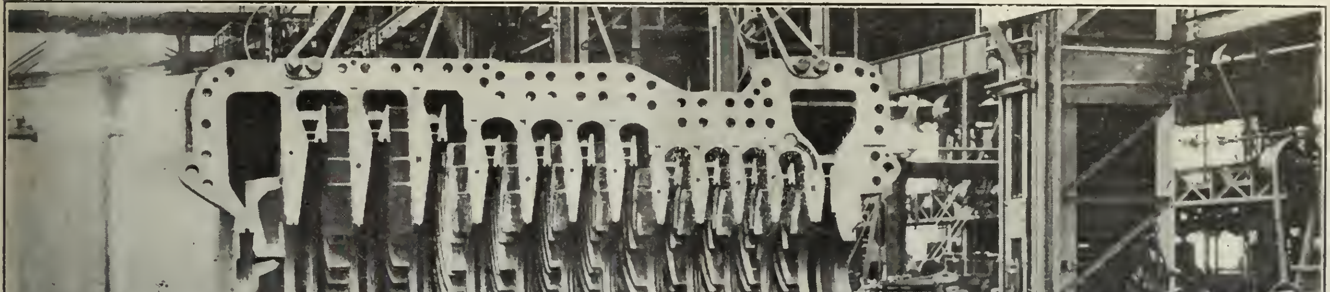


Fig. 13. A 12-stage Rotary Air Compressor

The regulation is effected by throttling, the admission valve being operated by a piston actuated by oil pressure under the control of the governor, through the arrangement, familiar from its use in damper regulators, whereby the movement of the main valve closes the small valve which has been opened by the governor to produce that movement. Oil- and mechanically actuated safety devices are provided which close the admission valve automatically when the speed exceeds a range of 10 per cent. above the normal. A small motor upon the governor allows the speed to be varied 5 per cent. from the switchboard.

Lubricating oil is circulated by means of a pump geared to the main shaft. As this pump will not operate properly until the turbine has attained some 500 or 600 revolutions, a small auxiliary pump, turbine-driven for the larger and hand-driven for the smaller machines, is installed in one corner of the base. The oil-cooling coils are also placed in the bases of the larger sizes, the water being inside the coils. The case is made in two parts with no transverse joints, the single central longitudinal joint being made metal to metal. It is anchored at the low-pressure end, the case as well as the shaft being free to expand lengthwise. These turbines have attained efficiencies of 6 kilograms (13.2 pounds) of steam per kilowatt-hour with steam of 150 pounds pressure, 165 degrees Centigrade (300 degrees Fahrenheit) superheat, and 96 to 97 per cent. vacuum. The largest units yet made at these shops are of 10,000 horsepower, but larger units of the same type are being made elsewhere, as above mentioned.

Rotary Air Compressors

Imagine a hall 30 feet wide, 10 feet high and 100 feet long. A quantity of air which would fill this chamber at atmospheric pressure must be compressed into half that volume, i.e., to a pressure of from 12 to 18 pounds every minute, to furnish the blast for a furnace producing 180 tons of iron in 24 hours. This is usually done by large blowing tubs with reciprocating pistons, but as another instance of the tendency to handle fluids by whirling rather than by squeezing them, by the kinetic rather than by the static method, we were shown a three-stage rotary compressor or blower for direct connection to a steam turbine which will do this work, when running at 2400 to 2850 revolutions per minute, with a weight about one-fifth of that of the reciprocating outfit required to do the same work and using from 1700 to 2200 horsepower. These blowers are made for all sorts of service and it is possible to compress with them to a pressure of 140 pounds with a capacity of not less than 6000 cubic feet of free air per minute. The internal arrangement of a 12-stage is shown in Fig. 13. The air entering at the left is taken in at the base of the tapering wings which run close to but not, of course, in contact with, the tapered faces of the hollow partitions, through which flows the cooling water. The passage contracts toward the points of the vanes, as less section is required to pass the same quantity of air at the increasing velocity. The air is delivered from the tops of the vanes into passages of increasing section formed by the stationary guide vanes seen in the casing, and its velocity changed into pressure. The process is repeated in each stage. The blower illustrated will compress 5300 cubic feet of free air per minute to 85 pounds per square inch when running at 4000 revolutions per minute, requiring 1100 horsepower. These blowers are built with an efficiency of 63 per cent. referred to the isothermal curve.

Waterwheels

For moderate heads, Escher Wyss & Co., furnish turbines of the Francis type. They have ten sets of 5000 horsepower each in hand for Laufenburg-on-the-Rhine, where they have to handle up to 100 cubic meters of water per second through the minimum head of 3.5 meters in each wheel, the normal head being about 10 meters. Numerous other sets are in process for use from Norway to Mexico.

For high heads they use the Pelton impact type. The highest head yet dealt with is that at Adamello, Italy, where under a head of 950 meters (3230 feet) 6500 horsepower is produced by a single jet having a diameter of 130 millimeters (5.2 inches). Six single-jet units of this capacity were furnished. Under this immense head the water passing through the nozzle issues like an immense glass rod and it is difficult to conceive that its mass has absorbed over 6500 horsepower to get it into motion and is ready to deliver that amount of energy when it is brought to rest.

At Rjukanfos, Norway, three units of 14,500 horsepower capacity each have been furnished. Each unit has two wheels and each wheel two nozzles. The head is 250 meters (820 feet).

The buckets for these large wheels are of cast steel, hardened and ground. There were upon the floor stacks of rings having 22 such buckets some 22 inches long and 16 inches wide. With lower-pressure Francis wheels, 85 to 86 per cent. efficiencies are obtained; for the Pelton wheels with very high heads, 80 to 81 per cent. The impact wheel is not always used for high heads. We saw one of the Francis type designed for a head of 140 meters (460 feet) with phosphor-bronze runners and fixed walls for the water channels.

Development of the Blowoff Valve

By Charles J. Simeon

As the invention of the globe valve made the use of higher steam pressures possible it was at once tried for blowoff purposes. In its original form, though suitable for steam, it failed as a blowoff valve, but in a modified form became satisfactory. Its development is traced in this article.

As one of the requisites of a good valve for blowoff purposes is that the passage shall be direct and as free from obstructions as is consistent with right construction, it will be seen that the "Y" type, though made in but few varieties, has much to commend it to the attention of the engineer.

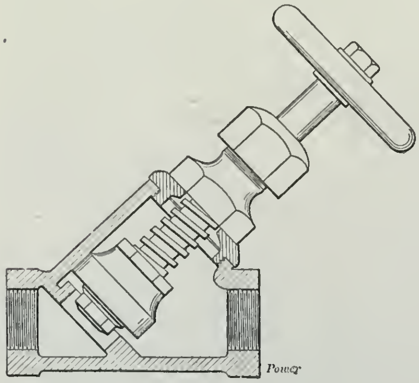


Fig. 1. Jenkins

In the design of all power-plant apparatus the aim has been to avoid complex construction and to make simplicity and directness the desired end. This practice has received especial attention in the matter of the design of piping and valves, particularly if connected with the blowoff, where restricted passages, short abrupt turns and pockets, or where solids or liquids are sure to be productive of annoyance, trouble and possible danger unless more than ordinary care is taken in operation. In many ways the necessity for a free unrestricted passage through the blowoff was demonstrated, and th designer readily adopted any idea that le toward the attainment of this feature.

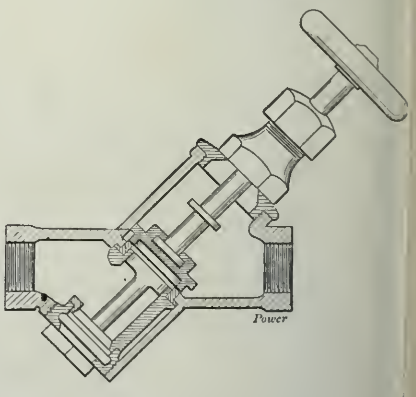


Fig. 2. Bashlin

Beside the cock several types of valve have been adapted to blowoff purpose The first valve proper to be produced w the globe valve invented by Robert Brig in 1863. It was tried as a blowoff valve but was found unsuitable for this purpose owing to scale lodging round the seat and under the diaphragm. The "Y" valve was then tried and is by many considered satisfactory as a blowoff, particularly when combined with the special features of self-washing seats, renewable disks, etc. Besides the comparatively straight and clear passage, this type has the advantage of a seat at such a slant that it is difficult for scale to lodge thereon, and also a recess in the bonnet into which the valve is withdrawn in the open position out of the way of the cutting action of the flow. Examples of this type of valve are the Jenkins, National Tube Company, Williams, Eynon-Evans, Powell, Bashlin and Lunkenheimer "Y" valves, each of which has special features of its own.

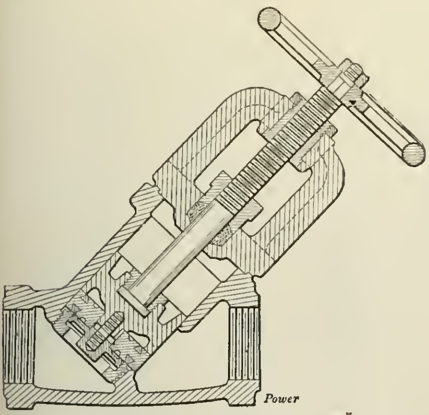


Fig. 3. Lunkenheimer

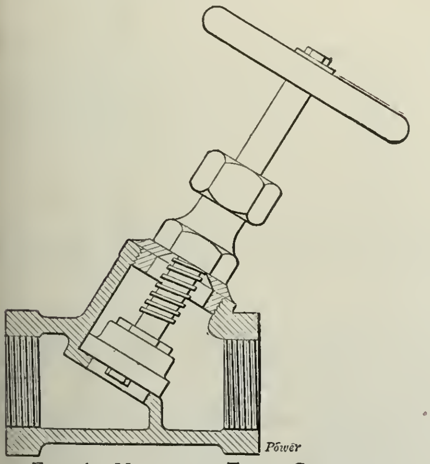


Fig. 4. National Tube Company

The Jenkins has a metal disk into which is set a ring of a composition that readily adapts itself to any irregularities in the seat or pieces of scale, etc., that may have lodged thereon.

The Bashlin has a hard gunmetal disk and a renewable seat whose special peculiarity is that it may be removed and renewed while under pressure. This is accomplished by carrying the seat on a hollow pillar, which may be removed through a hole in the bottom of the body, a flange on the disk furnishing a temporary seat, during its absence. The seat ring is made from copper for pressures above 125 pounds per square inch and of babbitt or asbestos-rubber for lower pressures. The Lunkenheimer company makes two patterns of this type. The regrinding pattern, which has a brass body, the diaphragm of which forms the seat, and a disk of bronze. The other pattern has a bronze seat ring screwed into an iron body, and a heavy reversible disk with babbitt-metal face rings. Both patterns have deeply recessed seats providing the self-cleaning feature.

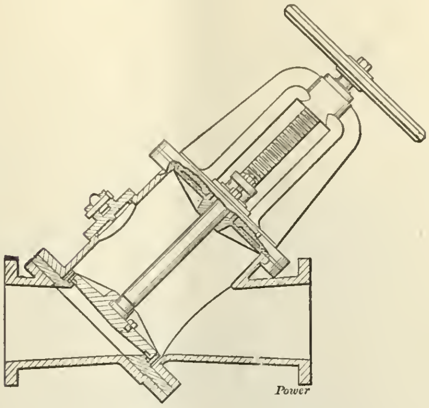


Fig. 5. Eynon-Evans

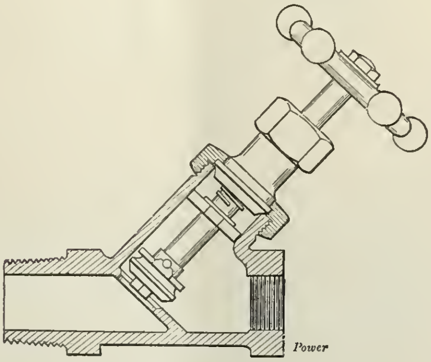


Fig. 6. Powell

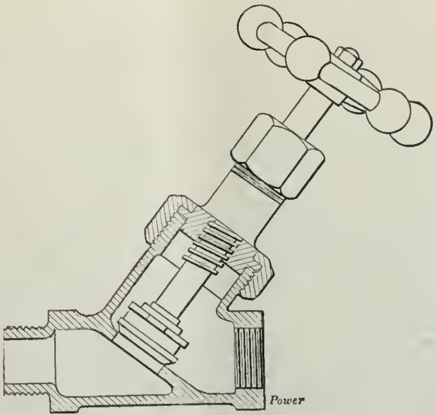


Fig. 7. Williams

Recognizing the simplicity, durability and nonleaking qualities of the plain slide valve under varying conditions of pressure, it was thought that a cover sliding freely over an orifice and forced to its seat when in position would closely, if not quite, approximate correct practice. The gate valve was the outcome of this line of thought and on their introduction into steam practice were at once tried for blowoff purposes on account of the obvious advantage of allowing a practically unobstructed passage when open. The Chapman is a good example. The Johnstone is a modification of the gate principle. It has a parallel-faced disk made of iron and loosely carried in a bronze stirrup. No spring or other mechanical means is used to keep the valve against the seat, the boiler pressure being relied upon to effect this. The seat consists of a cast-iron bush screwed into the body from the outlet side and is enlarged on the outside for either flanged or screwed connection to the discharge pipe.

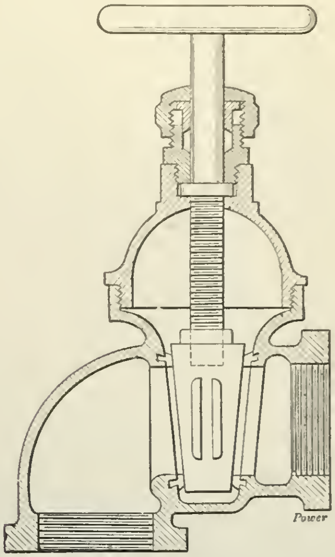


Fig. 8. Chapman

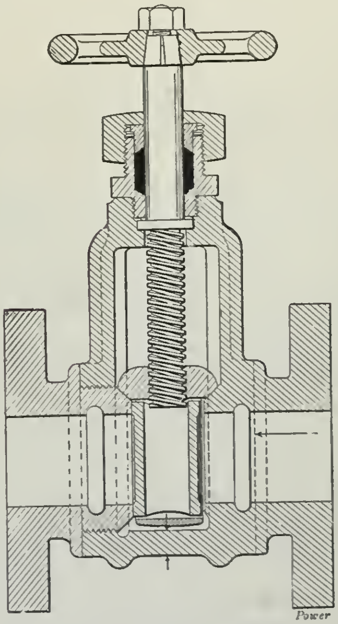


Fig. 9. Johnstone

One objection to a gate valve for blowoff purposes is that the space below the disk is liable to fill with mud or scale and prevent it from closing.

The trouble of a gate valve not closing properly because of mud or scale lodging below the clapper has been guarded against in the Hestberg, in which a door is provided in the bottom of the body, which may be opened while the valve is under pressure, and the accumulation of mud and scale removed. In the Watson & McDaniel Simplex, the disks are carried by a cage or stirrup which has a projection below it through which a hole is bored. When the valve is open, this projection completely fills the space occupied by the disk when closed, preventing sediment from lodging below the valve, while the hole allows a clear passage. The projection also forms a protection for the seat from the cutting action of the flow. Ease of manipulation together with tightness of the valve when closed are insured by using double disks separated by an internal wedge. The valve is operated by a key.

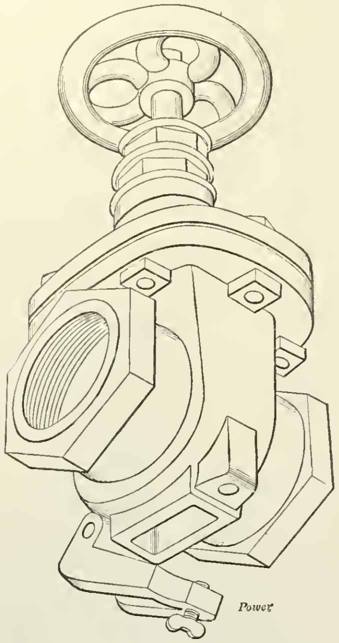


FIG. 10. HESTBERG

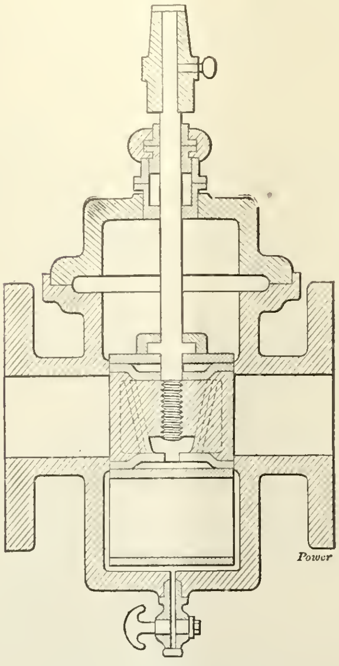


FIG. 11. SIMPLEX

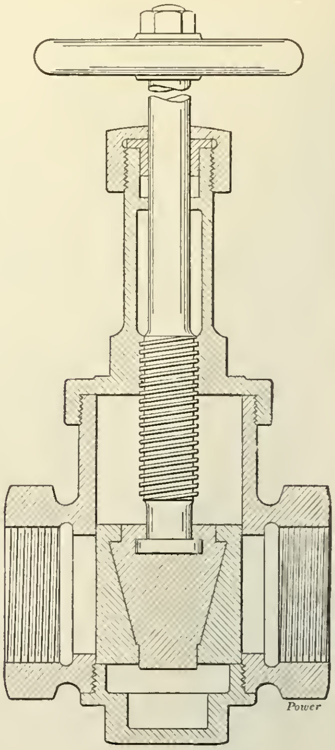


FIG. 12. STAR

The straight-way piston valve is a development of the gate valve. In it a piston or cylinder works in a cylindrical chamber, taking the place of the disk of the gate valve. The Star Brass blowoff has an expanding split cylinder, with a taper plug inside, that renders it easy to operate and insures a tight joint when closed. A plug in the body facilitates cleaning.

The Absorption Refrigerating System

By F. E. Matthews

> *In which the various elements of the absorption system, such as the absorber, generator, analyzer, rectifier, etc., are described and the cycle through which the ammonia passes plainly illustrated.*

A modern commercial absorption machine consists primarily of five parts, three of which are also present in the compression machine. This is shown in Fig. 1, which is a diagrammatic representation of an elementary absorption and compression machine having their condensers and complete expansion sides in common.

In the compression system it has been shown how the low-pressure cold gas returning from the expansion coils enters the suction end of the compressor cylinder, and how, after it has been transferred to the compression end of the cylinder, it is compressed and passed over into the condenser. Reference to the figure will show that in the absorption machine the compressor cylinder is replaced by an absorber, a liquid pump and a generator. In the absorption system the gas, returning from the expansion coils, enters the absorber, corresponding to the suction end of the compressor, is transferred to the generator, corresponding to the discharge end of the compressor, by a pump, through the valves of which it passes just as it flows through the valves of the compressor piston.

In the absorption plant the ammonia (liquor) pump can be made much smaller than the compressor gas pump used in the compression system, because the actual work of compressing the ammonia gas to the point at which it can be liquefied by the cooling water in the condenser is performed by the direct heat of steam rather than by the heat generated by the expenditure of energy behind the compressor piston. To facilitate the use of steam for this purpose, the cold ammonia gas returning from the expansion coils is absorbed or dissolved in water in the "absorber," and the resulting strong aqua ammonia, known as strong liquor, is pumped into the generator or ammonia boiler, where it is heated by a series o steam coils which drive off the gaseou ammonia at a high pressure, just as wate vapor or steam is driven off under hig pressure in a steam boiler.

The high-pressure ammonia gas drive off in the generator of an absorptio plant, like that discharged from the compressor of a compression plant, is conducted to the condenser, at which poin that part of the refrigerating cycle common to both systems begins.

The driving off of the ammonia vapo in the generator changes the strong liquo to weak liquor, which, being under higher pressure than the liquid in th absorber, readily flows back to the absorber. To adapt the elementary absorption machine just described to the requirements of economical commercial apparatus, a number of refinements must b added. Fig. 2 is a diagrammatic representation of a modern absorption machin of well known make.

On account of the similarity of th expansion side of the absorption system to that of the compression syste already described in detail, it will be ne essary to trace the working medium through only that part of the cycle between the expansion coils and the condenser.

COOLER

In the type of machine illustrated in Fig. 2, the expansion side consists of a brine cooler of the vertical cylindrical or shell type, such as is most commonly used in connection with absorption machines, but to some considerable extent with compression machines as well. The brine, usually calcium chloride, is circulated through nests of spiral coils within the cylindrical shell of the cooler, and the anhydrous ammonia is expanded into the space between the coils and the shell.

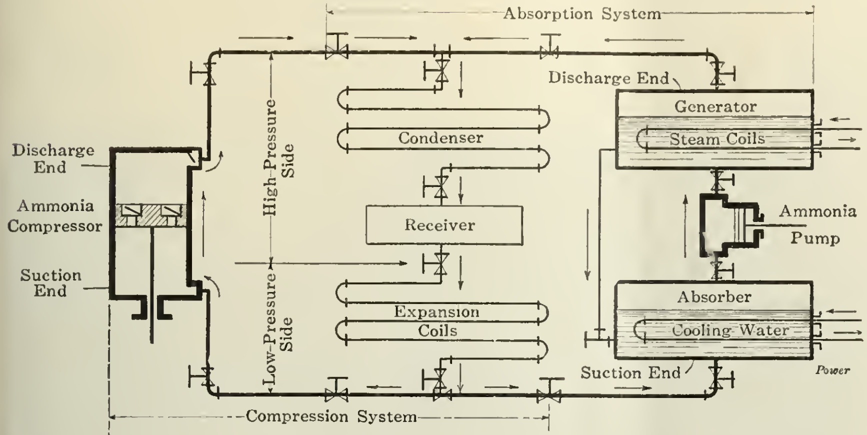


FIG. 1. DIAGRAMMATIC REPRESENTATION OF ELEMENTARY COMPRESSION AND ABSORPTION SYSTEMS

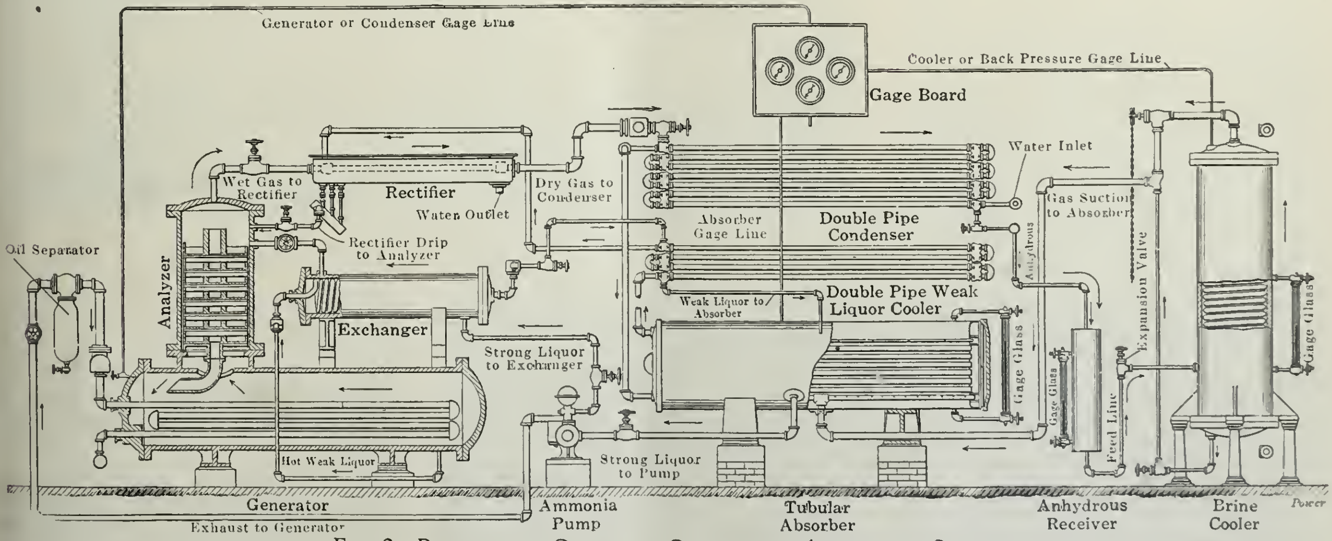


FIG. 2. DIAGRAM OF COMPLETE COMMERCIAL ABSORPTION SYSTEM

The amount of liquid present is readily observed by means of a gage glass. The cold ammonia vapor leaving the top of the cooler passes through the gas-suction line to the absorber, where it is joined by the weak liquor which has just undergone cooling in the "double-pipe weak-liquor cooler. Since the absorption of the ammonia gas into the weak liquor takes place with the evolution of a considerable amount of heat, further means for cooling the liquor must also be provided in the absorber.

ABSORBER

In some cases the absorber is of the double-pipe type, similar to double-pipe condensers and brine coolers. The type here illustrated consists of a horizontal cylinder containing coils of pipe through which the cooling water which has previously done duty in the ammonia condenser passes. The cold, weak liquor is admitted at the top of the cylinder, passed down among the cooling coils, and there mingles with and absorbs the cold ammonia gas from the brine cooler.

As the cooling water has already been heated through several degrees in passing through the pipes of the ammonia condenser, it is expedient that a countercurrent cooling effect may be carried out, in which the warmer, outgoing cooling water cools the weaker aqua ammonia, and the incoming cooler water is employed to reduce the temperature of the strong aqua ammonia on its way to the generator through the ammonia pump and exchanger.

EXCHANGER

The exchanger is a horizontal steel shell capable of carrying the full generator pressure through which the comparatively cool strong liquor is pumped on its way from the bottom of the absorber to the bottom of the generator. This rich liquor is shown entering the shell of the exchanger at the right-hand side, after which it traverses a spiral pipe coil and finally passes out into the top of the analyzer, shown just above the generator. The exchanger is provided with nests of pipe coils connected in parallel, through which passes the hot, weak aqua ammonia supplied to the manifold at the top of the exchanger by a pipe running to the bottom of the generator. In the countercurrent flow the hot weak liquor which must eventually be cooled in descending through the pipes gives up a part of its heat to the cooler rich liquor which must eventually be heated when ascending around the coils. By this heat exchange, cooling water is economized in the absorber and steam in the generator.

ANALYZER

Leaving the top of the exchanger shell, the cool rich liquor passes to the top of the analyzer. Here it is allowed to trickle down over a set of metal trays, where, coming in contact with the ammonia vapors rising from the generator, the countercurrent heat-exchanging effect is still further continued. The hot ammonia vapor, entraining more or less aqueous vapor as it rises from the surface of the liquid in the generator, in passing up through the analyzer on its way to the condenser encounters the descending rain of cool rich liquor on its way to the generator. The former, which must eventually be liquefied in the condenser, is cooled, and the latter, which must eventually be boiled in the generator, is heated. This advantageous heat interchange also has the effect of actually condensing and returning to the generator a large percentage of the entrained aqueous vapors passing off with the ammonia, and also of liberating some ammonia gas through the heating of the gas-saturated rich liquor; where ample analyzer capacities are employed, the incoming rich liquor should be within a very few degrees of the evaporating temperature by the time it finally reaches the surface of the boiling liquid.

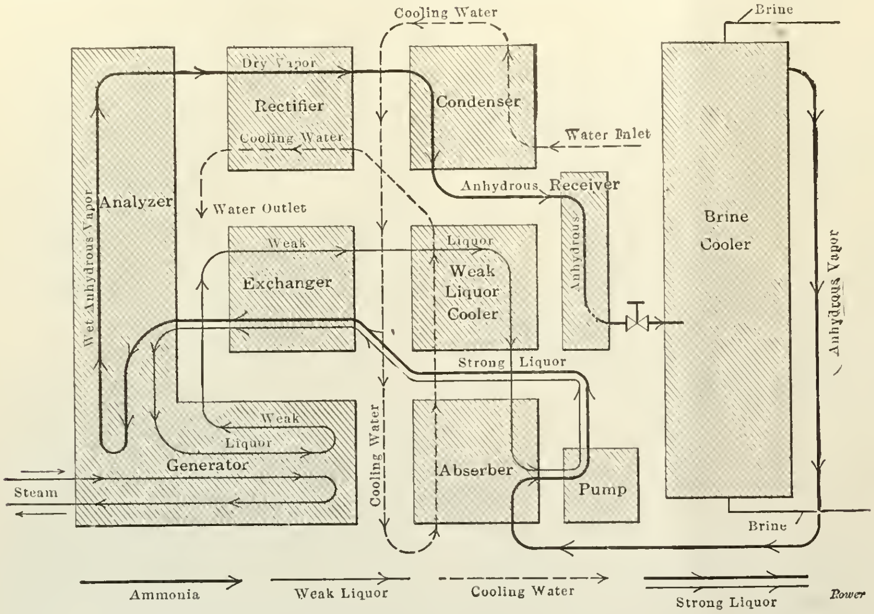


Fig. 3. Diagram Showing Paths of Ammonia, Cooling Water and Ammonia Liquor through Various Members of Absorption System

Generator

The generator, or still, as it is frequently called, is the ammonia boiler for the evaporation of the weak liquor enriched and changed into strong liquor by the absorption of ammonia gas direct from the expansion coils, in the absorber. It consists of a substantial steel shell provided with a heavy cast-iron head through which pass the ends of the steam coils provided for supplying the heat required for driving off the ammonia. The strong aqua ammonia enters at the top, where it remains by virtue of its specific gravity being lower than that of the weaker liquor at the bottom of the generator. The water of condensation from the steam used in the generator is usually returned to the boilers. This can be effected either by an automatically controlled pump, or by a suitable high-pressure trap.

Rectifier

After leaving the analyzer, the hot ammonia gas passes to the rectifier, a water-cooled coil of pipes of sufficient area to insure the condensation of any aqueous vapors that may have escaped condensation in the analyzer. The liquid condensed in the rectifier is rich, saturated liquor which is returned to the generator by way of the analyzer.

Condenser

From the rectifier the ammonia gas, which should now be practically free from aqueous vapors, is passed to the condenser. Condensers for absorption machines like those for compression machines may be of either the atmospheric double-pipe or shell type as preferred. The anhydrous ammonia liquefied in the condenser passes to an anhydrous receiver, similar to those used in the compression system, from which it is drawn as required for expansion in the brine cooler, its flow through the feed line being regulated by the usual expansion valve. After evaporation in the brine cooler the ammonia gas again passes to the absorber, after which the working cycle is repeated.

Cycle Traversed by Ammonia

This can be readily traced by following the course of the heavy arrows in Fig. 2. The circuits of both the gaseous and the aqueous components of the aqua-ammonia refrigerant, as well as that of the cooling water, can be more readily followed out by means of the diagram, Fig. 3, in which all mechanical details have been omitted, and the several members of the refrigerating system illustrated in Fig. 2 are represented by shaded areas occupying approximately the same relative positions on the diagram. The path of the ammonia is represented by a heavy solid line; that of the water component of the aqua-ammonia refrigerant by a narrow solid line; and that of the cooling water by a broken line. The direction of travel in each case is indicated by arrows.

From this diagram as well as from Fig. 2, it will be seen that, as "anhydrous ammonia," the refrigerant starting from the "anhydrous receiver" passes to the "brine cooler," where in changing to the gaseous state it performs its sole function of absorbing heat from the brine. As saturated low-temperature ammonia vapor, the refrigerant starting from the brine cooler passes to the absorber, where it enters into solution or is absorbed by the weak liquor from the generator, forming strong liquor. As hot strong liquor the refrigerant starting from the absorber passes through the exchanger, where it gives up some of its heat to the weak liquor on its way to the absorber, then on by way of the analyzer into the generator, where the ammonia gas is driven out of the strong liquor solution, under high pressure, by the application of heat, and passes through the analyzer and rectifier into the condenser, leaving the impoverished aqua ammonia or weak liquor behind in the generator.

In the condenser, the heat originally absorbed by the anhydrous ammonia in changing from the liquid to the gaseous state in the brine cooler, as well as that added to increase its temperature and drive it out of solution in the generator, is given up to the cooling water, circulated through the condenser, causing the ammonia to return to the liquid state, after which it flows to the anhydrous receiver, and the cycle is again traversed.

The aqueous component of the aqua-ammonia refrigerant, starting from the bottom of the absorber in company with the ammonia in the form of strong liquor, passes through the exchanger and analyzer into the generator. Here it is separated from the greater part of the ammonia and returns through the exchanger and weak-liquor cooler to the absorber. Here it again joins the anhydrous ammonia, forming strong liquor, and retraces the path just described.

Path of Cooling Water

The cooling water is admitted first to the ammonia condenser, where it performs its most important function of removing heat from and liquefying the ammonia gas. After leaving the ammonia condenser it is still cool enough to be capable of absorbing a considerable amount of heat from the strong liquor in the absorber, more from the weak liquor fresh from the generator in the weak-liquor cooler, and still more from the hot ammonia gas fresh from the generator in the rectifier, after which it usually passes to waste.

Still another line might have been drawn on the diagram in Fig. 3, indicating the path traversed by the heat from the point of its absorption from the brine in the brine cooler to that of its expulsion with the cooling water from the condenser. Such a line, however, would coincide with that representing the ammonia from the point where the heat and the vapors of the refrigerant leave the brine cooler, continuing to the condenser, where it would cross over and join that representing the cooling water. It would then follow this line through its circuitous passage to the point where, together with the water, the heat flows away to the sewer.

It should be noted that throughout the entire system, a countercurrent effect is carried out between the cooling and the cooled substances.

By these countercurrent-cooling effects in which the coldest cooled substance gives up its heat to the coldest cooling substance, and the hottest cooled substance to the warmest cooling substance, the outgoing substance is cooled more nearly to the temperature of the incoming cooling substance than would otherwise be possible, thus effecting economy not only in the amount of the cooling substance required but also in the operation of the system through the reduction in the amount of the refrigerating medium required for a given amount of cooling.

A Boiler Explosion in Hungary

By S. Hurley

The accompanying photographs show two views taken immediately after a boiler explosion that recently occurred in the town of Szeged, Hungary, and which resulted in killing ten men and injuring fifteen more. The boiler that exploded was one of two employed in a match factory and was a water-tube boiler 3½ to 4 feet in diameter, made of ⅜-inch plate and normally carrying a gage pressure of 117 pounds per square inch. It was torn in three pieces which were thrown about the neighborhood by the force of the explosion. The front end was separated from the body and hurled across the adjacent street on to the roof of a house, and the rear end, weighing about 700 pounds, was shot up into the air about 60 yards, and fell into a military yard 250 yards distant. The middle section, or shell, was thrown across the street in the same direction with the front end, but landed in a vacant lot.

The force of the explosion was so great that the second boiler, which was exactly like its mate, was thrown bodily through the wall of the building out into the street, where it is shown in Fig. 1 with the wrecked boiler house. Fig. 2 shows the front end of the wrecked boiler lodged on the house top where it was thrown.

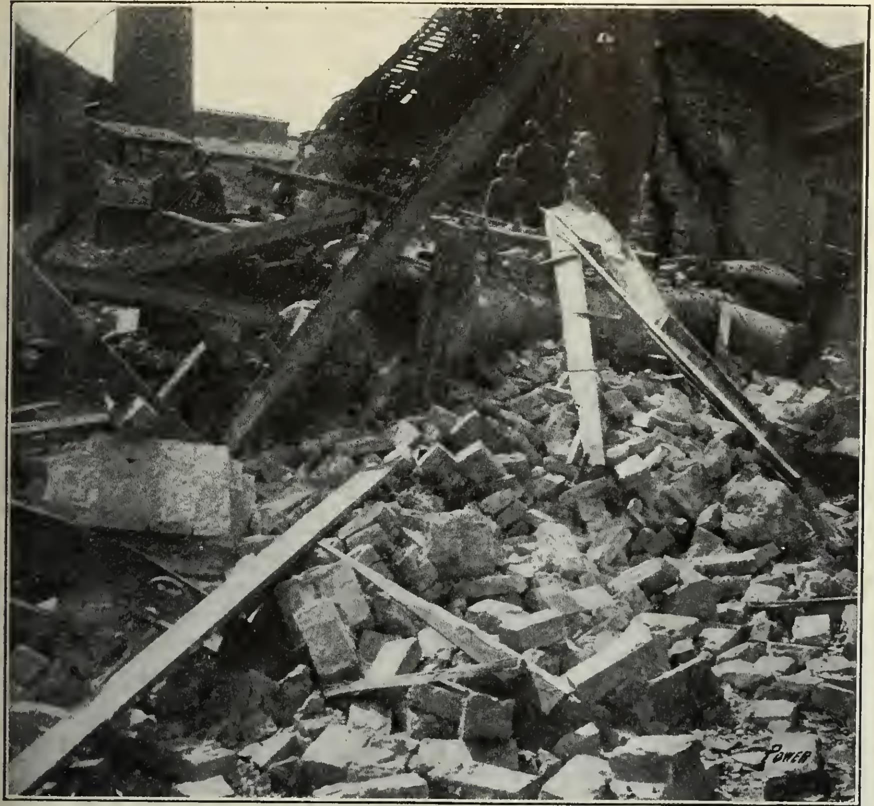


Fig. 1. The Boiler House after Explosion Occurred

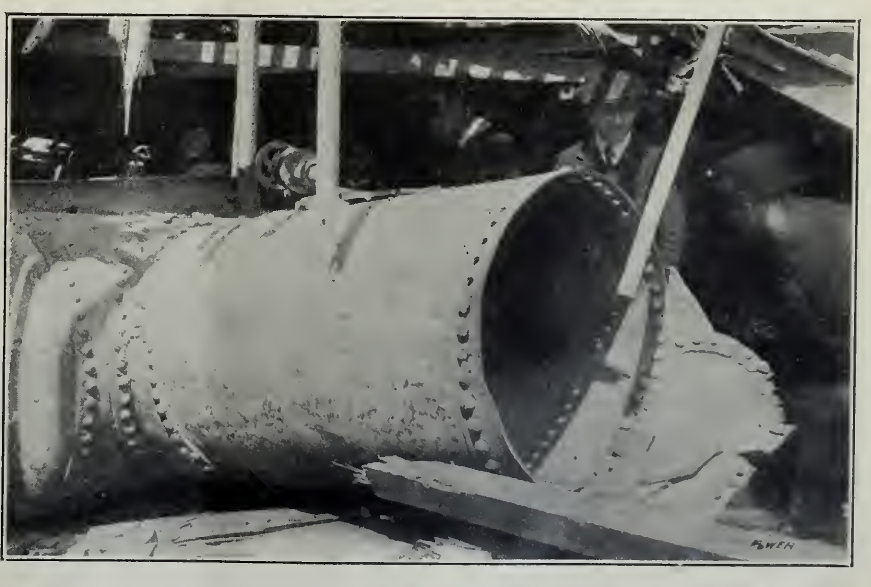


Fig. 2. Front End of Boiler Lodged on House Top

The cause of the disaster has not been very well established, but the boiler was evidently not in good condition, and a small rise in pressure was enough to rupture it. This was shown by the condition of the torn plate along the line where the end shown in Fig. 2 was separated from the main shell. Here the plate was transversely ruptured, just outside the row of rivets, for a distance of 18 inches, and along this entire break but 2 inches of sound metal was found. This would certainly indicate that the boiler was not in fit shape for use. Before the explosion occurred the engines were stopped to put on a belt and it is claimed that the accident was a result of the rise of pressure due to the closing of the throttle valves. This explanation would probably stand for some discussion because if the safety valve was working properly it should have taken care of any excessive increase of steam pressure. Any suggestions along this line by Power readers would be appreciated.

In case of foaming, close the throttle long enough to show the true water level in the boiler. If the level is sufficiently high, feeding and blowing down will usually correct the evil. If foaming becomes violent, check the draft and use the surface blowoff freely. Ascertain the cause and take steps to prevent the evil.

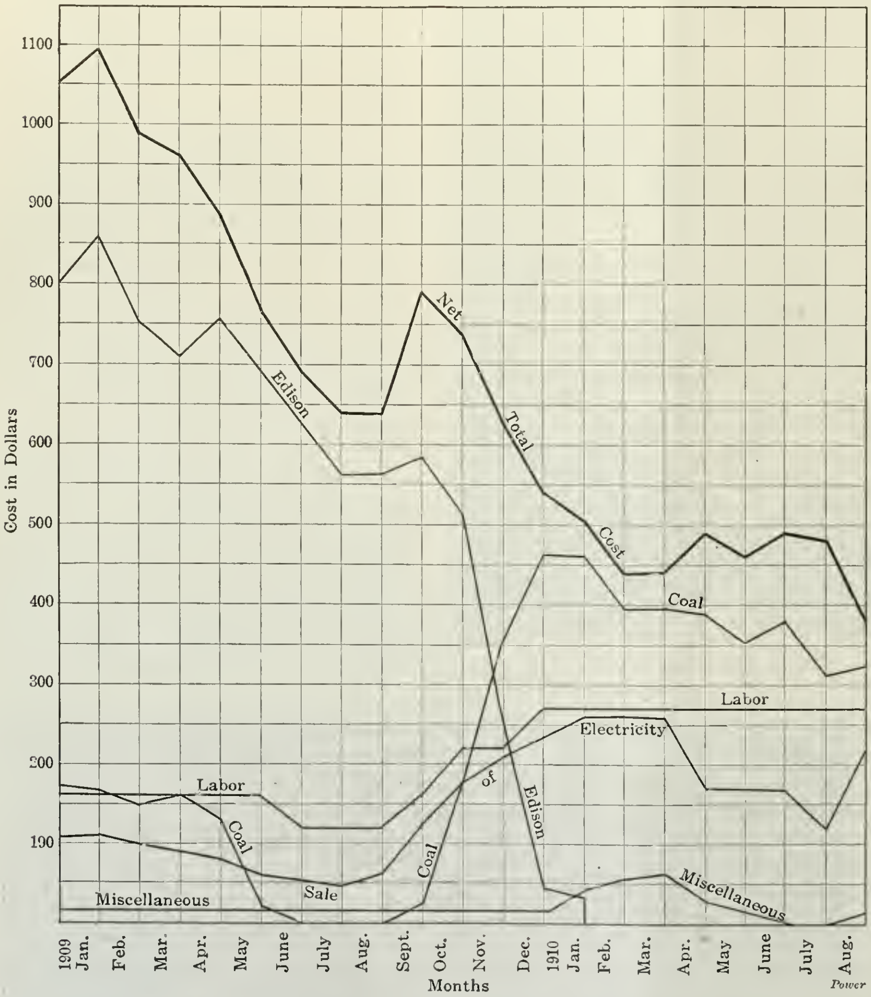
The Cost of Heat, Light and Power*

By C. M. Ripley

By quoting the actual operating costs for heat, light and power, in a number of New York office buildings, apartment houses and hotels, the author is able to present a strong argument in favor of the isolated plant.

In the purchase of lamps, brooms, soap, labor, etc., the shrewd buyer is always prepared to prove that if he is paying more than the other fellow "per unit," he nevertheless is paying less "per year." A discussion of unit costs of heat, light and power is still more misleading unless further evidence is provided regarding total yearly costs. In buildings containing independent electric plants, these items are so closely intertwined that the total yearly cost is the final test.

In the following discussion the total yearly operating cost is given for heat, light, power and engineers' services for repairs, etc., in a number of buildings in New York City, which information has come first hand and is authentic. The term "cost of electric plant" refers to the extra engineering investment in excess of the cost of a heating plant only. Depreciation is figured at 5 per cent. and interest at 5 per cent. on the original investment.



CURVES SHOWING COMPARATIVE COSTS AT THE MONOLITH BUILDING

OFFICE BUILDINGS

The Terminal building, at Park avenue and Forty-first street, is twelve stories high and contains an electric plant which cost $12,000. This building was formerly on Edison service. During the past year all the offices and stores have been rented and the electrical output has increased to an annual total of 156,340 kilowatt-hours. From October, 1909, to June, 1910, inclusive, 125,430 kilowatt-hours were produced with a total fuel bill of $2528. During the preceding period from October, 1908, to June, 1909, inclusive, the electrical output was only 91,770 kilowatt-hours and the coal bill was $2145. This shows what effect the filling of the building had upon the operating costs, the increased output of 33,660 kilowatt-hours costing only $383 in coal, as the fixed charges remained the same.

Following is the summary of the operating costs for light, heat and power during the past twelve months:

Screenings @ $2.22 per ton	$3098
Wages	3217
Water for boiler	100
Repairs, oil, ash removal and miscellaneous (not including lamps)	851
Total	$ 7266
Less sale of electricity	400
	$ 6866
Ten per cent. charges against plant	1200
Net total cost	$ 8066

Had the electricity been purchased from an outside source and the heat been produced on the premises, and men employed for repairs, etc., the cost would have been as follows:

No. 1 buckwheat coal, @ $3.35 per ton	$ 2,000
Fireman, eight months	400
Engineer and building manager	1,500
Electricity, 156,340 kilowatt-hours @ 5c. per kilowatt-hour	7,817
Care of elevators and repairs to heating system, plumbing and wiring	300
Oil and grease	50
Total estimated cost	$12,067
Less sale of electricity	400
Net total cost	$11,667

This shows a saving of $3601 attributable to the independent electric plant, which means that the plant will pay for itself in about three years. It means that a capital of $12,000 is invested and earns $3600 a year, or 30 per cent. per annum; and this, after paying 5 per cent. interest on itself and 5 per cent. for depreciation.

The curves in Fig. 1 show the operating costs for the past twenty months at the Monolith building. Three different kinds of operation are shown during this period. From December, 1908, to October, 1909. the building was on Edison service exclusively. From October, 1909, to January, 1910, they were using some Edison and some of their own electricity. Durin the remainder of the time they were usin their own electricity, exclusively. Th heavy black line at the top represents th total net cost, and it is obtained by add ing all operating expenses for the perio and subtracting therefrom the entire in come from the sale of electricity t tenants and neighbors. It is this differ ence month by month that locates th points on this curve. The curves shov the variations in the different items o expense, month by month, and that a ne saving on the manager's books of betwee $450 and $600 per month has been el fected. The itemized costs of the mai part of the electric plant are as follows

Three Ridgway engines and dynamos	$55
Switchboard	9
Storage battery	30
Electric meters	3
Total	$97

On lower Broadway, there is an offic building twelve stories high and coverir an area of 50x150 feet, which is equipped with an electric plant delivering 117,235 kilowatt-hours per year, although the building is only about half rented. Of this amount, 38,755 kilowatt-hours are for power and 78,480 kilowatt-hours for light. The first cost of the entire engineering equipment of the building was,

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*From an address to be delivered before the Blue Room Engineering Society of Greater New York on October 29.

Three Ames engines	$4 770
Three dynamos	3,600
Wiring of entire building	7,100
Boilers and piping for the entire building	20,400
Storage battery	3,000
Switchboard	1,600
All mason work	2,000
Stack	1,100

The operating costs per year are,

No. 1 buckwheat coal, @ $3.35 per ton	$2,214
Labor	3,329
Miscellaneous, including lamps	1,598
	$7,141
Receipts from tenants for electricity	1,710
	$5,131
Fixed charges, 10 per cent. of plant cost	1,400
Total net yearly cost	$6,831

At the market rate of 5 cents per kilowatt-hour, the electricity alone is worth

117.235 × $0.05 = $5862.

Coal and labor for heating would easily amount to $2400 more and a skilled engineer is necessary to repair elevators, etc., whether electricity is made or bought. Therefore, this plant is already saving almost $3000 per year, and when the building is fully rented the power plant will make a record to be proud of.

The Security Mutual Life Insurance Company's building, at Binghamton, N. Y., contains two hydraulic elevators and an electric plant which delivers 340,000 kilowatt-hours per year. The costs for the year's run during 1909 were as follows:

No. 1 buckwheat coal @ $3 per ton	$4062
Wages	3208
Miscellaneous	800
	$8070
Sale of electricity	600
	$7470
Ten per cent. fixed charges	1500
Total operating cost	$8970

It is estimated by the consulting engineer that it would cost $3200 per year in coal and labor to run this building in case electricity were purchased from an outside source. If this estimate is correct, the 340,000 kilowatt-hours are generated at a cost of 1.6 cents per kilowatt-hour.

In New York City there is an office building partly occupied by the owners, who are publishers of an evening newspaper. This building uses less electricity than the Security Mutual Life Insurance building, and yet it pays $12,000 per year for electricity alone—quite a contrast to less than $9000 per year for heat, repairs and electricity, including fixed charges, which is the expense at the former building.

The total comparative costs are as follows:

New York newspaper building, for heat and electricity per year	$23,000
Security Mutual Life building, gross cost heat and more electricity per year	9,570
Difference	$13,430

It should be remarked, however, that in the case of the former the labor charges are high because of the newspaper-union conditions; also the New York building has a greater number of hydraulic elevators. Yet, when considering the cost of electricity alone in the New York building it will be found to be greater by 25 per cent. than the cost of heat, light and power at Binghamton; hence the labor and coal prices are not large factors.

LOFT BUILDINGS FOR LIGHT MANUFACTURING

The Langedorf building is twelve stories high and occupies a plot 100x100 feet. The building originally had Edison service. A plant was designed consisting of two units, only one of which was installed at first to demonstrate the facts. The original installation, with provisions for the additional unit, cost about $8300 and consisted of a Corliss engine belted to a 125-kilowatt dynamo. Blowers were installed to facilitate the burning of refuse and sawdust from the wood-working plant. From July, 1909, to June, 1910, the cost of heat, light and power was as follows:

Screenings @ $2.22 per ton	$3178
Labor	2929
Ash removal, repairs and miscellaneous	950
Water from deep wells	000
	$7057
Less sale of electricity	2855
	$4202
Ten per cent. charges on equipment	830
Total net cost	$5032

This represents the cost of producing 180,910 kilowatt-hours, heating the building, furnishing live steam, repairing elevators, repairing the plant, cost of oil and miscellaneous, but not including lamps.

The following is the estimated yearly cost if electricity had been purchased:

No. 1 buckwheat coal @ $3.35 per ton for heat, hot water and live steam	$ 1,550
Chief engineer and superintendent	1,200
Fireman	660
Edison electricity, 180,910 kilowatt-hours @ 5 cents per kilowatt-hour	9,045
Care of elevators, repairs, oil and grease	350
Total	$12,805
Less sale of electricity	2,835
Net total cost if buying electricity	$ 9,970
Actual total cost when generating electricity	5,032
Difference in favor of generating electricity	$ 4,938

Here is an $8300 investment saving $4900 per year, or nearly 60 per cent. after it has saved money enough to pay interest on itself and 5 per cent. depreciation.

Late this past summer authorization was given for the completion of the plant with the addition of a 150-horsepower Brownell engine and an 80-kilowatt dynamo. These will cost approximately $2800 and are to be kept in reserve for an emergency. This building is a typical New York City loft building and contains a complete printing establishment, making a specialty of night work and delivering on the judge's desk at 9 a.m. the complete testimony and court proceedings of the day before. It also contains woodworking machines, buffing machines, hot tables and glue pots requiring live steam.

A nine-story loft building at 213 Grand street has two elevators and a small electric plant, the cost of the equipment being as follows:

Electric wiring for entire building	$ 850
One Ridgway engine	1135
One 75 kilowatt Sprague dynamo	1182
One 125-horsepower Titusville boiler	875
Steamfitting, including pumps, radiators, etc.	4950
Meters and later installations	213

This plant began operating last February and up to September 1, 1910, during part of which time the building was only partly rented, the operating costs were as follows:

Screenings @ $2.06 per ton	$ 596
Labor (the engine-room force operates one elevator)	879
Edison service	150
Repairs, ash removal, water and miscellaneous	450
	$2075
Ten per cent. charges on equipment	500
Total cost	$2575

During this time the receipts from tenants were $2803, including two tenants who purchase live steam the entire year at $160 per year. Therefore, the net operating cost for the first seven months has not only been a negative quantity for furnishing heat, hot water, public lights and electric power for pumping and for two elevators, but has actually shown a profit of several hundred dollars. As the electric output increases, this record should become even more favorable.

The Weil & Mayer buildings, at 580-590 Broadway, consist of three twelve-story loft buildings containing a total of seventeen electric elevators. The record of these buildings illustrates the advantage of supplying neighboring customers with heat, light, power and hot water. The actual cost of producing heat, light, power and live steam for twelve months were as follows:

Screenings @ $2.50 per ton	$ 6,830
Wages	8,617
Water for boilers from deep wells	000
Repairs, ash removal, oil, miscellaneous	1 881
	$17,328
Less sale of electricity	13,336
	$ 3,992
Ten per cent. charges against plant	2,000
Net total cost	$ 5,992

This amounts to but little over $150 a month for each building for complete service of heat, light and power for pumping and elevators. The electrical output for the year was 402,000 kilowatt-hours. The following figures are a fair estimate of the cost of heat, light and power had the buildings been operated together and electricity bought from an outside source:

402,000 kilowatt-hours	$18,780
No. 1 buckwheat coal @ $3.35 per ton for heat and live steam	4,500
Wages, including fireman	1,100
Oil, grease, ash removal, repairs, miscellaneous	1,000
Total	$28,380
Less sale of electricity	13,336
Net operating cost when buying electricity	$15,044
Actual net operating cost when producing own electricity	5,992
Saving by producing own electricity	$ 9,052

If the three buildings had been operated separately, the following estimate would have applied:

No. 1 buckwheat coal @ $3.35 per ton for each building	$ 1,500
Wages, including fireman	1,800
Electricity for each building	6,700
Repairs, oil, etc	400
Total for each building	$10,400
Total for three buildings	31,200
Less sale of electricity	13,336
Net total cost for three buildings operated separately, if buying electricity	$17,864
Actual net total cost producing electricity and operating together	5,992
Saving by operating buildings together and producing own electricity	$11,872

The conclusion to be drawn from these figures is that by operating the three buildings together $2820 is saved yearly, and by producing their own electricity an additional yearly saving of $9052 is effected. The saving of $9052 is equivalent to the annual income on a capital of $180,000 invested at 5 per cent., yet the actual cost of the plant was only $20,000; and being installed twelve years ago it paid for itself in two and a half years. Also, the annual earning power of the plant is about 45 per cent. per annum on the investment.

These records show that the first investment was paid back, in savings effected, nearly ten years ago, and yet in order to be perfectly fair I am still charging against the record of the plant 5 per cent. interest on the investment. Furthermore, although the plant has paid for itself many times over, I still charge against the record of the plant another 5 per cent. for depreciation. The plant is in excellent condition today after twelve years of service and will probably last another ten years.

Apartment Houses

The Hendrik Hudson apartment house on Riverside drive is nine stories high and contains seventy apartments. Being 200 feet on the side facing the river it is exposed to the cold winds from that direction. The boilers are used for heating and in the coldest days for pumping also. All the electricity is furnished by the New York Edison Company.

It has been found that central-station electric service is more profitable in a medium-sized apartment house without a refrigerating plant than in any other type of building in New York City, except the theater. This is because the cost of operating the plant to provide a 24-hour service is high and yet the consumption of electricity is much smaller than in an office building of the same size with a 12-hour service. Also, the consumption is much less than in a hotel of the same size, as is also the elevator travel. In an apartment house the income for light, etc., from the tenants is almost negligible during the summer months where the tenants go away for the summer.

From August, 1909, to July, 1910, the operating costs at the Hendrik Hudson were:

Screenings @ $1.98 per ton	$ 2,185
Wages	2,233
Edison service	5,105
Miscellaneous, including lamps	1,149
Total operating expense	$10,672
Less sale of electricity	4,059
Total net operating expenses	$ 6,613

The total consumption of electricity was 102,103 kilowatt-hours. It is necessary to use 100,000 kilowatt-hours per annum in order to obtain the 5-cent rate in New York, and the custom is to sell to the tenants at 10 cents per kilowatt-hour. In winter the pumping is done largely by steam, the pressure being kept at about 60 pounds. The exhaust is used to help heat the building besides furnishing hot water for about one hundred and fifty bath rooms. The fuel costs $1.98 per ton and in addition the garbage and refuse are burned under the boilers.

Hotels

The Hotel Hargrave covers an area 50x200 feet, is twelve stories high and contains three hundred guest rooms, two hundred baths, a restaurant, four electric elevators and complete ice and electric plants, but no laundry. Being a hotel, there is a 24-hour service. The electric plant, including a storage battery, cost $13,063. The operating costs for the past year were:

Screenings @ $2.22 per ton	$ 6,237
Wages	4,895
Boiler water (estimated)	150
Ash removal	120
Repairs, oil, lamps and miscellaneous	1,400
Total	$12,802
Ten per cent. charges against plant	1,306
Total net operating cost	$14,108

This is the cost of producing 289,877 kilowatt-hours of electricity, ice, refrigeration, heat and hot water.

The estimated cost of running this hotel in case electricity were purchased from an outside source is as follows:

Edison service, 289,877 kilowatt-hours @ 5 and 4½ cents per kilowatt-hour	$13,944
No. 1 buckwheat coal @ $3.35 per ton for heat, ice and hot water	4,500
Wages	3,480
Repairs, ash removal, oil, water, etc	600
Total net operating cost if buying electricity	$22,524
Net total cost when making own electricity	14,108
Saving by making own electricity	$ 8,416

Thus the plant paid for itself in less than two years, or looking at it in another way, it pays 65 per cent. per annum on the investment.

It happens that when this hotel was half its present size it did not have any electric plant, and our office is in possession of the records of operating expense for the ice and heating plant before it was supplemented by the electric plant. It is from these records that I estimate the cost of the ice, heating and hot-water service.

Increased Income from Engine Rooms

In New York City in the past it has been difficult for the owners of one building to sell electricity to their neighbors, on account of the threatened classification of the building as a central station by the insurance companies and the resulting increase in insurance rates. For instance, two buildings located side by side and separated by a party wall were owned by one man. An electric plant was installed in one building from which both buildings were supplied with heat, light, power and hot water. Later, one of the buildings was sold, and just because the title had changed hands, a move was made to increase the insurance rates on the presumption that the fire hazard had been increased.

Besides selling electricity to the tenants, a Broadway apartment house with an electric plant is getting $1800 per year by selling exhaust steam to a neighboring building. Another apartment house on Park avenue is heating every house on the block and deriving a good profit.

It is easy to add a few thousand dollars to the annual income of a building if it has an independent electric plant. If it is thought that the engine, dynamo or boiler capacity is not large enough to do this, tungsten lamps may be installed throughout, and the facilities thereby almost trebled. Tungsten lamps are a good investment in almost any building, especially if they are rated three or four volts higher than the actual voltage.

The United States Geological Survey, at Washington, has been testing fuels for several years and will gladly send its literature upon request. The gist of the report for the building manager is the conclusion stated by a Government engineer: "Not always, but generally, the cheapest coal to buy in any locality is the lowest-priced one, provided conditions in the plant will permit its use." The report shows that for almost any character of coal, old furnaces may be adapted, or new ones designed, which will burn such fuel with reasonable efficiency, and that practically no fuel should be rejected as worthless. The post-office department some hotels, hospitals, apartments and office buildings have been using screenings with considerable economy, althoug there are many theoretical objections t the use of such fuels.

Electric Pumps versus Steam Pumps for Hydraulic-elevator Service

At the Security Mutual Life Insuranc Company's building, in Binghamton, series of four different tests was ru upon the elevators. One week the electric-elevator pump was used and the nex week the steam-elevator pump. The coa consumption, electrical output and number of elevator trips were carefully note The following data give in brief the resu of a typical test run for two five-day periods:

	Electric Pumps.	Steam Pumps.
Coal burned, tons	17 .3	20
Number of elevator trips	3937	3363
Kilowatt-hours of electricity	4780	3230

This shows that 18 per cent. more serce was delivered with 16 per cent. less iel when electric pumps were used. With al at $3 per ton, the electric pumps ived $8.10 during the five days' test, but if ie additional 1550 kilowatt-hours reuired to run these pumps had been ught from an electric company at 5 nts per kilowatt-hour, it would have st $77.50. Or, if bought as low as 3 nts per kilowatt-hour the electricity ould have cost $43.50, entirely wiping ut the saving of $8.10 in fuel which the ectric pumping effected.

Present Practice in Elevators

For a building higher than twenty-five ories—the traction type of electric eleators.

For the greatest degree of safety—the ydraulic-plunger elevator.

For the greatest competition and lowst first cost—the electric elevator.

For the lowest repair expenses—the ydraulic elevator.

For the most flexible equipment—the lectric elevator, as it can use central-staon or isolated-plant electricity, if deigned intelligently.

For the engine-room crew with the ast experience in elevators—the plunger levator.

For the lowest operating cost in most uildings—the electric elevator.

For the least amount of machinery in ie basement—the overhead electric eleator.

For the lowest operating cost where the levator travel is small—the electric eleator.

For the most popular elevator, with irgest sales—the electric elevator.

For a building later to be increased in ight—the traction type of electric eleator.

For the elevator with the least bounce i stopping—the electric traction elevator.

For a six-story apartment covering an rea of 100x100 feet—one electric eleator at a speed of 250 feet per minute.

For a twelve-story apartment covering n area of 100x100 feet—two electric elevators at 250 to 300 feet per minute.

For a twelve-story hotel covering an area of 100x100 feet—one freight and two passenger electric elevators at 300 feet per minute.

For a twelve-story office building covering an area of 100x100 feet—one passenger electric elevator at 350 to 400 feet per minute.

For a twelve-story loft building covering an area of 100x100 feet for light manufacturing—two freight and two passenger electric elevators.

For a six-story apartment covering an area of 100x100 feet—an electrical consumption of 350 to 400 kilowatt-hours per month.

For a twelve-story apartment (any size)—an electrical consumption of 750 kilowatt-hours per year per apartment.

For a twelve-story apartment hotel covering an area of 60x100 feet—an electrical consumption of 3300 kilowatt-hours per month.

For a twelve-story office building covering an area of 100x100 feet—66,000 kilowatt-hours per year—three elevators.

For a twelve-story loft building covering an area of 100x100 feet—an electrical consumption of 500 to 750 kilowatt-hours per month for each elevator.

Cause of Variations in Cost

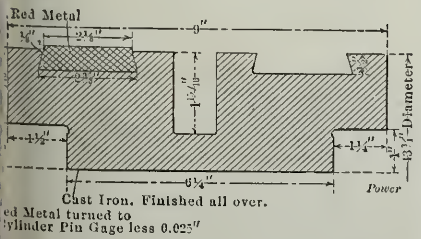
If an engineer is told that a neighboring building, similar in size and character, is spending only $10,000 per year for light, heat and power, while the expenses of his plant are $15,000 per year, nine times out of ten he will reply that "Conditions there are different." This may be true, perhaps, but not the whole truth. Then if the engineer of the neighboring building is asked how it is that his plant only costs $10,000 per year for light, heat and power, whereas the first building costs $15,000, he will probably answer, "I know my business and prevent waste." This, again, while true, is not the whole truth. The point I wish to emphasize is that the operating engineers as a class shift all the blame for a bad operating record and claim all the credit for a good operating record. The only way for building managers to decide whether the operating man is at fault or not, is to delve into old records of coal bills. If it is found that during some preceding year when the building was practically full the coal bill was 20 per cent. less and the price of coal practically the same, with no mitigating circumstances such as a very mild winter, the later addition of an ice plant, another elevator, a vacuum-sweeping installation or the construction of a large building on either side, then the manager may be sure that it is time to make a change. It is time for him to get a new engineer and pay him $20 more per month and change to another coal dealer. Or better yet, give the new engineer 10 or 20 per cent. of what he saves. Then, for every dollar's worth that he expends he feels that he is taking 10 or 20 cents out of his own pocket.

On the other hand, if the coal bill has not increased a substantial amount over past years, one may be sure that the reason for the high cost is not to be laid at the door of the operating engineer, but antedates the time at which he was hired. In this case it may be attributed to an error in the design of the plant. As the chief engineer of the great Metropolitan building, Mr. Bavier, said: "Anyone taking over a new building is like a man adopting a child; someone else created it and hereditary weaknesses on the part of the child must not be charged against the new parent who adopted it." Nor should errors in the design of the engineering equipment be charged against the building manager, for he is only too often doing the best he can with what someone else provided. Too many times that someone else is a man who has his mind intent, first, on landing a contract; second, on cutting down installation cost, and third, on (perhaps) the reduction of operating expenses and repairs in the future. The engineering salesman gets paid for new business and not for saving money in the basement ten or twenty years after the job is landed.

Packing and Junk Rings

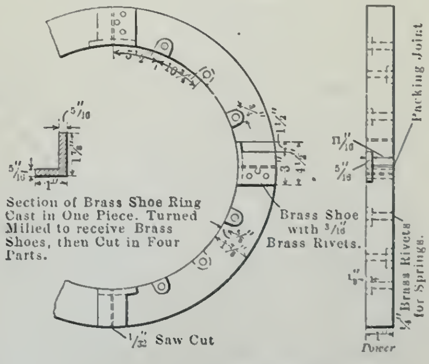
By W. J. Barker

Fig. 1 represents the section of a cast-on junk ring lined with Allen's red etal. The latter projects ⅛ inch beyond e cast iron and is 2⅛ inches wide. From past experience I can say that low-pressure junk rings made up in this way will last from five to six years before it is necessary to renew the red metal.



IG. 1. SECTION OF LOW-PRESSURE JUNK RING

Fig. 2 shows half of a 44-inch cast-iron packing ring. The ring is narrow and deep and has a good bearing on the sides of the groove which helps to keep the ring from rocking. It is held against the walls of the cylinder by flat steel springs ⅛ inch thick and the width of the groove. These springs are hinged on the ring in the recesses made for this purpose and are self-adjusting.



5½″
10⅞″
5/16″
1⅞″
5/16″
1″
1½″
3″
4½″
Section of Brass Shoe Ring Cast in One Piece. Turned Milled to receive Brass Shoes, then Cut in Four Parts.
Brass Shoe with 3/16″ Brass Rivets.
1⅛″
1/32″ Saw Cut
Packing Joint
11/16″
5/16″
¼″ Brass Rivets for Springs.
1″
Power

FIG. 2. LOW-PRESSURE PACKING RING

Significant Figures for Engineers

How many figures should be retained in the answer of an engineering problem? This matter is very confusing unless some idea of significant figures is understood.

A digit is any figure from 1 to 9 inclusive. A significant figure is any digit which is employed to signify the amount of the quantity to be used. The zero is, therefore, as important as any of the digits and should be retained if the work warrants it. For instance, suppose a measurement to be made and that it is found to be 451.10 inches. The measurement is one that can be made to 1/100 of an inch, and if the final zero were dropped, making the measurement 451.1 inches, it would not be known how accurately the work had been done, as it would seem that the measurement had been made only to 1/10 of an inch. When it is necessary to reject figures, the last one of those retained should be increased by one, if the next digit is over five. If an answer came out 387.567 and the required accuracy warranted retaining only four figures, the answer would be 387.6 instead of 387.5.

In a long calculation there will be a series of errors or approximations, and the answer would have some figures which would be correct and others which would be of doubtful exactness. If only the correct ones were retained, and *one* place of the approximate figures, the average of the observations would be still doubtful as to the last figure. On the other hand, if two doubtful figures were retained, the accuracy of the second last would be reasonably sure. As an example of this, take the case of the value of the circumference of a circle divided by the diameter. Suppose the determination were made a number of times, say six or seven, and then the results averaged. It is supposed that some method of measuring both the circumference and the diameter was employed and that the following figures represent the results of dividing the values obtained for the circumference by those obtained for the diameter:

3.1412
3.1420
3.1417
3.1411
3.1419
3.1413

6)18.8492
3.1415333+

In the above, one may be absolutely sure of the first *three* figures, the last two being doubtful. In averaging it was necessary to retain the two doubtful ones or be uncertain of the reliability of the fourth figure in the answer. The answer should be 3.14153, every figure of which it is perfectly legitimate to retain. The

By H. J. Macintire

Practical examples which show the uselessness of carrying mathematical calculations to extremes when the factors involved contain error.

last figure is, of course, very doubtful. The rule to be followed is then:

(1) *In data to be averaged, retain two places of unreliable figures. The average may contain one additional figure.*

Suppose that an engine test was being figured with the following data:

$P = 53.4$ pounds per square inch;
$L = 12$ inches;
$A = 113.1$ square inches;
$N = 200$ strokes per minute;

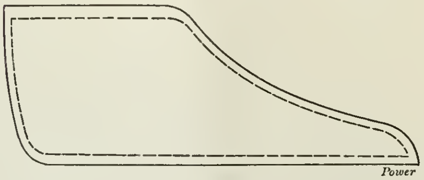
$$H.P. = \frac{PLAN}{33{,}000} = \frac{53.4 \times 12 \times 113.1 \times 200}{33{,}000 \times 12}$$

$$= 36.6033$$

The question arises, how many figures should be retained in the numerator and the answer? Consider the factors of the numerator. The stroke can be measured to 1/100 of an inch, also the diameter. The precision here is within one in

$$100 \times 12 = 1200$$

or 1/12 of 1 per cent. It is assumed that the revolutions per minute are ascertained with an ordinary counter, and a precision of only one in 200 or ½ of 1 per cent. can be assumed. When it comes to the mean effective pressure it is much different, for a number of things unite to make the possible error great. First, the reducing motion is a variable quantity; it may give anything from a very bad to an exact reproduction of the motion of the piston. Then if the indicator is some distance away from the reducing motion, requiring a long length of cord, there is sure to be a stretch to it unless soft piano wire is used or something equally as good. Backlash in the pantograph of the indicator will give a diagram as shown herewith, which is smaller than the correct diagram, and finally, but least by no means, the indicator may be incorrect. To be sure, the indicator should be calibrated before and after the test, an an average calibration taken. As a rul a 2-pound error in an 80-pound sprin is not unusual for a good spring, or 4- or 5-pound error for a spring that ha had considerable use. Assuming that it impracticable to calibrate the instrumen and the error, as is usually the case, unknown, a variation of two pounds in th spring may be assumed. Such an erro in an 80-pound spring is 2½ per cen or in this factor the test is in error 2 per cent. Therefore, there is a doubt the extent of 2½ per cent. of 36.60 0.9 horsepower. The answer as regar the first digit is right, but there is dou of the second; hence, the answer shou be 36.6 horsepower. More figures wou be absurd and useless.



ERROR IN DIAGRAM DUE TO BACKLASH

(2) *In multiplying and dividing th least reliable factor should be determin first. Find its precision and retain on enough figures in the answer to give th same precision.*

Certain errors, although great at fir if the test is continued for a comparativ ly long time, will become very mu smaller in the total percentage err Take, for example, the boiler test whe the hight of the fire or the grate is es mated before and after the test. T possible error is about 2 inches in ge ting the hight of the fire. The enti error would be,

$$\frac{2 \times \textit{grate area}}{12}$$

cubic feet of coal at 60 pounds per cu foot. For a short run, or in one usi a small amount of coal, the error wou be great, but if the amount of coal us were large, the resulting percentage err would be inappreciable. Errors in t weights of the coal and the water, ho ever, are accumulative, for if the scal are out of order and only weigh with a few pounds in 500, the error will just as great a percentage at the end at first. The kind of measurement, a the length of the time of the test, the fore, affect the precision and the answ

The following is a good example of perfect disregard of significant figu taken in detail from a well known har book. The data are of a boiler test:

Heat of feed water, 79 B.t.u.
Total heat imparted to each pound of ste
1189.34 — 79 = 1110.34 B.t.u.
Total weight steam and water passing thro
engine, 216,492 pounds.
Total amount of heat given to engine
216,492 × 1110.34 = 240,379,727.28 B.t.

Granted that the total heat of steam accurate to 1/200 of 1 per cent., which is not, and that the water could weighed to one pound in 5000 poun how many figures should there be in t answer? The least precision here is c in 5000 and, therefore, the answer ca not possibly be more precise. The answ should be 24,380,000 B.t.u.

Electrical Department

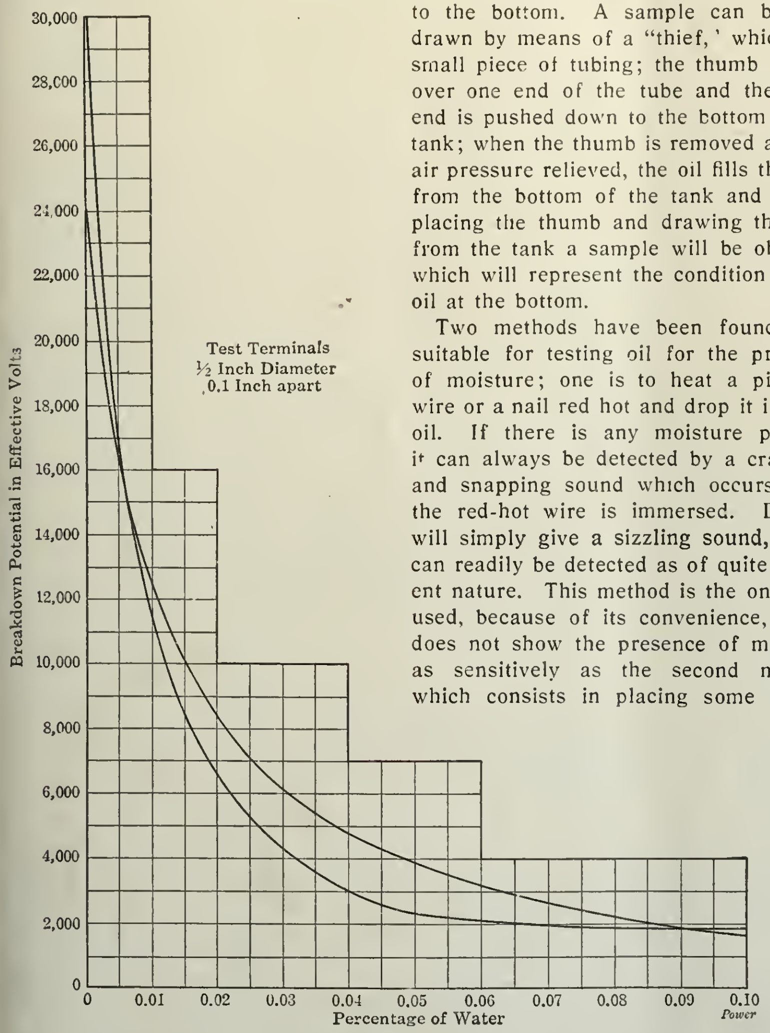
Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Moisture and Impurities in Transformer Oil

By J. Vincent Hunter

In a previous article* I pointed out te importance of keeping transformer d switch-tank oil (commonly called 'ansil" oil) free from moisture, becuse of the rapid decrease in inslating quality that is produced by en minute additions of water t the oil. The next points for nsideration are the detection and reoval of moisture and, incidentally, any inor impurities.

The presence of moisture in transil oil ay be detected in various ways, but in der to test for moisture, it is customary draw a sample of oil from the suscted transformer tank or switch box d test the sample. This should be ken from the bottom of the tank because the water has a tendency to settle to the bottom. A sample can best be drawn by means of a "thief," which is a small piece of tubing; the thumb is held over one end of the tube and the other end is pushed down to the bottom of the tank; when the thumb is removed and the air pressure relieved, the oil fills the tube from the bottom of the tank and by replacing the thumb and drawing the tube from the tank a sample will be obtained which will represent the condition of the oil at the bottom.



Effect of Moisture upon Insulation Resistance of Oil

Two methods have been found most suitable for testing oil for the presence of moisture; one is to heat a piece of wire or a nail red hot and drop it into the oil. If there is any moisture present, it can always be detected by a crackling and snapping sound which occurs when the red-hot wire is immersed. Dry oil will simply give a sizzling sound, which can readily be detected as of quite different nature. This method is the one most used, because of its convenience, but it does not show the presence of moisture as sensitively as the second method, which consists in placing some of the oil in a porcelain or glass dish and heating it very rapidly. A crackling sound similar to that produced by dropping in a red-hot wire will be noticeable if moisture is present. Neither method will detect very small percentages of moisture.

A number of schemes have been tried for "drying" transil oil to remove the moisture and several have been mentioned briefly at different times in the columns of Power. H. W. Tobey, in a recent number of the *Proceedings* of the American Institute of Electrical Engineers, has described these methods very thoroughly, but as it is probable that a majority of the readers of Power do not see the *Proceedings*, it may be well to give the gist of Mr. Tobey's paper. The chief methods of "drying" are (1) by the use of chemicals; (2) by the application of heat in some form; (3) by mechanical means and (4) by filters.

The first method is the one most used by central-station men, because they do not, as a rule, have the use of the elaborate paraphernalia usually required for the other treatments. The best known of the chemical methods is dehydration of the oil by means of metallic sodium. When metallic sodium is dropped in oil containing moisture it combines with the moisture very readily, forming caustic soda and liberating hydrogen gas; hydrogen is, of course, highly inflammable and care must be taken that no flame is in the vicinity when the "drying" is being done. This method is open to the objection that the caustic soda is very apt to attack the insulating varnishes and compounds used in the manufacture of the transformer and soften and weaken them considerably. Whether or not this will take place when the insulation is actually protected by the oil has not yet been determined, but it would not be considered good practice to run this risk.

The best and surest chemical drier is calcium chloride, which has a strong affinity for water and consequently has a strong dehydrating action. No objectionable chemical combination is formed when this is used, and the calcium chloride will not attack any of the material of the transformer. Dry, unslacked lime or calcium oxide can also be used for this purpose and so can calcium carbide, but both of these are comparatively slow in action, as compared with sodium and calcium chloride; the latter is the most rapid of any. These "drying" agents can best be used by putting them in a can plentifully perforated with small holes and hanging it in the oil. If they are applied in this way instead of being thrown in loose it is possible to remove them as soon as the oil is freed of its moisture and thus avoid the chance of having dirty oil. It usually requires nearly 75 hours for calcium chloride to restore transformer oil to its original dielectric strength and sodium requires as much as 120 to 150 hours, so it can readily be seen that these cannot be considered very rapid "driers."

It is often customary to dehydrate transil oil by means of heat, but this process is very slow and not at all sure. Great care must be taken not to overheat the oil for fear of carbonizing it. This process can be varied by putting the oil in a closed tank in which a vacuum is maintained, as this will cause the moisture to dry out more rapidly, or the oil may be heated and dry air blown through it, the air being employed as an agent to carry the moisture away.

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*Page 1862, October 18, 1910.

Moisture can be removed from transil oil to some extent by means of centrifugal machines, such as a cream separator, and a mixture of oil and water, which will contain the greater part of the water that was in the oil before treatment, can be removed from the circumference of the separator runner. This will not remove the moisture entirely and it is customary to filter the oil afterward to remove the remainder of the moisture. It is also possible to remove much of the water, if present in large proportion, by allowing the oil to stand undisturbed long enough for the water to settle to the bottom. This, however, will not bring the oil up to the standard which it should have.

The most modern means of dehydrating transil oil thoroughly is the filter press. It is well known that some substances will permit oils to pass through them, but will not pass water. Those who have strained gasolene through chamois skin to remove water will appreciate this. For this process a common type of large-capacity pressure filter is used such as may sometimes be found in large chemical laboratories. A quantity of perfectly dry blotting or filtering paper is placed between the plates and the oil forced through at a pressure in the neighborhood of 50 pounds per square inch. The filtering paper not only has the effect of removing all moisture, and probably absorbs the moisture which it does remove, but it also has the advantage of removing all dirt which may be mixed with the oil. Oil treated by this process will always come through in excellent condition, and if it does not test out as well as can be wished for, it is a very easy matter to run it through a second time. The filtering paper must be replaced from time to time as it becomes clogged with dirt, but it is only necessary to replace the top sheet in each pack of paper, as the first one always catches all the dirt. This is probably not only the best but the cheapest and easiest way to clean oil, as it is only necessary to start the apparatus and it will keep going until the tank is empty.

Some care must be taken that the oil does not come in contact with other substances besides moisture which may affect it. Chief among these is sulphur, which is to some extent soluble in oil and has often in the past been used in transformers as a cement. This is now considered particularly bad practice and the best transformer builders have long since abandoned any use of this material in their apparatus; moreover, they are usually careful to see that little sulphur is contained in the rubber-covered leads which are often used in transformer cables on the terminals within the case. It formerly was thought that sulphur would reduce the dielectric strength of the oil, but tests that I have made have failed to show this to be a fact, and the dielectric strength has never at any time shown impairment, even with a large percentage of sulphur.

The really serious objection to the presence of sulphur is that it attacks and corrodes copper very readily, and where there were fine-wire windings in the apparatus it has been known even to eat the copper away until an open circuit was formed. The presence of sulphur is easily tested for on account of the fact that it will blacken copper so readily; if a small portion of the suspected oil be heated and a bright copper wire hung in it, the copper will usually show the blackening effect of the sulphur within a few hours, if any sulphur be present.

Oil which has been overheated for a great length of time and to a very considerable extent grows black and dirty due to the carbonizing effect. It will also attack to a limited extent the insulating and waterproofing compounds with which most transformer windings and other coils are impregnated. The formation of the carbon in the oil will tend to reduce the insulating quality, but it is doubtful whether the solution of the impregnating gums from the coils will seriously injure the oil, although it does give it a dirty appearance. Running such an oil through a filter press, as mentioned before, will remove both of these impurities, so that an oil thus treated will usually be restored to its original appearance.

With reasonable care the life of transil oil can be prolonged beyond that which is usually attained, and as this is not at all an inexpensive supply it is well worth the trouble of a little study of its care and treatment.

Single Phase Traction in France

One of the latest examples of the substitution of electric for steam locomotives in France is the Midi railway. The portion to be "electrified" is some 70 miles long; the country is very hilly and there are several bad grades, one of 3½ per cent. being about seven miles long. It is stated that the electrical equipment will eventually be extended to the entire Toulouse-Bayonne Line, a distance of 200 miles.

The Midi Railway Company has ordered equipments for 30 double-truck motor coaches for passenger service and an electric locomotive for freight service. Each of the passenger coaches will seat about 50 passengers and will be equipped with four 125-horsepower Westinghouse single-phase motors built for 16⅔ cycles and 285 volts. These motor coaches will be able to haul trailers weighing 44 metric tons (97,000 pounds), besides its own weight and load, at a speed of 45 miles per hour on level track.

The Midi locomotive will be provided with five axles, three of which will be driven by two 600-horsepower motors through jack shafts and connecting rods. The locomotive will weigh 80 metric tons (176,000 pounds) and will be able to haul trains weighing 320 metric tons, exclusive of the locomotive weight. With a haulage load of 280 metric tons the speed will be 25 miles per hour, and with 100 metric tons about 38 miles per hour.

The current will be supplied to the locomotive and motor cars by means of a 12,000-volt overhead catenary line and stepped down to motor voltage by transformers on the cars. The pantograph type of trolley will be used.

LETTERS

Too Much Bolt and Washer

During the latter part of July, our company bought three second-hand electric coal-cutting machines, with chain cutters, and we started in to overhaul them and put them in good running order. Two of the machines tested clear with the magneto but the third showed a ground the bell ringing at whatever point the magneto terminal touched. After disconnecting the field terminals and removing the commutator brushes, we tested these and found them all clear, showing tha the trouble was with the rheostat. When testing the rheostat the magneto rang as soon as the terminal touched any of the various contacts or even the machine frame, and the fact of getting a ring when touching the frame led us to look for trouble at the points where the rheostat is screwed to the machine frame At each of these points a cap screw passes through the front plate and through a short distance piece of pipe, and then enters the screw hole in the machine frame. A washer is inserted between the pipe and the rheostat plate, and thi washer was so much larger than necessary that it touched the bottom of the resistance conductor, giving a path fo the current to pass to the machine frame By putting on smaller washers thi trouble was cleared, but the main trouble remained, that is, the rheostat tested up at every point grounded on its frame. We finally discovered that in mounting the cable binding post on the face of the rheostat, the post had been placed directly opposite one of the screws which hold the bottom resistance frame to the faceplate and the two screws from opposite sides of the faceplate came together, mak ing the short-circuit direct from the live cable binding post, through the screws to the resistance conductor, and thence through the washer and cap screw to the machine frame.

Richard Clemens.

Canal Dover, O.

Polarity Indicators

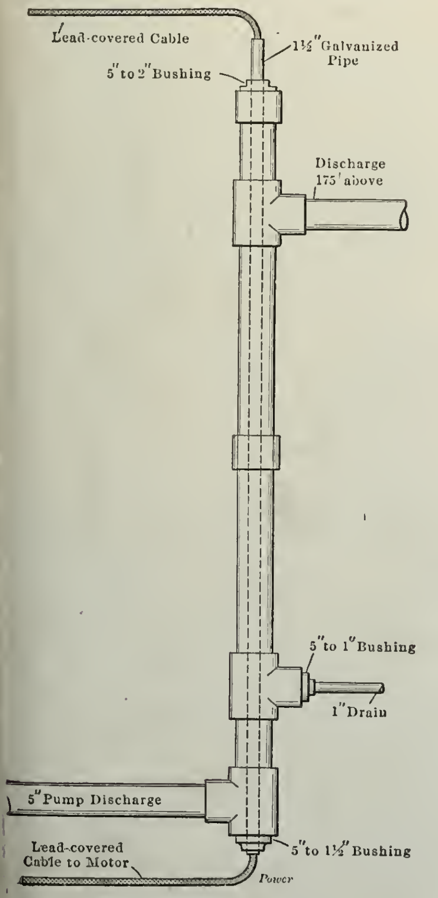
The simple method of determining the polarity of two direct-current conductors described by Wesley McArdell in the September 27 issue is all right except that it is not always convenient to get a potato in a power plant. Were Mr. McArdell to spit on a board and insert his two wires in it about ½ or ⅝ inch apart he would find around one of the wires the same green discoloration due to the copper dissolved from the positive wire.

William T. Garlitz.

McKees Rocks, Penn.

A Novel Wiring Conduit

Some time ago we installed an electric pump about 2000 feet underground. The drill hole for the discharge pipe had been put down and a 5-inch discharge pipe set in it. At that time we could not wait for another drill hole to be sunk through which to put the lead-covered cable to connect up the motor, and the nearest hole was that containing the exhaust pipe from the steam pump which the electric was to displace; this was approximately 700 feet away. We did not like to put the steam pump out of commission because something might go wrong with the electric pump, and the water must be kept out. After a good deal of thought as to the best way out of the dilemma, I used the scheme illustrated in sketch, and this has not given a moment's trouble since installed. It consists merely of running the motor cable through a 1½-inch pipe and putting that in the 5-inch discharge pipe of the pump.



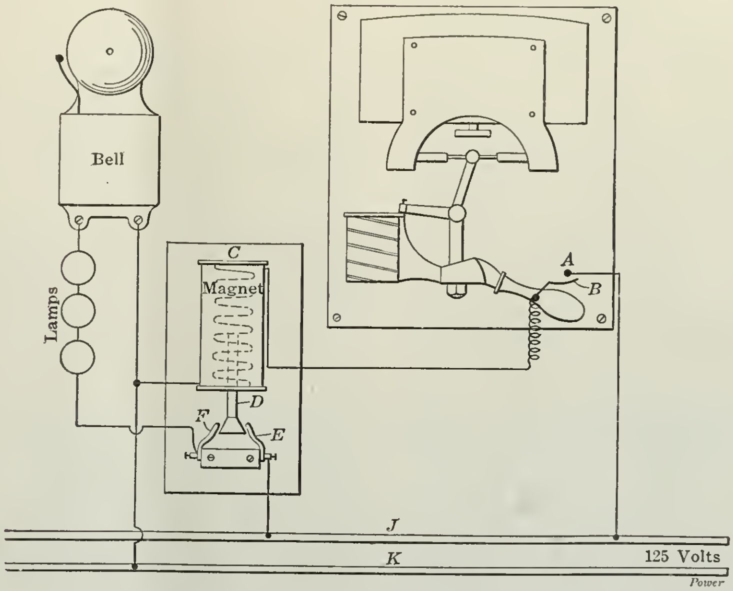
A Novel Wiring Conduit

L. Earle Brown.

Roundup, Mont.

Another Circuit Breaker Alarm

Mr. Warren's description last week of a simple mechanical arrangement for closing a bell circuit when a circuit-breaker goes out brings to mind an electrical system for accomplishing the same result which I saw some time ago in a Southern power house. This system is illustrated elementally by the accompanying diagram. To the circuit-breaker handle is attached a piece of spring wire *B* which is brought into contact with the pin *A* when the circuit-breaker opens. This pin is connected to the busbar *J* and the spring wire *B* is connected to one terminal of the magnet coil *C*, the other terminal of which is connected to the busbar *K*.



Relay-operated Circuit-breaker Alarm

Consequently, when the circuit-breaker opens, the magnet *C* is energized by current from the 125-volt busbars *J* and *K*, and draws the plunger *D* up until the conical end bridges the spring contacts *E* and *F*, thereby closing a circuit from the busbars through the alarm bell and three incandescent lamps. The lamps serve as resistance to prevent too much current from passing through the bell magnet and also offset largely the inductance of the bell magnet, thereby preventing excessive flashing at the vibrator contacts and at the pin *A* and spring *B* when they are separated by closing the circuit-breaker.

This arrangement has the advantages that the alarm bell may be located at any desired place in the plant and that one alarm system will serve for any number of circuit-breakers. When any circuit-breaker is left open for a considerable length of time, a piece of rubber tubing is slipped over the pin *A* to prevent the bell from ringing uselessly.

Geo. W. Malcolm.

Brooklyn, N. Y.

Humming Armature Cores

I noticed in an article recently published in Power a discussion of humming in armatures and rotors. I have assembled several thousand cores of both kinds and only had one to hum. The cause was an improper fit between the disks and the housing of a stationary core, which caused a wavy appearance of the finished assembly. No machine assembled under a reasonable pressure per square inch of disk surface, where the variation of size is not over 0.0005 inch, will hum when running. The trouble with too tight a fit is that the disks will buckle and be tight at one point and loose diametrically opposite. A slight buckle will not hurt, if all the vent ribs have a bearing on the flanges.

Alfred N. Hammond.

Erie, Penn.

The Inland Waterways Commission in America has reported that 150,000,000 water horsepower are running to waste, and if we remember aright this referred to North America alone.

Gas Power Department

Developments at the Deutz Gas Engine Works

By F. R. Low

At the conclusion of the recent European meeting of the American Society of Mechanical Engineers, some of the delegates visited a few of the prominent manufacturing establishments on the Continent, among them the Deutz Works at Cologne, which were established in 1864 by Doctor Otto and his partner, Eugene Langen, for the manufacture of gas engines. Large oil paintings and busts of the founders adorn the room in which we were received by the chief engineer, F. Hundeshagen, who kindly conducted us through the shops and told of the work being done there.

One of the interesting things that were seen was the first of the large double-acting gas engines; a 250-horsepower unit made in 1902. Readers of Power of ten years ago will remember that the question of making the gas engine double acting was actively discussed at the time of the Paris Exposition. The difficulty lay in providing a stuffing box that would stand the temperature, and in water cooling the piston and rod. The engine just mentioned is said to be the first one of considerable size in which this was satisfactorily done, and it is still running successfully.

It is interesting to record that this company, which was so active in developing the double-acting engine, has been among the first to recognize its limitations and now builds everything single acting up to capacities which require the piston to be water cooled. The simple trunk piston, nothing but a hollow cylinder of cast iron with a few snap rings and a wristpin for the connecting rod, is so concrete and satisfactory an arrangement to use with the high temperatures and great forces involved in gas-engine work that the Deutz people have come back to it wherever it is practicable and have concluded that it is better to make the engine a little larger and to run it a little faster, or even to use more cylinders for a given engine, than it is to complicate the engine with stuffing boxes and water-cooled rods and to add to its length, cost and number of parts by putting in crossheads and guides. They can get 200 horsepower out of a single cylinder without water cooling the piston, and by using four such cylinders can supply a compact simple engine of 800 horsepower, free from liability to lubrication troubles and leaky cooling systems, and at a lower cost than that of the more complicated and delicate double-acting machine.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

The company has a design in which two cylinders are cast together, side by side horizontally, and deliver power to cranks set at 180 degrees, avoiding the necessity of counterbalances for the crank webs and giving a fairly uniform turning moment. The shaft extends at one side and is supported by an outboard bearing. They have had no difficulty with cylinder breakage and do not hesitate to cast working barrels up to 75 millimeters(practically 3 inches) thick. They avoid tying together parts that are subjected to different temperatures and avoid breakage in rods by a liberal use of nickel steel. Their valves are of cast iron with nickel-steel stems. They get 1000 horsepower from a single cylinder of 1000 to 1100 millimeters (39 to 43 inches) bore and 1300 millimeters (50 inches) stroke, with 90 to 92 revolutions per minute, and have built several tandem two-cylinder and twin-tandem four-cylinder engines of 2000 horsepower.

The Deutz company builds producers for all sorts of fuel and service. Peat, Herr Hundeshagen said, works well in a producer when it contains not more than 25 per cent. of moisture.

The company also makes many engines to run upon benzol, a coke-oven byproduct. From 230 to 250 grams (8.1 to 8.8 ounces) per brake horsepower-hour are required and it costs from 16 to 17 marks per 100 kilos (1.8 to 1.9 cents per pound). Its heat value is from 9300 to 9500 calories per kilo (about 16,750 to 17,000 B.t.u. per pound). Little, compact six-cylinder engines, three cylinders on each side, inclined away from the vertical, are made for use with this fuel on railway cars for local service. The engine drives a generator which supplies current to motors upon the axles. There is automatic regulation of the generator output to give the large amperage and low voltage required for starting and the higher voltage and lesser current required for running at speed, and as the engine runs at constant speed, the full power is available for starting up. The company builds also a kerosene engine called the Brons, self-igniting without flame or spark, but with high compression like the Diesel. It is built up to 32 horsepower and is much used for fishing and freight boats. Large engines of the Diesel type, working on the four-stroke cycle for stationary service and the two-stroke cycle for marine service, are also built. Little narrow-gage locomotives of 12 to 50 horsepower are another attractive product, of which about 900 have been built.

The Deutz foundry work is admirable—so fine, in fact, that the company has abandoned bronze and cut spiral gears for running lay shafts in favor of cast gears run as they come from the mold, without any machine work upon their working faces. They are split so as to be easily placed upon the shaft, the split extending along the diagonal bottoms of opposite teeth, and run in oil.

Efficient Gas Cleansing

As an example of modern methods of purifying producer gas, the equipment of the Penn-American Plate Glass Company, at Alexandria, Ind., is of interest. A portion of the company's power equipment consists of a producer-gas engine plant comprising three 500-horsepower Körting engines and a 300-horsepower Westinghouse engine, all direct connected to electric generators. These engines are supplied with gas by four 500-horsepower up-draft soft-coal producers of the pressure type. No gas holder is used but the producers are supplied with forced draft by means of steam jets which provide sufficient pressure to overcome resistance of the fuel bed and to deliver the gas into the main under a pressure of from 1½ to 2½ inches of water; from the main the gas passes through a cooling tower to a two-stage centrifugal scrubber and tar extractor of the turbine type, built by the Buffalo Forge Company.

The four gas engines run continuously 24 hours a day and 6 days a week, and it is reported that examination at the end of each week has shown cleaning of the engines to be unnecessary. We are also informed that after several months of regular service not the slightest trace of tar or lampblack was to be found in the gas main leading from the scrubber, and that the freedom from tar and lampblack is so complete that the governors regulate perfectly, no trouble having been experienced from this source since the installation of the Buffalo scrubber.

Accident Due to Careless Governor Construction

By Earl Pagett

A peculiar accident occurred recently to the governor on one of the gas engines under my charge, both of the governor arms being broken off about as indicated at *I* in Fig. 1. The governor was of the centrifugal flyball type, geared to the main shaft through one set of spur gears and a set of miter gears.

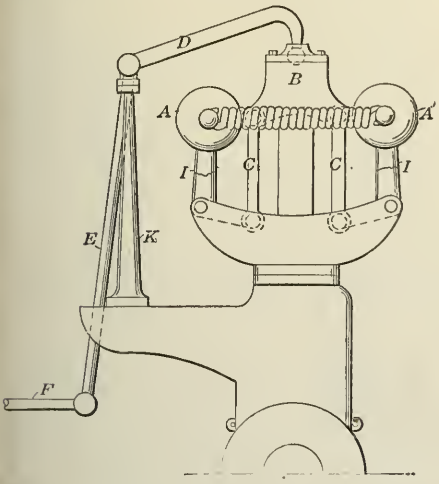


Fig. 1. The Governor

The sketch, Fig. 1, will serve to give an idea of its general construction. It is evident that centrifugal force tends to throw the weights *A* and *A'* apart, thereby raising the cap *B* through the links *C*, and giving motion to the levers *D* and *E* and the reach rod *F* which operates the gas valve. Two stout helical springs oppose the centrifugal effect on the balls.

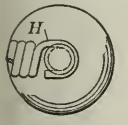


Fig. 2.

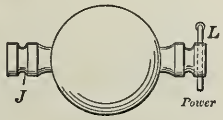


Fig. 3.

The ends of the springs are bent around the pins in the balls as at *H*, Fig. 2, these eyes fitting into grooves *J*, Fig. 3.

This engine and another of about the same size are belted to the same line shaft, each one having its own clutch. Ignition current is normally supplied by magnetos driven from the line shaft, but batteries are used in starting up. The man who operates the clutches is instructed to throw the igniter circuit switches over so that the igniters will take current from the batteries as soon as he throws out the clutches for any reason, in order not to shut down the engines by the shutting down of the magnetos.

About noon, while I was in one of the other engine rooms, the clutches were thrown out, and the magnetos stopped. The foreman did not throw the switches over to the batteries until the engines had slowed down considerably; then he changed the switches over to the batteries and things began to happen that sounded like a ragged volley of rifle shots.

I had just blown the whistle for noon and was on my way to stop these two engines when I heard the noise. When I reached the room the governor weights and springs were lying around promiscuously, but the foreman had had enough presence of mind to pull the igniter switches and shut down the engines.

The only theory that I can suggest to explain the accident is that one of the springs jumped off its pins, allowing the weights to be thrown out far enough to strike the pedestal *K*.

Of course, they would have caused the governor to shut the gas entirely off before striking if the engine had been running alone, but as it was connected to the line shaft with the other engine, the speed was kept up enough to cause the balls to strike; after they broke off, the centrifugal force was not great enough to hold the stub ends out and the engine ran away when the switch was thrown over to the battery.

To prevent any chance of the springs getting off again, I drilled holes and put in cotter pins, as at *L*, Fig. 3.

Lloyd's Rules for Marine Engines

Lloyd's Register of British and Foreign Shipping recently issued the following rules for the survey of internal-combustion engines for marine purposes:

Section 1. General

In vessels propelled by internal-combustion engines the rules as regards machinery will be the same as those relating to steam engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes and propellers.

Section 2. Construction

The following points should be observed in connection with the design of the engines:

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed, and easily accessible for examination and adjustment.

4. In engines of above 60 brake horsepower which are not reversible, and which are manœuvered by a clutch, a governor or other arrangement must be fitted to prevent racing of the engine when declutched.

5. Efficient positive means of lubrication—preferably sight feed—must be fitted to each part requiring continuous lubrication.

6. If the engines are of the closed-in type they must be so arranged that the lubricating oil can be drained when necessary, and in wood vessels an easily drained metal or metal-lined tray must be fitted to prevent leakage of either fuel oil or of lubricating oil from saturating the woodwork.

7. Carbureters and vaporizers should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carbureter or vaporizer, a gauze-covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic-gauze diaphragms should be fitted either between the carbureter (or vaporizer) and cylinders or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and battery, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from fuel pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be inclosed, and the spark coils must not be placed where they can be exposed to explosive vapors.

9. No exposed spark gap should be used.

10. In kerosene and heavy-oil engines, where lamps are used for ignition or for vaporizing, these lamps should be supported by some suitable bracket and the flame inclosed when in use.

11. The circulating-pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position, and the circulating pipe is to be provided with an efficient strainer inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

12. A bilge pump worked by the engines, or an independent power-driven bilge pump, is to be arranged to draw from each part of the vessel. In open launches this bilge pump may be omitted, provided suitable hand pumps are provided.

13. The cylinders are to be tested by hydraulic pressure to twice the working pressure to which they will be subjected, the water jackets of the cylinders to 50 pounds per square inch and the exhaust pipes and silencer to 100 pounds per square inch.

14. The exhaust pipes and silencer should be efficiently water cooled or lagged to prevent damage by heat, and if the exhaust is led overboard near the water line, means must be arranged to prevent water from being siphoned back to the engine.

15. The machinery must be tried under full working conditions, the report stating the approximate speed of vessel,

the number of revolutions of the engines at full power, both ahead and astern, and the lowest number of revolutions of the engines which can be maintained for manœuvering purposes.

SECTION 3. RULES FOR DETERMINING SIZES OF SHAFTS

The crank, intermediate and other shafts, if of ordinarily mild steel, are to be of not less diameters than as given in the following table. When special steel is used the sizes are to be submitted for consideration.

1. For gasolene or kerosene engines for smooth-water service:

Diameter of crank shaft, in inches, =

$$C\sqrt[3]{D^2 S}$$

where

D = Diameter of cylinder in inches;
S = Stroke of piston in inches.

Four-stroke Cycle.	Two-stroke Cycle.	Bearing between Each Crank.	Two Cranks between the Bearings.
For 1, 2, 3 or 4 cylinders. .	1 or 2 cyls.	$C = 0.34$	$C = 0.38$
6 cylinders. .	3 cyls.	$C = 0.36$	$C = 0.40$
8 cylinders. .	4 cyls.	$C = 0.38$	$C = 0.425$
12 cylinders. .	6 cyls.	$C = 0.40$	$C = 0.49$

For open-sea service add 0.02 to C.

Diameter of intermediate and screw shafts, in inches, $= C\sqrt[3]{D^2 S(n+3)}$

where

D = Diameter of cylinder in inches;
S = Stroke of piston in inches;
n = Number of cylinders.

For smooth-water service,

$C = 0.155$ for intermediate shafts;
$C = 0.170$ for screw shafts fitted with continuous liners;
$C = 0.180$ for screw shafts fitted with separate liners or with no liners.

For open-sea service,

$C = 0.165$;
$C = 0.180$;
$C = 0.190$.

In engines working on the two-stroke cycle n is to be taken as twice the number of cylinders.

2. When ordinary deep-thrust collars are used, the diameter of the shaft between the collars is to be at least 21/20ths of that of the intermediate shaft.

3. In the cases of Diesel and other engines in which very high initial pressures are employed, particulars should be submitted for special consideration.

SECTION 4. FUEL TANKS AND CONNECTIONS

1. Separate fuel tanks are to be tested with all fittings to a head of at least 15 feet of water. If pressure-feed tanks are employed, they are to be tested to twice the working pressure which will come on them, but at least to a head of 15 feet of water. If the tanks are made of iron or steel they should be galvanized.

2. Strong and readily removable metallic-gauze diaphragms should be fitted at all openings on gasolene tanks.

3. Kerosene or heavy-oil tanks, not used under pressure, are to be fitted with air pipes leading above deck. Pressure-feed tanks and tanks containing gasolene should be provided with safety valves discharging into pipes leading to the atmosphere above deck. The upper ends of all pipes are to be turned down, and pipes above 1 inch in diameter are to be provided with gauze diaphragms at the end.

4. No glass gages are to be fitted to fuel tanks containing either gasolene, kerosene or heavy oil.

5. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

6. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For gasolene tanks the trays must have drains leading overboard where possible, or they should be gauze-covered trays with means for draining them.

7. All fuel pipes are to be of annealed seamless copper, with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carbureter or vaporizer. The fuel pipes should be installed in position where they are protected from mechanical injury and can be exposed to view throughout their whole length.

8. The engine room and the compartment in which the fuel tanks are situated are to be efficiently ventilated.

9. An approved fire-extinguishing apparatus must be supplied.

SECTION 5. PERIODICAL SURVEYS

1. The machinery is to be submitted to survey annually. At these surveys the cylinders, pistons, connecting rods, crank and other shafts, inlet and exhaust valves and gear, clutches, reversing gear, propeller, sea connections and pumps are to be examined. The electric ignition is to be examined and the electric leads tested. The fuel tanks and all connections are to be examined, and, if deemed necessary by the surveyor, to be tested to the same pressure as required when new. If practicable, the engines should be tested under working conditions.

2. The screw shaft is to be taken out at intervals of not more than two years.

Setting Small Engines

In the issue of September 20, George W. Malcolm, in criticizing my method of setting small gas engines, evidently overlooked the word "small" in the title of my article. However, I do not know what the limit of size is for engines that can be set in the way I described, but I have set them weighing 2½ tons without doing any "acrobatic stunts." The only acrobatic stunts that I have found necessary are those of the mental order, and I find that those are necessary on about all classes of work.

No fixed rule for placing the jacks can be given, as engines and conditions vary. In setting a horizontal engine, I use three jacks, one under the head and one under each end of the crank shaft. The engine can very easily be braced sidewise to maintain the proper alinement during the operation.

H. K. WILSON.

New Bedford, Mass.

A Lucky Escape

A rather unusual accident happened recently in a large plant of Diesel engines where I am employed. During the work of changing over the exhaust piping so it would pass out at the roof a ¾-inch bolt 4 inches long was left, unknowingly, in one of the admission-valve cages. On starting up the next morning the engine ran all right for a few minutes; then the bolt worked by the admission valve and dropped into the cylinder, causing a heavy jar. The engine was at once shut down and the trouble located by barring the flywheel around until the affected cylinder was identified. The head was taken off and the stray bolt discovered and removed.

We found the admission and exhaust valves badly sprung, but that was the only damage that was done. The valves were straightened and reseated and by night the engine was ready for its load again. We thought ourselves lucky to get off so lightly; the cylinder head might easily have been cracked under such a strain.

RICHARD L. SUTER.

Southbridge, Mass.

The total water power of the State of New York, without including that of the Niagara and St. Lawrence rivers, has been estimated at a million and a half. Eight hundred and eighty thousand of this is undeveloped. The New York Water Supply Commission has recently recommended that the State expend $20,000,000 to develop this water power, which it estimates would return an annual profit of $1,400,000.

In 1902 Russia produced 2,000,000 tons of peat fuel. This quantity is more than any other nation produced in that year.

Readers with Something to Say

Thermometers versus Gages

The engineering profession appears to be definitely committed to measuring steam pressure in pounds per square inch above the atmosphere, and, to add confusion, pressure below the atmosphere is stated as a vacuum in inches of mercury. This is an age of reform, and a simple and logical change, that has already found favor with experts, is to substitute temperature as measured by the thermometer.

When the theoretical side is considered it can be shown convincingly that the engineer must learn to think in terms of temperature if he wishes to get the right perspective, but it is not so easy to convince the practical engineer that he ought to substitute a little glass thermometer for his big nickeled gage. Unless the thermometer makers will provide instruments that can be read easily across the engine room the gage will have a decided advantage. Fortunately, the gage usually tends to read high, which is on the side of safety, though it may lead to disappointment in performance. The engineer frequently shows a naïve confidence in an imposing array of gages, but he would do well to occasionally pull out the pin at the zero mark and note how much too slow his gage registers.

There appears to be no reason why the thermometer in the boiler room should not be placed by the side of the gage glass, nor any reason why it could not be made to show up as positively as the latter. In the engine room there might be some difficulty in maintaining a circulation of steam through the thermometer cups if they are to be set side by side on a "thermometer board" but greater difficulties than this have been solved.

The great advantage of the thermometer would be that when once correctly graduated it would remain accurate until broken, which disaster would be improbable with the proper care. Physicists nay bother about the zero point changing a fraction of a degree, but this would not concern the engineer.

A simple problem solved by the aid of he steam and entropy tables will show he theoretical advantages of this contention. Let the steam be expanded diabatically from 420 to 100 degrees ahrenheit; that is, from 293 pounds gage 28 inches vacuum. To bring out the oint in question let the expansion take lace in ten equal steps. The heat contents in the entropy column 1.52 and eir difference are as follows:

At 420 degrees	....	1212.5 B.t.u.
At 100 degrees	....	845.9 B.t.u.
ifference 320 degrees	....	366.6 B.t.u.

> **Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.**

Divide this heat drop into ten parts of 36.66 B.t.u. each and subtract these successively in the following table:

Heat Contents.	Temperature.	Pressure.	
		Absolute.	Gage or Vacuum.
1212.5	420	308	293
36.66	3	108	
1175.84	382	200	185
36.66	37	74	
1139.18	345	126	111
36.66	35	48	
1102.52	310	78	65
36.66	34	32	
1065.86	276	46	31
36.66	32	19	
1029.20	244	27	12
36.66	31	12	
992.54	213	15	0
36.66	30	7	
955.88	183	8	14
36.66	28	4	
919.22	155	4	22
36.66	28	2	
882.56	127	2	26
36.66	27	1	
845.90	100	1	28

The first column gives the heat contents for the intermediate points, the small figures being the differences. Looking in Peabody's "Steam and Entropy Tables" under the entropy column 1.52 the nearest corresponding temperatures and pressures are found, neglecting fractions of a degree or pound. These quantities are inserted in the preceding table under the headings "temperatures and pressures absolute." Thus the first drop of 36.66 B.t.u. causes a fall in temperature of 38 degrees, while the pressure drops from 308 to 200 pounds. The temperature differences diminish as 38, 37, 35, etc., to 27; but the pressure differences run, 108, 74, 48, etc., to 1. The last column, with the break from gage pressure to inches of vacuum, is given to emphasize the inconvenience of the customary method.

How can engineers keep in mind that a drop of 2 inches of mercury, from 26 to 28, represents the same number of heat units given up as in a drop in pressure of 108 pounds, from 293 to 185 pounds gage. While it is true that 27 degrees are not the same as 38 degrees, it is the same kind of a magnitude.

For the sake of simplicity the drop was taken as adiabatic; had it been taken as varying from the adiabatic, the computation would have been somewhat more intricate, but the advantage of counting in degrees instead of in pounds would have been equally evident. This is one of the advantages that has come from the attention that steam-turbine engineers have given to theoretical considerations.

C. H. PEABODY.

Boston, Mass.

Why Are There Not More Writers among Engineers?

It is surprising that so few engineers take an interest in contributing to the "practical information" columns of engineering journals. I can see no method that could be a greater help to young engineers than following the teachings obtained through the experience of others. It behooves engineers to give out all the information they can to help the younger and less experienced men, as well as for the betterment of the trade in general. At times it seems that there exists a feeling of jealousy among the engineers, for there are many well informed men in the country who could contribute a few lines now and then which would be of much help, but from these people little of the much desired information is to be had.

To illustrate how a few lines along most any line might help someone, I will relate a personal experience of some years ago. I had seen much of steam-pipe covering in place, also covering on heaters, separators, etc., but it was some time before I obtained exact facts as to applying the covering to traps, heaters, small boilers, etc. I was working as oiler with an experienced engineer and under a first-class chief, but found that they lacked the best knowledge along this line as they attempted to cover a trap and separator by applying a paste so fast that it came off before having a chance to dry and they were forced to give up the attempt. I concluded it must be quite a trick and set out to obtain the required information. After looking over many engineering papers and books, I secured very little information on the subject, but by watching out and picking up points here and there, along with experiments of my own, after about two years I had satisfied myself that I could do the work. For the benefit of those that have not acquired that knowledge I will give a description of my method of applying covering to small boilers, heaters, etc.

To start with, it is best to have the surfaces warm (not hot) and free from dust or grease. Apply the covering in coats; the first coat, being the most difficult, is put on by having the covering mixed with good paste and applied by pressing it against the surface in small balls until the entire surface is covered, with the edges of each ball just touching. After this has dried, go over it and plaster, by hand or trowel, between the balls until the whole surface is covered, and continue in this manner until the required thickness is reached, which should not be less than ½ inch. Now, bind it tightly with a fine iron wire or a small-mesh chicken-wire netting and apply another coat of asbestos about ½ inch thick. After this has dried repeat the operation until the required thickness is obtained. The last coat, which is about ½ inch thick, consists of a mixture of one-half asbestos and one-half portland cement. This should be applied with a trowel, a smooth finish being attained by dipping the trowel in water. This gives a hard finish which is much easier to keep clean and resists moisture from the dripping water and condensed steam better than a covering of canvas. The great trouble in applying covering of this kind is that the inexperienced man is apt to put on too much paste in the beginning and not allow it to dry before applying the second coat. By taking time and having patience most anyone can make a good job of covering any kind of a vessel.

Gas and gasolene engines seem to be a sort of bugbear to a great many steam engineers, but I believe that if they would read a good engineering journal regularly and concentrate their minds on it as much as they do on the baseball news, it would not be long before they would be walking around with a chip on their shoulders looking for someone to say something against the internal-combustion engine.

When a good engineer finds that his boiler needs washing he hustles into his jumper and overalls and cleans it. Just so with the engineer that keeps abreast of the times; when he sees a new thing that looks good, he gets busy with the books and magazines and learns all about it. These engineers are the ones that are holding the good jobs, from whom we hear no complaint and from whom the less important members of the trade receive but little information as compared with what they should.

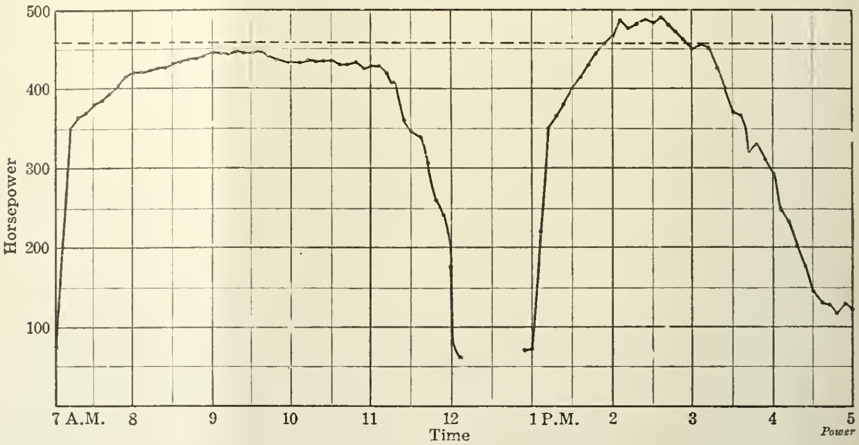
One of the subjects that I would like to see discussed more in the columns of engineering journals is that of handling men and running the business end of plants. I believe there are any number of men whose experience along these lines would be of interest and value to many readers of Power.

B. P. Gage.

Keshena, Wis.

The Engineer Proved That He Was onto His Job

In a certain manufacturing plant in New England, it was told around that out in Illinois, where another branch of the firm was situated, they were producing the same article much cheaper than we could. Of course, the old man began to wonder as to the cause. First he attributed it to too much clerical, office and drafting-room expense, but after going into the matter carefully he decided that it would not help matters much even if a couple of men were dispensed with.



Load Curve Showing Peaks at Wrong Hours

The shop had good machines, employed good help, had the latest ideas on piecework, the boilers were burning all the sawdust to help out the fuel bill, the engine was in good condition, and from the superintendent to the least important foreman, nobody knew why it cost more in the Eastern than in the Western plant.

It was finally decided that because the Western plant ran the machines on each floor by an independent motor, whereas the home plant took all the power direct from the engine by means of belts and shafting, that much power was lost through friction in driving the shafting. The only way to compete with the other plant then was to install the same system of motive power or a new electric system and do away with all the big shafting, heavy belts, etc.

The engineer was consulted and asked to take an indicator diagram about five minutes to seven in the morning, another ten minutes past seven and another at noon. Each man was to leave his machine running for a few minutes in order to find out the power required to run the shafting, the peak load and the load of the shafting and machinery combined. The engineer did as he was told, finding that it required 75 horsepower to run the shafting, and that the peak load did not come at a quarter past seven but near 10 a.m. and 2 p.m. This fact, however, he kept to himself, reporting only a peak load of 475 horsepower, and a load of 150 horsepower for the shafting and machinery after the whistle had blown. After sending these figures to the office he began to think that perhaps the indicator was wrong or that some mistake had been made in using the planimeter, for why did the load drop off at four o'clock?

The next day, after looking over every joint on the indicator, and commencing at ten minutes to seven in the morning, he took a diagram every ten minutes all morning until ten minutes past twelve, then again at ten minutes to one until ten minutes past five at night. He put these in his pocket when he went home to supper and that night figured them all out. After getting them into shape he plotted the curves shown in the accompanying diagram; these showed that the fellows let up on their work by three o'clock and visited until the whistle blew. These were the cold facts and it was up to him to disclose them.

The next morning after the superintendent had been in the office a few minutes, the engineer walked in and showed the results of the day before, pointing out where the actual loss was. After the engineer left, there were some lively doings in the office where the schedules were made up. They figured out the area, as shown by the dotted lines, then the actual amount covered by the curves and decided that a 15 to 25 per cent. cut in the wage rates would help matters. The notice went around and on Monday morning the new scale went into effect. Strange to say, it was actually welcomed, except by some of the old fossils who sighed a little. Some of the private boxes under the benches had to yield up their contents for a few days, but the men very soon had everything going on as before the cut. This proved to be where the extra expenses was located. As for the engineer, I do not know whether he received a raise or not, but I do know that if he had asked for one he would have received it.

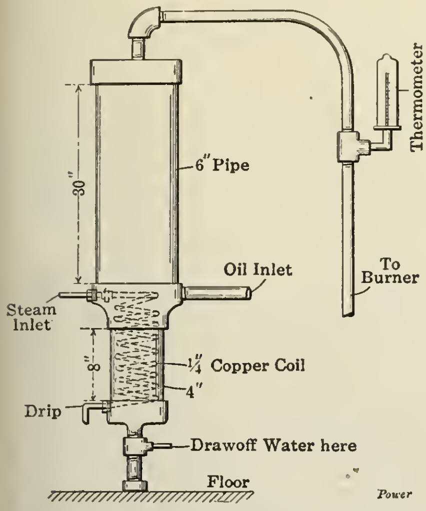
G. C. Abbe.

Brooklyn, N. Y.

Separating Water from Oil

I have found that in using oil fuel, enough water sometimes gets into the oil to cause much annoyance. The oil is shipped to this city by boat, and at times three to five barrels of water are present in a tank containing twenty-five barrels of oil.

As a usual thing the suction pipe reaches to within 2 to 4 inches of the bottom of the storage tank, and after a few barrels of water get into the storage tank, it is not unusual for the oil pump to draw nothing but water during several strokes. This becomes a nuisance when one is using a small fire in a large furnace, as a very little water will put out the fire. The furnace being hot and no flame being present when the oil is again ejected from the burner, a gas is immediately generated which is decidedly unpleasant as well as dangerous.



SEPARATOR

After having had a siege of this sort for several days, I thought out and constructed the separator shown in the accompanying sketch, which requires very little room and not much ingenuity to make. The separator is made of 6- and 4-inch pipe with ¼-inch copper pipe for the heating coil. Inside the 4x6-inch reducer a ¼-inch ground union is placed and the lower end of the coil is tapped into a 1¼x4-inch reducing coupling.

When there is any evidence of water in the oil, steam is turned into the inlet and the drip pipe is throttled just enough to allow the condensation to escape. The oil inlet is on a level with the steam inlet and when the oil becomes hot the water separates very readily and can be drawn off through the drain provided for that purpose. I have used this apparatus for over two years and have found it very satisfactory.

H. R. ROGERS.

Seattle, Wash.

Device for Taking Up Slack in Indicator Cord

Nearly all of the steam-engine indicators upon the market are equipped with a detent motion, but in the majority of cases this means simply a pawl which can be thrown in to engage with teeth cut on the bottom of the indicator drum and no provision is made for taking up the slack cord between the indicator drum and the bushing of the reducing wheel. This slack cord is very apt to catch on some part of the instrument and either break or cause damage, much to the annoyance of the engineer. There are several devices upon the market for taking up the slack in the cord but after trying several I found them unsatisfactory on account of their short life, due to the use of springs which soon weakened or broke when used on high-speed engines, or otherwise they were clumsy and in the way of the operator. I therefore submit the following description of a device for taking up the slack in the cord which has neither of the foregoing undesirable features and which can be made by any engineer.

The plate shown in Fig. 1 is cut out of a piece of ⅛-inch sheet brass with the diameter *A* equal to the diameter of the indicator drum, the arm *H* extending out a distance equal to *A* and the arm *J* ¾ inch long. A ⅛-inch hole is drilled at *C* and the swivel pulley *P*, Fig. 2, is secured by riveting the post over on the under side of the plate. This pulley may be purchased from any indicator manufacturer, but the post should be just high enough so that the pulley guides the cord in the center of the band of the indicator drum when the drum is clamped by the nut *N*, Fig. 3. The bearing *F*, Fig. 2, for the plate to rotate upon, is made from a piece of ordinary 3/16-inch brass tubing, such as that used on oil pumps. A piece of this tubing, 11/16 inch long, is used and six ¼-inch slits are cut in each end; then after being placed in the hole *B*, Fig. 1, these ends are bent over so that they form a flange, as shown at *F*, Fig. 2. Care should be taken to see that the plate moves easily in this bearing after it is clamped in position. The device is now secured in place and from the hook at the end of the arm *H* an ordinary light rubber band *R* is stretched to some stationary part of the reducing motion. (I fasten it to the screw that adjusts the tension of the drum spring).

Fig. 4 illustrates the operation of this device, looking down at the top of the indicator drum. When the pawl *D* is thrown in and the drum stops, the rubber band *R* pulls the arm *H* and consequently the pulley *P* to the right to the position shown by the dotted lines *H'* and *P'*, keeping a tension on the cord at all times and winding the slack around the drum. When the pawl is disengaged, the drum spring, being stronger than the rubber band, holds the pulley in its normal position.

I have used this take-up device on several different indicators both at high and low speeds and have always found it satisfactory in every way and it will not distort the diagram nor spoil the neat appearance of the indicator.

ROSCOE I. DUNTEN.

West Hurley, N. Y.

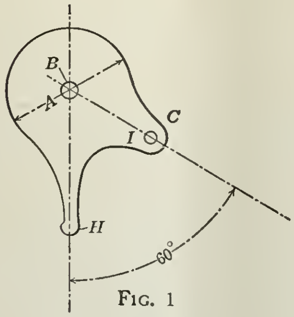


FIG. 1

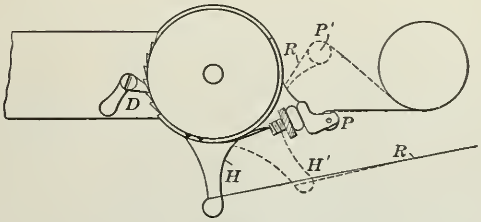


FIG. 4

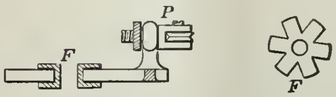


FIG. 2

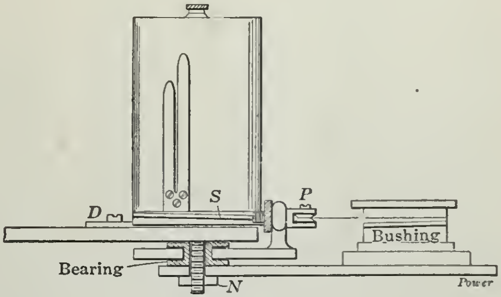


FIG. 3

DETAILS AND ASSEMBLY OF PARTS

A. V. Clayton told the Institute of Electrical Engineers last April that some 8,600,000 water horsepower are available in Sweden and Norway; there are over 7,000,000 in Japan.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Back Pressure

As R. P. Wilcox correctly states in the September 6 issue of POWER, there is much confusion regarding the term "back pressure" as applied to a heating system and to a compression refrigerating system, but his remarks seem only to add to the confusion. He says that a high back pressure or "suction pressure" in a refrigerating system helps the engine driving the compressor. He compares the engine with one driving a pump which receives its water from two tanks, one at a greater elevation than the other. When the pump receives its water under pressure, it naturally requires less power to run it than when it must "suck" the water, that is, receive it only under the pressure of the atmosphere, always providing that the speed and the discharge pressure each remain the same. But, when he asserts that the same thing is true in a compressor, he makes a mistake, as just the opposite is true. The higher the suction pressure the greater will be the power required to drive the compressor, the speed and discharge pressure remaining the same.

Let us prove this by an example. Suppose we have a single-acting ammonia compressor having no clearance, a piston one square foot in area, a 2-foot stroke and therefore a piston displacement of two cubic feet per revolution, running at 50 revolutions per minute, with a discharge pressure of 160 pounds per square inch absolute. Let us figure the power required to run the compressor first, with a "suction" or back pressure of 20 pounds per square inch absolute and then with a pressure of 40 pounds absolute. Neglect friction and assume a perfect action of the valves. In almost all ammonia compressors working on the dry-compression system, the compression is very nearly adiabatic, the cooling water in the jacket having very little influence on the shape of the compression curve of the indicator card. If we assume adiabatic compression then, according to Voorhees the mean effective pressure will be

$$M.E.P. = 4.333\, p_1 \left[\left(\frac{p_2}{p_1}\right)^{0.231} - 1\right]$$

when

p_1 = Absolute suction pressure in pounds per square inch and
p_2 = Absolute discharge pressure in pounds per square inch.

Substituting first 20 and then 40 for p_1 and 160 for p_2 and solving the equations, we find that when the "suction" pressure is 20 pounds the mean effective pressure is 53.43 pounds per square inch. When the suction pressure is 40 pounds the mean effective pressure is 65.43 pounds per square inch. This alone shows that the higher "suction" or back pressure requires the greater power, but let us see how much that will be in horsepower.

The horsepower required by a compressor is figured by the same time-honored formula used for finding the horsepower of an engine, that is,

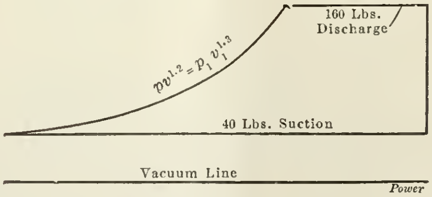


FIG. 1. DIAGRAM FOR 40 POUNDS BACK PRESSURE

$$H.P. = \frac{PLAN}{33,000}$$

When the suction pressure is 20 pounds,

$$H.P. = \frac{53.43 \times 2 \times 144 \times 50}{33,000} = 23.3$$

horsepower. When the "suction" pressure is 40 pounds,

$$H.P. = \frac{65.43 \times 2 \times 144 \times 50}{33,000} = 28.53$$

horsepower.

In case it should not be clear why the formula for the mean effective pressure should be true, a glance at the accompanying diagrams may make things clearer. Fig. 1 is the theoretical indicator diagram for the case in which the "suction" pressure is 40 pounds per square inch absolute, and in order to show the difference more clearly, Fig. 2 was drawn for the case in which the suction pressure is 10 pounds. They are both drawn to the same scale and the compression curve in each is the adiabatic-compression curve for ammonia drawn according to the formula,

$$p v^{1.3} = p_1 v_1^{1.3}$$

A glance at them will show that the diagram for the higher "suction" pressure has the greater area and consequently the higher mean effective pressure.

If we were to reduce the suction pressure to zero, neither the suction nor the discharge valves would open and the piston would move back and forth without doing any work and without requiring any power. Then, if we were to raise the suction pressure, the mean effective pressure would also be raised. But, we know that when the suction pressure is equal to the discharge pressure, the mean effective pressure and the power required will again have fallen to zero, so we see that somewhere between the limits of zero suction pressure and the suction pressure equal to the discharge, there will be a certain "suction" pressure at which the mean effective pressure will be maximum. This can be shown to be the case when,

$$\text{Suction pressure} = \frac{\text{Discharge pressure}}{3.113}$$

From this we see that when the absolute suction pressure is higher than $\frac{1}{3.113}$ of the absolute discharge pressure, Mr. Wilcox is right and an increase in the back pressure will help the engine. But, I do not believe there are many plants that are regularly run with a suction pressure as high as that and for all lower suction pressures an increase in the back pressure requires an increase of power if the speed and the discharge pressure remain constant.

In all books on refrigeration we are advised to carry as high a back pressure as we can; but if a high back pressure takes more power, then where is the gain? The answer is simple. The refrigerating capacity of a compressor depends on the weight of ammonia gas it can take from the cooling coils and force to the condenser. In the case of the higher back pressure we have a denser and heavier gas, and the increase in the weight of the gas handled and the consequent gain in refrigeration much more than offsets the greater power required. In the first example, the compressor

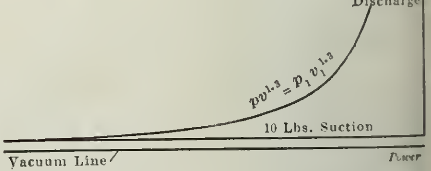


FIG. 2. DIAGRAM FOR 10 POUNDS BACK PRESSURE

handled 2 cubic feet per revolution or,

$$50 \times 2 = 100$$

cubic feet of gas per minute. According to Wood's ammonia tables, 1 cubic foot of saturated ammonia gas at 20 pounds pressure weighs 0.0744 pound and 1 cubic foot at 40 pounds pressure weighs 0.144 pound. Then, when the suction pressure is 20 pounds per square inch, the compressor handles

$$100 \times 0.0744 = 7.44$$

pounds of gas per minute with an expenditure of 23.3 horsepower or

$$\frac{7.44}{23.3} = 0.319$$

pound of gas per horsepower per minute. When the suction pressure is 40 pounds per square inch, the compressor handles,

$$\frac{100 \times 0.144}{28.53} = 0.505$$

pound of gas per minute per horsepower.

Although we come to the conclusion that a high back pressure in an ammonia-compression refrigerating system requires more power per stroke of the compressor, the total power is less because owing to the increased refrigerating capacity of the compressor the engine and compressor can be run so much slower. So, after all, a high back pressure does help the engine though the action of the gas in the compressor is in no way analogous to the action of the water in the pump.

A. E. Mueller.

Mayville, Wis.

Verdict on Canton Explosion

In regard to the recent explosion at anton it is my opinion that the operaion of cutting in a boiler cannot be perormed rapidly enough to cause waterammer. With an ordinary gate, globe r angle valve, the opening movement is ecessarily comparatively slow, and unss the operator "cracks" the valve and hen spins the wheel open at a terrific peed, I do not believe that conditions ver exist in common practice under hich waterhammer can take place.

If waterhammer *is* possible when a aster valve is opened, even with haste, hy is it not present with all its alleged estructive effects every time the safety alve blows? Where is the engineer ho has not seen a boiler suddenly rease a quantity of steam through the afety valve equal to its generative capaty, when, for any reason, all other outts were wholly or partially closed? If tting a boiler in on the line causes aterhammer, then a safety valve (escially a pop valve) will do it; and if is is so, why is it that more engineers d firemen do not take the most direct ute to "Canaan"?

If waterhammer were going to take place at all, why should it wait till the master valve is two-thirds open, as was shown by the testimony at Canton? The theory advanced by John Dodd, that possibly Austin went up on the boiler to tighten the gland on the master valve just previous to the explosion is extremely probable. Valves are prone to leak badly under just such circumstances, and there is not a man in the business who has not done exactly the same thing that Mr. Dodd described.

The average coroner thinks that "what he says goes," and that the public has to accept his verdict as final, whether it is within 40 miles of the truth or not. Usually the position of coroner is passed out to some good henchman of the party in power "for services rendered," and he is elected by a huge majority, gets into the public eye, gets chesty and proceeds to render decisions in accordance with either his own conceited ideas, or those of the demagogues who put him in power.

George H. Wallace.

Racine, Wis.

The Expansion Valve

In the August 30 number of Power, Mr. Nash makes what seems to me to be an erroneous statement regarding the action of the liquid anhydrous ammonia in a refrigerating system when it is passing the expansion valve. He says that "Many seem to think that the valve is for the purpose of regulating the rate of expansion of the ammonia liquid; that this valve is to be opened only enough to allow the liquid to instantly flash into a vapor." Considering the promptness with which the temperatures respond to the adjustment of the expansion valve it would seem to indicate that the action as quoted is exactly what takes place. Taking for example the absorption system having a brine cooler consisting of a vertical cast-iron cylinder with interior coils through which the brine is circulated, the expansion valve, which will connect with the bottom of the cooler, will be adjusted to cause the liquid to wiredraw through it and pass in as a fine stream. The liquid being relieved of pressure will immediately vaporize and in so doing absorb heat from the surrounding brine. If the liquid is allowed to pass in faster than it will vaporize it will accumulate in the bottom of the cooler and all further control with the expansion valve is lost. The evaporation would depend on the back pressure which, from excess of vapor, would go up and with it the temperatures. That the liquid ammonia does not in ordinary operation lie in the bottom of the cooler is indicated by the fact that the brine temperature immediately goes up and the back pressure down if the valve is closed. Furthermore, if the blowdown valve in the bottom of the cooler is opened no anhydrous ammonia will pass out, although if the machine has not been making dry gas a very strong aqua solution will be obtained. If, on the other hand, the expansion valve is opened a minimum amount so that expansion takes place at the valve or in the pipe leading from it, it will frost back of the valve excessively. Of course, it would be impossible to view the action of the ammonia in the cooler, but a study of conditions obtained would indicate that the flow of the liquid is regulated by the expansion valve to keep up a given back pressure, that the ammonia instantly vaporizes, producing large quantities of gas of intense cold, which, passing up through the brine coils, produces the refrigerating effect. The gas returning is cold enough to frost the return pipe up to the absorber, and its proper adjustment is usually determined by the point to which the frosting extends.

Lewis C. Reynolds.

Willard, N. Y.

Rope Drives

In his letter in the September 13 issue commenting upon my article on "The Design of Rope Drives," published in the August 16 number, Henry D. Jackson hits upon what is probably the chief point of contention in connection with the subject. The curve referred to was plotted from cotton-rope data for which the firm of T. Hart, of Blackburn, England, is the authority, and it is certain that it represents orthodox ideas on the subjects.

Since reading Mr. Jackson's letter I have consulted many textbooks and have spoken to several engineers who have installed rope drives, and I find that they agree in putting the limiting speed of ropes of any kind at about 4800 feet per minute. Above this speed it is always assumed that, owing to the effect of centrifugal force, the power it is possible to transmit commences to fall off. I am aware that Mr. Kenyon holds that this is fallacious, and as one of the principal makers of ropes his opinion must be accorded full weight. Indeed, he has at various times cited particular cases in which ropes of his manufacture were actually transmitting their full power at 7000 feet per minute. This is a difficult fact to get over, but, on the other hand, it is necessary also to give full weight to the opinions of other eminent rope makers who definitely affirm the opposite. Where experts in their particular business disagree so widely it is impossible for the ordinary engineer to judge, and I am afraid that we must content ourselves with retaining an open mind until such time as the matter is definitely cleared up one way or the other beyond dispute. Personally, I prefer in the meantime to adhere to the more general conception that the limit is somewhere below 5000 feet per minute because we know we are safe at that. So far as belt transmission is concerned we know definitely that there is a limiting speed, but, obviously, with a rope running in a groove the problem is quite different, and Mr. Kenyon founds his contention that centrifugal force is inoperative, at any rate, up to 7000 feet per minute, upon the basis of "stiction" in the grooves. I agree with Mr. Kenyon that cotton rope should be preferable to hemp in this respect owing to its softness and tendency to cling. The C. W. Hunt Company, of New York, publishes data for hempen rope, and if these are referred to it will be seen that they put the limiting speed at about 4800 feet per minute, as, I note, does also the Plymouth Cordage Company.

Francis H. Davies.

London, England.

Trouble with a Pumping System

I agree with Mr. Kropidlowski, in the September 20 issue, that the original trouble must have been with either the pump or pipe line as the hight of the suction and the loss of head, resulting in friction in the long length of suction pipe, does not bring the lift beyond the limit of suction. The loss of head due to friction in the suction pipe in the original layout would be

$$h = 0.062\, f \frac{L\,V^2}{D}$$

where f is a constant determined by experiment and which for new 4-inch cast-iron pipe is 0.0056; L is the length of the pipe in feet; V is the velocity of the water in feet per second and D is the diameter of the pipe in feet.

Assuming that there was an ordinary velocity in the suction pipe of 150 feet per minute or 2.5 feet per second, in the 920 feet of suction pipe there would be a loss of head due to friction of

$$0.062 \frac{0.0056 \times 920 \times 6.25}{0.66} = 3$$

feet. This means that the head that the pump has to overcome in order to bring the water to the pump is

$$19 + 3 = 22$$

feet, which shows that the hight of the pump is well within the limit of suction.

I would, however, criticize the discharge piping, as shown in his Fig. 2. In the first place, I fail to see why the piping was put up with so many bends. There is quite a loss of head due to friction in the extra pipe and especially since this pipe is reduced in diameter to 4 inches part of the way. If the discharge pipe were run direct from the pump to the base of the tank tower and connected to the riser with an elbow, and if the pipe were made 8 inches in diameter right up to the tank, instead of there being a frictional head of about 98 feet there would be one of about 5 feet only to overcome.

It is probable that the saving in power which would result from a change in the piping arrangement would in a short time pay for the cost of the change.

George F. Atwater.

South Norwalk, Conn.

One or Two Boilers

In answer to Mr. Lindblade's question in Power for September 13, I will say that he was right.

There are many reasons why two small boilers should be installed instead of a larger one. First, economy. Second, less smoke will be created with the smaller ones as the heat can be kept at a much higher degree, and with the aid of automatic steam jets and air regulation a nearly smokeless condition can be maintained. Every engineer knows that much better results can be obtained by running return-tubular boilers with a slight overload than by running them under their rated capacity. Third, if an accident happens to one boiler the other can be run while repairs are being made.

G. A. Gustafoon.

Chicago, Ill.

In Power for September 13, Mr. Lindblade asks for information regarding the proper boiler installation for a school-heating plant.

I am sure that two boilers would make the best installation, because one can be cut out in the fall and spring and fuel can thereby be saved. By keeping a hotter fire under a smaller boiler it is more easy to keep the smoke down and save the washday brigade on Monday mornings.

Dan A. Nickel.

Austin, Ill.

In answer to O. Lindblade's question in the September 13 issue regarding the choice of boilers for a school building, I would say I would install two small boilers in preference to one large one. If, for instance, he needs a boiler capacity of 500 horsepower, the installation to make is two 300-horsepower boilers, for the following reasons:

While the initial cost is more for two small boilers than for one large one, the increase in economy of coal will overbalance the difference in cost within a few years. Boilers give the best economy at normal rating or under. For this reason the two 300-horsepower boilers will give the better economy. When cleaning fires, the steam pressure can be kept constant with two boilers, for one can be worked while the other is being cleaned. In case of a breakdown and when one boiler is out to be cleaned, the other, by forcing it, will carry the load.

W. C. Fisher.

Brooklyn, N. Y.

In the issue of September 13, O. Lindblade asks, "Should One or Two Boilers be Installed?" in a heating plant in which he is interested. With only the information given in his letter this is hard to determine. It would seem that the reason for using two boilers which is advanced by him is not the strongest argument that could be put forward.

While in mild weather there might be a small saving in fuel by the use of a small boiler instead of a large one, the large boiler should need less fuel than the two small boilers in severe weather, and as long as there would be the necessity of having fires under both boilers, there would be more loss with the fires banked in both than there would be if the fire had to be kept up in only one. Whether there would be much saving would depend upon the proportion of mild to severe weather at the city mentioned.

The first cost of the two boilers with their settings and the additional piping would be more than for one boiler, and unless there was enough saving in fuel to offset this additional cost, the only reason for purchasing the two instead of one would be on account of the ability to supply heat from one if the other were out of commission for repairs. There could be at all times heat from one of the small boilers, while if the single large one had to be shut down there would be no heat at all.

While the ordinary low-pressure boiler needs repairs comparatively seldom, necessity for cutting it out may arise at a time when heat is needed and if there is only one boiler no heat can then be supplied. This, rather than the saving in fuel, should be what would decide the question. Whether it is wise to pay the extra amount for the two boilers so as to get the ability to have heat at all times depends upon how much more the two boilers would cost than the single one.

G. H. McKelway.

Brooklyn, N. Y.

Referring to the article by Mr. Lindblade in the September 13 issue, entitled "Should One or Two Boilers Be Installed?" I beg to say that two boilers should be installed. As long as there are boilers there will be accidents that will require them to be shut down for repairs or cleaning, and these accidents usually happen when least expected. If two boilers are installed it will reduce the danger from a shutdown one-half. I know of two cases where one boiler was installed in school buildings. In one case the boiler became disabled and school had to be dismissed on account of cold. The other case was in an orphan school that housed 400 children. The boiler became disabled in such a manner that it required two days to make repairs. The school could not be dismissed and it was only by quick action that the children were saved from freezing o catching severe colds.

Another reason why two boilers should

be installed is that when a boiler is out of service in cold weather and the piping gets cold, there is more or less danger from frozen pipes, and the expense of repairs in one shutdown in cold weather will more than make up for the difference in the cost of one large boiler and two small ones.

W. H. Keller.

Austed, W. Va.

In the September 13 issue of Power Mr. Lindblade asks which would be better, to use one large boiler or two small ones for heating a school.

In choosing the number and size of boiler units there are two things to be taken into consideration, reliability of service and efficiency of operation. In all cases where possible, it is of the utmost importance to have a boiler plant so laid out that if it is necessary to shut down a boiler for repairs or cleaning, the operation of the plant as a whole will not be interfered with. In the case under consideration it would be unthinkable to install only one boiler, for if the boiler should go out of service for any reason it would mean no heat until operation could be restored. If two boilers were installed, heat could be supplied at all times, unless both boilers should go out of service at the same time, which is highly improbable. Also, with one boiler it would only be run at its full-load capacity for a very short period of each year and the rest of the time at a fraction of its rated capacity, with a corresponding decrease in efficiency. With two boilers it would be possible to run one at full-load capacity practically all of the time; and when the weather required the use of both boilers, they would never be running at less than 50 per cent. of their rated load and often nearer full load.

W. L. Durand.

Washington, D. C.

Position of Throttle

In the issue of September 20 are a short article and a picture of a Corliss cylinder setting forth the supposed advantages of placing the throttle valve with the stem and wheel extending diagonally over the corner of the cylinder and apparently far beyond the head-end valve gear. The article states that, "It is certainly easier o handle the valve gear with a throttle of this design than if it were of the hort type," etc. I disagree, claiming hat it is very inconvenient and the ngineer will get many a bump by rising p against the wheel which is in the way, or it is at this point that the engineer must stand to handle the starting bars. It has been my experience that nearly all ngines of over 150 horsepower are fit-ed with a bypass around the throttle and n large engines, up to 56-inch stroke, ith a quick-opening valve the lever of hich is in a convenient position to reach, sually near the side of the cylinder jacket and back of the wristplate and about parallel with the main throttle stem on the type of engine shown. This valve is used in warming up and stopping or moving the engine part of a stroke.

In stopping, the main valve is closed tight; then, as the engine slows down, the reach rods are unhooked, the hand levers put in and the bypass operated to admit steam as required to stop the engine in the proper position.

Imagine opening an 8- or 10-inch valve or even a 6-inch on the end of a long stem with one hand, while the other one is busy with the bars, and admitting a little steam to block the piston to keep the crank off the center and then closing it tight and quickly enough to keep the engine from making a revolution or two the other way. If it looks easy, try it and you will know differently.

The proper place for the throttle-valve stem on a Corliss engine (horizontal) is at the side of the cylinder and with the wheel just inside of the wristplate with the bypass back of and parallel to it, the bypass being on the side of the cylinder toward the head end.

J. Case.

Hyattsville, Md.

Trouble with the Belt

In connection with the belt-drive problem presented by Mr. Hamill in the September 27 issue, I offer the following suggestions:

There seems to be no reason why the method proposed cannot be used unless the shafts are located exceptionally close together and the belt is therefore very short. In order to guide the belt to prevent it from running off the pulley, the idler should be placed close to the driven pulley and in this position it should keep the belt on and at the same time not be near enough to the cross to cause any troublesome friction. It would not be practicable to use a cone pulley for the idler because different points on its surface would have different peripheral speeds while the speed of the belt is, of course, uniform over its entire width. This would cause friction between the belt and the guiding pulley and probably result in more trouble than if no idler were used.

If the belt is tipped at the point where the guiding pulley is applied, the difficulty may be overcome by tipping the idler—that is, setting it at an angle—to correspond to the angle of the belt. The belt may then be guided by adjusting the idler to tighten it on the side toward which it should run. The belt will run toward the tight side, following the principle that a belt always climbs to the largest diameter of its pulley.

In case the pulleys are too close together to use an idler, a fork or other device used for guiding might be employed. The chief objection to this is that the edge of the belt running against a fixed guide is very liable to become worn by the constant rubbing, especially if much power is transmitted. For an 8-inch belt running on 48-inch pulleys having a speed of 400 revolutions per minute, the guiding pulley or idler would probably be more satisfactory.

S. Hurley.

Scranton, Penn.

License Laws

Judging from his letter in the September 27 issue, I do not look at this subject in the same light as Mr. Nigh does. I believe that every State should have license laws. They not only make an engineer get down to business but also make him a better man for the position he holds and qualify him to secure a more responsible and better paying one. They also serve to put the "would-be" engineer out of the licensed man's way.

I not only believe that each State should have a good license law but also a good law for boiler inspection.

It is true, as some say, the license does not seem to reduce the number of explosions of boilers. Usually the blame falls unjustly on the shoulders of the engineer. Often he gets killed in the explosion and is not able to tell what he told his superior about the condition of the boiler. The engineer is blamed and thus, indirectly at least, the license laws are held to be deficient. In reality the inspection law or the lack of one is the cause for the bad conditions.

If every State had a good inspection law and good inspectors, I guess some of the boiler owners would have to sit up and take notice. Conditions about the plant would probably be improved and in this way things would be better and safer for the men.

A man I know once worked for a firm which owned some boilers that were in rather bad shape. He told the superintendent that new boilers should be installed. The latter inquired why the engineer thought that the boilers were not good; they had been in service *only twenty-five years*. The engineer said that when he started to calk the leaky girth seam of one, the leak spread, showing that many of the rivets were not holding properly. The engineer was instructed to calk that seam. He did; he made the seam tight and then quit his job. A few weeks later one of the boilers moved and moved with it everything that was in its path. It moved the south wall and part of the east wall of a four-story brick building right out onto the railroad tracks. The damage amounted to about $15,000. The engineer held a license and did his part. He warned the owners of the danger. Decidedly, the license law was in no way to blame for that disaster.

Clinton Miller.

Covington, O.

Reducing the Labor Cost in a Plant

Every owner of a power plant is interested in the reduction of the operating expenses and it behooves the engineer to discover a method of effecting the reduction. If the cost of handling the coal is reduced to the minimum, an economical method of disposing of the ashes should next receive attention. There are certain seasons in the year when the ashes may be disposed of at a profit, but during the winter months usually the ashes are an expense and the labor in handling them is quite an item. In some localities quantities of ashes are consumed in cement pavements, as well as for fireproofing and concreting. In other places, they are simply used for filling in, or are wheeled out on a dump and in the spring are carted away.

> *Under the old system the ashes were dumped into the cellar, from which they had to be shoveled up to the street level and carted away at a cost of $1.50 per wagon load. By the installation of an ash elevator this figure was greatly reduced.*

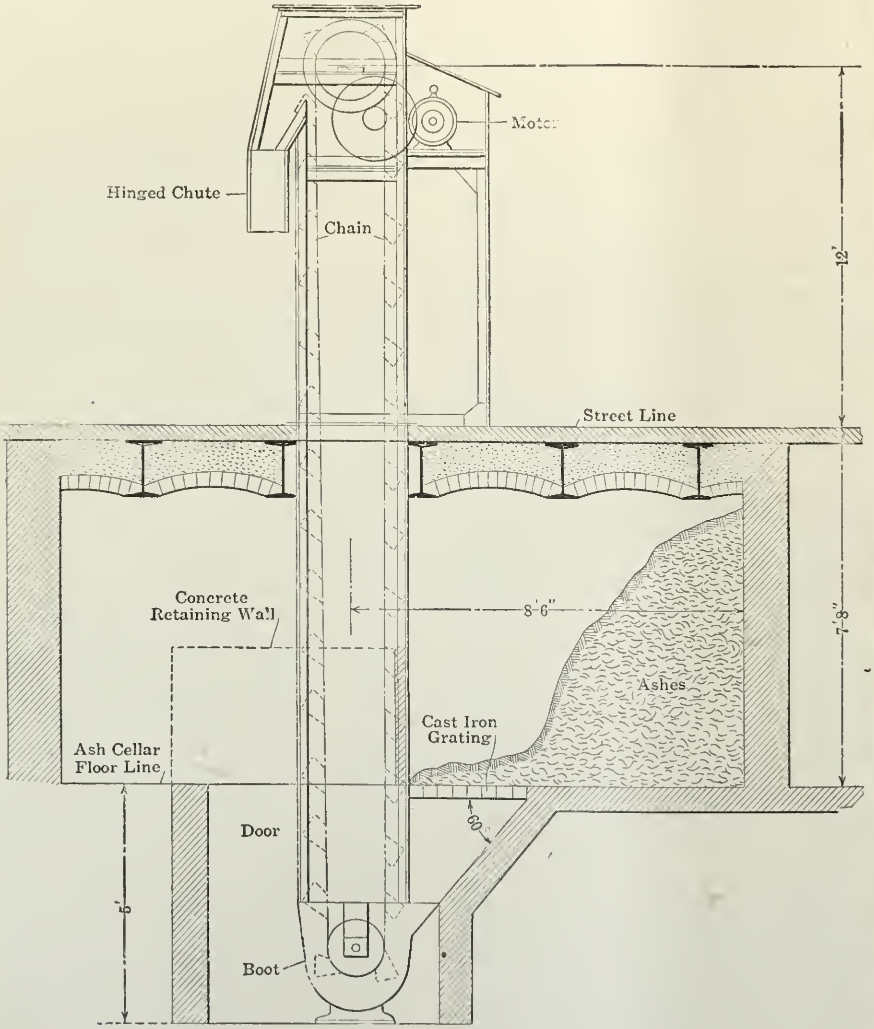


FIG. 1. COMPLETE LAYOUT OF THE CONVEYER

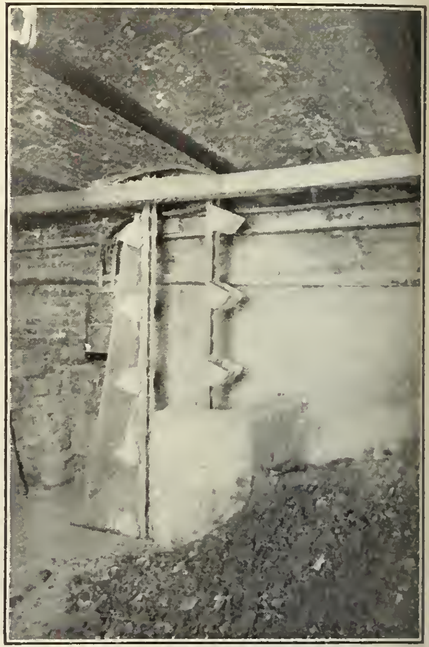


FIG. 2. CONVEYER INSTALLED IN CELLAR

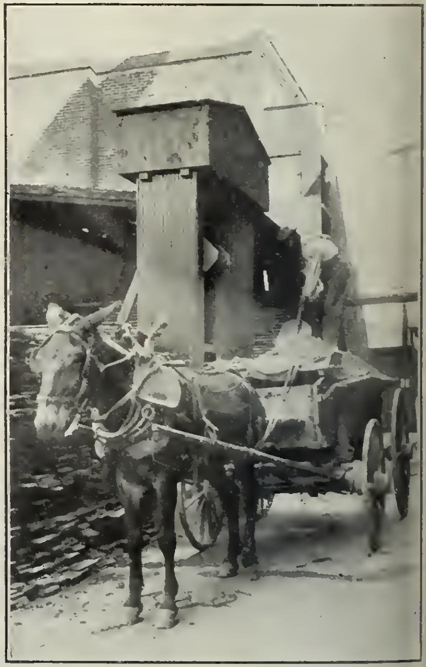


FIG. 3. METHOD OF CHUTING ASHES INTO WAGON

A very unique and practical system was recently installed in the power plant of the Lilly Manufacturing Company, at Columbus, O. For several years past this concern has stored the vast accumulation of ashes in a cellar under the alley pavement, adjoining the boiler room. Owing to the storage capacity being limited, they were obliged to employ two extra men with a wagon to remove these ashes at a cost of $1.50 per wagon load. After a period of time, it was discovered that this method was particularly crude and very inconvenient; one man was usually stationed in the cellar, shoveling the ashes through a manhole into the roadway overhead, while the other man was obliged to shovel the ashes into a wagon from the alley pavement. The process of hand shoveling was, therefore, a double one as the ashes had to be hauled away by a wagon.

The company grew tired of this unprogressive arrangement and decided to install a conveying system. Fig. 1 shows the method adopted for handling these ashes. A Jeffrey bucket elevator slightly over 23 feet between centers was used consisting of a chain with 32 malleable-iron buckets and provided with a concrete boot, or "loading leg," on a level with the floor line of the ash cellar. This boot is provided with a cast-iron grating which prevents large clinkers from entering the elevator and choking it. These clinkers are cast aside and broken up, after which they are delivered to the elevator. The latter is driven through spur gearing by a 3-horsepower electric motor.

Fig. 2 is a reproduction of a flash-light photograph of the cellar; a brick wall having been constructed around one side of the conveyer to enable the ashes to be piled up.

Fig. 3 shows the method of chuting the ashes directly into the wagon. The chute is hinged, which arrangement allows it to be dropped out of the way on account of the narrow passageway. The elevator has a capacity of loading 5 cubic yards of ashes in less than 20 minutes.

The Street Railway Question Box

Attention is paid to a number of practical questions such as Fuel Economy from Bonus System, Use of Boiler Compounds, Lignite Coal as Fuel, Flywheel Insurance, Forced Draft with Chain Grate Stokers, etc.

The question box of the American Street and Interurban Railway Engineering Association contains some items of interest to POWER readers.

What increase in fuel economy have you been able to accomplish under regular operating conditions? Upon what bonus, or premium, if any, has this been based?

In the six answers received there is not a bonus. All but one have been able to make some increase in their fuel economy, one 8 per cent. by the use of a "fine shaking grate," one 5 per cent. by "improving boiler conditions," and the others unspecified amounts by attention to details in the boiler room. One lays much of the advantage gained to the use of recording gages wherever possible; and one to purchasing the coal under specifications in which is inserted penalty for furnishing poor coal and bonus given for furnishing good coal. The specification is based on B.t.u., percentage of ash, sulphur and moisture.

Is the use of boiler compounds to be avoided if possible? Why?

Of the eleven answers received to this question four favor the use of compound, and are evidently inspired by favorable experience with it. Five advise against the use of compound "because of its deteriorating effect on steel in the boilers," "because of expense and danger of foaming," "on account of its pitting the tubes" and "because of unnecessary expense." The wording of the question invites an affirmative answer, for naturally anything which involves trouble and expense, and the use of which may lead to second troubles is, of course, to be avoided *if possible;* that is, if its avoidance does not nvolve greater expense and trouble than ts use will save and prevent. One respondent says that "the use of boiler compound should be avoided if possible, but if necessary to use same, care should be taken not to use any more than is absolutely necessary. Practically all boiler compounds will attack the packing in the stuffing boxes, on the steam valves and oints on steam mains. In some plants where economizers are used it will attack the caps and joints, starting them o leaking. Where it is necessary to use compound in large quantities on account f the condition of the feed water, I ave noticed that the compound has affected the walls and piston in the cylinders and caused what is commonly called 'grooving.' Therefore, there are many reasons why boiler compounds should not be used, but in a large number of plants the condition of the feed water is such that the use of compounds cannot be avoided, as it will prevent the formation of scale in the boilers to a certain extent."

Still another says that "the use of compounds should be avoided if the water is such that you can do so. Otherwise, the very highest grade boiler compound should be used, because poor quality may cause steam pipes to eat through and may damage cylinders." He has tried many kinds and found one which meets his case and keeps his boilers in good condition.

What is the nature of tests commonly applied at the boiler rooms of member companies by insurance inspectors as a basis for certificates guaranteeing safe working pressure on boilers?

The nine replies to this question simply brought out the fact that the hammer test is ordinarily used and that the inspection is circumstantial and thorough.

Have you ever used lignite coal? If so, what is your opinion of it as a fuel?

Of the nine answers to this question one, from Texas, says that lignite compares favorably with any other kind of fuel. The other eight simply assert that the writers have never used lignite.

How can the air space between the bridgewall and grate bars of a chain-grate stoker be automatically stopped or reduced in area?

H. Gilliam, electrical superintendent, New York, New Haven & Hartford Railroad Company, says: "As far as I know there are two types of chain-grate stokers, the incline and the level type. The incline type generally has a stationary dump grate attached to the back wall. The other type generally uses overhanging bridgewalls. The area can be adjusted by moving the grate closer or farther away from the back wall, as the chain-grate stokers are mounted on a truck. I do not know of any automatic device that will make this change."

W. A. House, president United Railways and Electric Company, of Baltimore, writes: "Adjust the distance between the rear end of stoker and bridgewall so that the required thickness of fire may be carried without allowing the ashes to bank up or fall off, leaving an open space.

Can forced draft be used on a chain-grate stoker? If so, how, how?

The replies generally favor the use of induced draft, although some of the respondents think that forced draft can be used if provision is made for stopping the air space at the end of the grate.

Is it good practice to operate large flywheel engines of a reputable make without flywheel insurance?

Of the nine answers to this question four favor flywheel insurance; the other five assume that accidents will happen anyhow and that the risk is of the same order as that of loss by earthquakes, torpedo or floods. One insists, however, that the flywheel should be properly inspected at certain intervals, "say four times a year."

What should be the cost per 100 kilowatt-hours for lubricating the high- and low-pressure cylinders of a 1650-kilowatt vertical cross-compound condensing engine, operating, at full load, 75 revolutions per minute, saturated steam, 150 pounds pressure and 25 inches vacuum?

The three replies to this question show a remarkable agreement, although coming from widely different parts of the country. One from St. Louis fixes the figure at 0.32 cent, one from Baltimore at 0.30 cent, with oil at 30 cents per gallon, and one from Louisville, Ky., at 0.50 cent, with 50-cent cylinder oil for the high-pressure cylinder and 25-cent oil for the low.

Do you use a damper regulator? If so, is there any way to prevent smoke when the damper is closed?

There seems to have been some confusion as to whether this question referred to smoke coming out into the fire room, or smoke at the chimney due to insufficient air supply.

Of the ten members responding to this question four do not use damper regulators, but that used by another "pays for itself in three months' time." The general recommendation is to so adjust the chains, or so provide in the installing of the damper that it cannot close off completely, nor sufficiently to prevent the minimum draft required.

What is your experience with and your estimation of the value of grooving commutators to date?

Of the twenty replies received seventeen are favorable to the use of the grooved commutator. Two members confessed to no experience and one has had very good success with large motors which have good protection from dust, but on small motors has had trouble from oil and carbon dust. Several of the respondents are enthusiastic in its favor, crediting it with the saving of at least half the cost and maintenance of commutators and brushes, one in particular saying that "commutators can be grooved at a cost of not over 10 cents per commutator, and the grooving increases the life of the commutator from 100 to 150 per cent., as well as increasing the life of the brush by about 40 per cent."

What are the causes of broken brushes on 101-B motor and remedy?

The responses generally lay the trouble to high mica and inferior brushes, and advise the grooving of the commutators and the use of high-grade brushes, not too brittle. One says that "The probable cause for the broken brushes on the Westinghouse 101-B motor is the absence of shunts ['pigtails'] on the brush-holders, which produces an undue wear in the brush slots, and in turn such undue movement is given to the brush as leads to its being rapidly broken. The remedy is to see that the brush-holders are equipped with proper shunts and these maintained with good firm contact." Another advises the use of two brushes per holder in place of one.

Forced Draft and Boiler Economy*

By C. E. Roehl

The limitations of forced draft and its bearing upon the capacity and efficiency of boilers are discussed together with the personal equation of the fireman.

Draft, as used in its most general sense, as air supplied for combustion under steam boilers, may be obtained naturally by high stacks or artificially by a blowing apparatus. Artificially it may be produced by a reduction in pressure above the fire or by an increase in pressure below the fire; the former is usually termed induced draft, and the latter forced draft. It is in this sense that the phrase forced draft is used in the following discussion. Artificial draft is used for obtaining capacity above the normal with ordinary fuel or for securing a satisfactory capacity with low-grade fuel.

Induced draft may be regarded as a substitute for natural draft, instead of using prohibitively high stacks; but this is not true of forced draft. In viewing the relation of forced draft to high stacks it should be recognized that where low-grade fuel is used forced draft is a necessity. Where low-grade fuel means anthracite screenings, forced draft cannot be looked upon as one of two equally efficacious forms of artificial draft substituted for natural draft with high stacks, but possesses an inherent advantage over induced draft. This lies in its ability to cope with the packing of fuel on the grate, whereby great resistance is offered to the passage of air. A fuel bed of anthracite screenings, even when actively burning, offers far greater resistance to the passage of air than the walls of air ducts and ashpit connections. Each particle of coal is struck by the air currents and is moved around in layers near the grate before the air passes through the bed sufficiently to reach the incandescent particles for oxidation. This shifting of the individual particles of coal, which may be slight, moderate, or excessive, depending upon the air pressure, does much to break up the otherwise abnormally high resistance of the fuel bed. When excessive, the coal is blown from the grate and much of it is kept dancing in the furnace, sometimes to a hight of two feet. In induced draft this lowering of the fuel-bed resistance does not occur. As the ashpits are always open to the atmosphere the energy of the air beneath the fire is exclusively in potential form, save for the very moderate velocity.

Conditions Governing Capacity

Fundamentally, capacity is merely a matter of rate of firing which varies widely, whereas efficiencies with a given grade of fuel do not vary. Therefore, it follows that it is necessary only to put coal into a furnace and burn it to cause the rate of steaming to vary nearly in proportion to the rate of firing. This practical response of the rate of steaming to the rate of firing follows logically from the modern recognition that the limitation of the heat-transmitting ability of the heating surfaces is not approached. In large power stations much attention can well be given, in an effort to advance the present standard, to studies of the relation of rapid firing by one man in charge of a moderate grate area as against moderately slow firing of a large grate area represented in practice by having one man fire many furnaces.

Relation between Capacity and Efficiency

With low-grade fuel, forced draft nearly always secures increased efficiency, the efficiency increasing with the capacity within the limits of possible combustion. This appears to be strictly true when the heating surface alone is concerned. At very high combustion rates the furnace efficiency is apt to fall sufficiently to offset the rising boiler efficiency. This, however, under proper conditions of control occurs only near the upper limit of the usual attempts at forcing and as efficiency is not a ruling factor when forcing is necessary this is not of importance.

Higher steaming capacity usually reduces labor charges and inevitably reduces investment charges. It is true that maintenance charges increase, but these are confined to the furnace and, except in very special cases, while an offsetting factor, it is not a ruling one.

In considering air pressure and its relation to capacity and efficiency, one of the most serious limiting factors, in what otherwise appears a field where the higher the pressure the higher the overall station efficiency, is the blowing of coal from the grate after a critical pressure has been exceeded. Very fine coal is not only blown from the grate, which would be no serious matter if it came down again, but is blown over the bridge-wall, and in some cases is blown in considerable quantities through the heating passes and into the flues. The economic loss is not often serious, but in cities the trouble and expense may be considerable. Another limiting condition is the inability of the lower water tubes to stand the severe demands which may be made upon them. While it is probable that the capacity of the tubes to absorb practically unlimited quantities of heat may be counted upon, it is not true that this capacity is unaffected by the temperature at which the heat is delivered to the tubes of the lower row, and even of the second and third rows. This seems to be a limitation particularly concerned with high combustion rates secured with forced draft. When forced draft is used with very small-sized coal and a high average combustion rate is being maintained, it is practically impossible to avoid spots where for a time the combustion rate becomes extremely high. Blasts of gas of intense incandescence are delivered, which, while not notably increasing the total heat delivered from the entire grate to the heating surface, cause the endurance of isolated sections of the tubes in the lower rows to be severely tested. The approach to the safe limit of endurance of the metal is probably not a matter of approaching its limit of thermal transmitting ability but of the failure of the water-circulating arrangements to provide properly for heat absorption. It is not unlikely that in the trend of boiler modification which will come about to meet the approaching demand for higher steaming rates this matter of circulation will receive large attention. Another limiting feature against higher air pressure for the realization of higher capacity is the increased cost of the air supply at extremely high pressures. However, a moderate increase of 100 or 150 per cent. over the present nominal evaporating rate of 3.45 pounds of water per hour per square foot of heating surface from and at 212 degrees Fahrenheit must be brought about. The installation of very large turbines has forced this and in isolated instances much has already been accomplished.

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*From the report of the committee on "Power Generation," of the American Street and Interurban Railway Engineering Association.

Conditions Governing Efficiency

In railway power-plant work, forced draft must be adapted to varying steaming rates. The maintenance of steam pressures just slightly below the setting of the safety valves is the desideratum, and as the fluctuations in the steaming demand are large in relation to the steam-storage capacity of water-tube boilers, the adaptation of the forced-draft system to varying rates must be accurate and prompt. Steaming rates are varied, of course, only by a variation in the combustion rates. Efficiency, therefore, demands that the forced-draft system be adapted to permit variation of the combustion rate over wide ranges and this requires variation of the air pressure. Although a variation of air pressure will cause a variation in the rate of combustion of the coal already in the furnace, it will do no more. Therefore, the problem is, where hand labor is concerned (and this alone is considered as high air pressures are not commonly used with automatic stokers), to cause the firemen to vary their rate of firing to meet the varying rate of combustion. This is the crucial problem in the attainment of a high efficiency.

In many cases the quality of the labor is most unfavorable, lacking individuality in the relation of the man to the boiler. The labor, while reasonably well paid, is largely foreign labor and not infrequently is low in intelligence, thereby excluding the skill which in isolated-plant work is sometimes of sufficient merit to make firing approach the level of a skilled trade. In addition to the other handicaps rapid changes in the personnel are not rare.

A few years ago great interest was aroused by the development of automatic CO_2 recorders, but the hopes which this development aroused as to a solution of the problem of high efficiency have not been realized. More light is at present available on the cause of the disappointment than was available after the CO_2 recorders had been in use sufficiently to make the disappointment evident. This concerns the relation between gas velocities and the thermal transmitting ability of the heating surface. The efficiency of the combustion, as measured by the percentage of CO_2, is not the ruling criterion of overall efficiency nor of boiler efficiency separated from furnace efficiency, because a high percentage of CO_2 with unchanged combustion reduces the gas velocity across the heating surfaces, which is an offsetting factor. This reduction of velocity with the accompanying deleterious effect on the thermal transmission coefficient probably explains the practically invariable rise of flue temperature with increased CO_2.

The same strictures may be made against the somewhat extensively exploited balanced-draft systems. Unquestionably they at once gave very high CO_2 values. The claims of their sponsors in this respect were not exaggerated. Necessarily used only in conjunction with forced-draft systems, they involved the danger of flare-backs from the furnace doors, and aside from this they raised the furnace temperature so high that the tax on the firemen was severe.

It has been recognized that high CO_2 values, which are very readily obtained with forced-draft systems, may easily become too high, and these are a direct measure of the furnace temperature. Before the research work of the United States Geological Survey and others had made it evident that the highest CO_2 values did not correspond with the highest efficiencies, the best practice had settled down to ruling against more than 12 to 13 per cent. of CO_2.

The problem of handling forced-draft systems efficiently in railway power-plant service reverts to that of procuring firemen to fire well at the varying rates which are demanded by the varying load. The general tendency of poor firemen is to throw so much coal in one door without stopping, that a long interval elapses before the firing operation is repeated. A general corrective expedient for this, and one which is decidedly practicable, is the limitation of the number of shovelfuls permitted to be fired in any one door without stopping. This number may be six or eight, preferably six, and, although forming no limitation to the rate of firing, it at once goes a long way toward establishing the practice of light, thin fires as against heavy thick ones. Equally, if not more important than thin fires, is the correct adaptation of the firing rate to the draft, or in more specific terms, the ashpit or air-duct pressure. With a 1-inch air pressure in the duct, should the rate of firing with a given grade of coal and boiler equipment be 16 or 20 pounds per hour per square foot of grate surface, or some other rate? On the answer to this the resulting economy of the evaporation apparently hinges. The individual fireman of the order found in large power plants cannot be expected to answer this himself or, in fact, to know anything about it. Extensive experimental work appears to show that the error is sometimes on one side and sometimes on the other, but that with a given fuel and boiler equipment there is a definite relation between air pressure and rate of firing, which gives the best evaporative economy over wide ranges of steaming. Differential gages reading to 0.01 inch of water have been tried and are accurate and reliable but the difficulty is to devise a simple system for their use.

The problems which in their solution will probably mark the most significant advance in forced-draft practice appear to be the better proportioning of air ducts and sealed ashpits, particularly the intakes of the latter from the former, as now large losses of pressure are frequent; the modification of furnaces and boilers to allow the use of higher pressures and gas velocities which are necessary for the utilization of more intense evaporative rates; and the development of a simpler means for the control of the efficiency, which leaves at present more to be done in the way of controlling the firing than in controlling the air pressure or draft. It is hard to say whether the second of the problems, that of controlling the capacity, or the third, that of controlling the efficiency, is the more important.

Old Pete Blowoff sez he kin run his ingin on less cylinder ile than any ingineer in th' kentry. Pete never fills his lubricater 'cept when he aint full hisself, an' thet accounts fer it.

Th' humane sassiety hed Pete arrested fer runnin' his ingin when it was groanin'. Pete swore thet it wasn't sufferin' any but thet it hed th' heaves on account uv burnin' dusty coal.

Pete went up t' get license tother day. Th' inspecter feller ast him what he'd do ef he opened his furnis door an' seen th' bottom uv his biler red hot an' comin' down on th' grates.

Pete told him he'd drop th' shovel an' run like h—l. Th' inspecter sed "kerrect," an' give him a fust-class license.

I've noticed thet it aint always th' man thet kin shoot off th' most hot air 'bout algebray, triggernomerty, an' sich thet has th' best runnin' ingin, by a dum sight.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Size of Condensing Chamber

What is the rule for finding the size of the condensing chamber of jet condensers?

S. C. C.

The chamber should be large enough to provide for the thorough intermingling of the steam and water. The opinions of designers vary and the proportions range from a slight reduction in the size of the exhaust pipe to a chamber about four times the diameter of the pipe.

Lap for Given Cutoff

How can I find out how much lap to give a slide valve to cut off at one-half stroke?

L. G. C.

Multiply the valve travel in inches by 0.354.

Cushion by Lead

How much lead shall I give the valve of a 12x20-inch engine running 175 revolutions per minute to cushion the piston at the end of the stroke?

C. B. L.

None. Steam admitted to the cylinder in front of the advancing piston will not cushion it but will have the reverse effect. Compression alone will furnish the cushion needed, and may be increased without altering the lead by adding to the inside lap of the valve.

Flow of Air through Orifice

How can I determine how many cubic feet of air will be lost through a hole in a pipe or hose, the pressure and the size of the hole being known?

P. T. O.

The formula for the theoretical discharge is

$$V = A \times 37.8 \sqrt{P \times P'}$$

in which

V = Cubic feet of free air discharged per minute;
A = Area of orifice in square inches;
P = Absolute pressure in pounds per square inch in the pipe;
P' = Difference in pressure between the inside and the outside of the pipe.

The actual discharge for pressures in the neighborhood of 80 pounds per square inch will probably be 70 per cent. of the theoretical.

Submerged Hydraulic Ram

Can a hydraulic ram be made to operate if submerged?

J. W. L.

As the operation of the ram depends upon the velocity of the flow of the escaping water and as submergence will decrease the velocity of the flow, it may be seen that only a slight depth of submergence will render it inoperative.

Thermometer in Condenser

Of what use is a thermometer in a condenser?

T. I. C.

Knowing the temperature in the condenser may lead to the detection of the cause of insufficient vacuum. A high temperature indicates too little cooling water and a low temperature accompanied by poor vacuum indicates air leaks.

Compressor Discharge Valve Area

What is considered good practice in proportioning the relative areas of the valves and pistons of high-speed air compressors?

C. D. A.

Modern practice for piston speeds ranging from 500 to 700 feet per minute dictates that the effective area of the discharge-valve opening shall be 15 to 25 per cent. that of the piston and that the inlet-valve area shall be the same, if possible.

Wire Drawing Defined

What is the meaning of the term "wire drawing" when applied to steam, and how does it act?

W. D. D.

Steam is said to be wire drawn when it is passed from a region of one pressure to that of a lower through a restricted opening, as in the use of a throttling-governor valve, a pressure-reducing valve, a partially opened valve or cock, etc. It is said to be superheated because in passing to the region of lower pressure it parts with little or no heat and has the temperature of the higher pressure. Wire drawing causes no change in its attributes.

Grunting Feed Pump

I have a double-acting feed pump supplying hot water from exhaust tank and heater to boiler. This pump sets up a loud grunting noise at almost every stroke. What is the cause and how can I remedy it?

K. M.

The first thing to do is to determine whether the noise comes from the steam or water cylinders. If from the steam end, the lubrication is inefficient. If it comes from the water end, the packing is too tight and should be loosened.

Difference between Steam and Vacuum Gages

What is the difference between a steam gage and a vacuum gage?

D. S. G.

None, except the graduations on the dial and the purposes for which they are used. Each is used to indicate the difference in pressure existing between the outside and inside of the spring tube of the instrument. When used to indicate pressures below that of the atmosphere it is called a vacuum gage.

Safe Working Pressure

Please give me the rule for figuring the safe working pressure of a steel boiler 7/16 material, butt-strap joint, 72 inches diameter using the factor 5 for safety.

H. H. S.

The working pressure of boiler shells is calculated by the formula

$$\frac{T.\,S. \times t \times \%}{R \times F.\,S.} = \textit{Working pressure per square inch}$$

in which

$T.\,S.$ = Tensile strength of plate in pounds per square inch of cross-section;
t = Thickness of plate in inches;
$\%$ = Efficiency of joint;
R = Inside radius of outside course of shell;
$F.\,S.$ = Factor of safety.

As the tensile strength and the efficiency of the joint are lacking in the question they are assumed to be 60,000 pounds and 87 per cent. respectively.

Then,

$$\frac{60{,}000 \times 0.4375 \times 0.87}{36 \times 5} = 127$$

pounds per square inch.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for September, 1910, *was* 134,000.

October 4	33,000
October 11	33,000
October 18	33,000
October 25	33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

Safe Boiler Appliances

There are four essential adjuncts to a boiler that are necessary for its safe operation; namely, a blowoff valve, safety valve, water columns and gage glass and steam-pressure gage. All of these devices are shamefully abused in many small steam plants, and if the truth were known the same criticism might be applied to a good many large plants as well.

The safety valve is generally left to take care of itself, if it will but operate when the steam pressure reaches a certain predetermined point. The blowoff valve is often allowed to leak so long as it does not lower the water in the boiler enough to cause trouble. The water column and steam gage are the two safety factors which require watching, and yet they are often the ones most neglected.

On vertical boilers the water glass is frequently so poorly illuminated that even an expert could not tell to a certainty the exact location of the water level. Under such circumstances the boiler attendant depends upon the high- and low-water alarm and this device is frequently heard operating in many steam plants, which is a reflection upon the method of handling the plant.

A good engineer or chief fireman will keep the water glass on each boiler under his charge clean, both inside and outside, and a light properly placed so that the watertender or fireman can readily see the water level from various points of view.

The proper method of determining the water level in a boiler is to try the gage cock and not to depend upon the water glass. There are few firemen, however, who do not place almost implicit trust in the water glass; hence it should be kept clean and in proper working condition.

The pressure gage is by no means of secondary importance and yet few register correctly after a few years' use. Frequently the face is so dirty or the gage is so located that it is almost impossible to read the steam pressure. This is, of course, bad engineering and should never be countenanced by an engineer.

Placing a steam gage at a distance from the boiler is not good practice, because the long pipe is almost certain to become clogged with sediment. Connecting the steam gage to the metal front of a boiler, where heat from the boiler setting is transmitted to the gage, is a practice not to be commended.

A false steam gage is a dangerous thing and if it and the safety valve are both defective at the same time, a boiler to which they are attached is as dangerous as a bomb with a lighted fuse. Nevertheless, there are a large number of boilers operating under just such conditions. How about yours?

Of course, there is but one way to obtain safe operation of a steam boiler and that is to keep everything connected with its operation in the best of condition, first, last and all of the time.

The Operator and the Boiler Manufacturer

There are no two classes of people who should have more interests in common than the men who operate steam boilers and those who make them. The manufacturer is interested in making a good boiler to maintain his reputation, while the engineer is interested in having sound boilers to operate because his life depends on it. For twenty-two years the American Boiler Manufacturers' Association has stood for better material, better workmanship and methods and consequent greater safety in steam-boiler operation. One of the objects of the organization is the adoption of a standard set of specifications for boilers, so that a boiler built in one State will be accepted as safe in any other State, which is decidedly not the case at present. If a man wants to buy a boiler for a steam launch he can buy of any boilermaker and know that the boiler will be as good as high quality, material, workmanship and inspection can make it, because the boilermaker must take his plans to the proper Government authorities and prove that the sheets are strong enough, that the stays are adequate and properly spaced and that the general design is in keeping with the service the boiler will be called upon to perform. After the plans have been approved, the construction will proceed under competent inspection, and when finished the job will be a credit to the maker.

If the same man wants to buy a boiler for his office building he has only to specify the capacity required and then out of those offered take the cheapest, regardless of quality of steel, workmanship or suitability for the conditions. This is something which should be corrected and which the association is working for. While the subject of materials and workmanship occupies the attention of the makers, discussions among operating engineers are almost exclusively on methods of handling to avoid accident. The two questions go hand in hand and there is unlimited opportunity for coöperation if the work is carried on in the right spirit.

The National Association of Stationary Engineers should have a representative at every meeting of the Boiler Manufacturers' Association and should lend its influence wherever possible to bettering the quality of material entering into steam boilers, equalizing the boiler regulations throughout the country and raising the standard both of construction and operation.

Gas Power and the Engineer

In a well equipped steam plant, operating condensing and containing all the refinements which tend toward economy, the ultimate efficiency depends upon the combination of a number of factors, including the boilers, engines, steam piping, condensers and other auxiliaries, as well as the coal and the care exercised in firing. With all these operating at their best, the highest thermal efficiency thus far attainable has been only about fifteen per cent.; and this has been possible only where a steam turbine or a compound Corliss engine was employed. Of the eighty-five per cent. of the energy in the coal which is unutilized, some passes up the stack both as sensible heat and as heat in unburned gases, and some is lost by radiation, but perhaps the greatest loss may be attributed to the inability of the steam engine to utilize all of the energy contained in the steam. Expressing these facts in a more concrete form, it may be said that for every ton of coal costing three dollars, only forty-five cents' worth of energy is converted into useful work.

In view of this relatively low efficiency of the steam plant it is somewhat surprising that the gas engine and gas producer have not come into more general use for medium-sized plants in this country. For with a gas engine and producer, a thermal efficiency about double that of the steam plant is attainable; that is, for a pound of a given coal twice as much useful work can be obtained. Moreover, low-grade coals may be used to advantage in a producer. The fact that the gas engine in relatively large units is a recent development, and the consequent lack of knowledge regarding it on the part of buyers and operators, may account largely for the slowness with which the use of medium-size units has increased. However, the number of such units is increasing, even though not by leaps and bounds, and it behooves the steam engineer to become familiar with their operation; he cannot afford to sit idly by until they are forced upon him, but should educate himself as to all their salient features and be prepared to take advantage of an opportunity (or to meet an exigency, as the case may be), which may develop any day.

Lest We Forget

In these days of many magazines, books, papers and reviews, the average man is sorely put to it to know what to read and after he has read to know how to remember. This is as true of technical as it is of general literature; to the engineer, the manufacturer, the business manager, there comes the troublesome question, "How much of this mass of material must I read or skim or index, how can I tell what I may want and how can I ever find it if I do want it?"

The reading tables of our best technical libraries and clubs contain from fifteen to thirty periodicals, all bearing more or less directly on technical and commercial life and containing what is newest and best of information concerning the business of the world.

The engineering books on the shelves are frequently out of date when they leave the hands of the printer and it is to periodical literature that the engineer must look for the latest developments in any branch of technology or science.

It is out of the question for a busy man of affairs to read even a small part of the articles thus displayed, and yet, such is the caprice of fate, any one he neglects may later be the one most necessary to him. Many plans for solving the problem have been proposed and one rarely scans a technical journal that he does not find some new recipe for canning information. The trouble with most of them is the gradual accumulation of comparatively useless material, bushels of chaff with but a few grains of wheat. It is something like the old New England mania for saving all scraps in the hope that they would sometime be useful. Buckles, bolts, hinges, hasps, straps, sheet metal, wire, accumulated in the shed or the attic until those receptacles overflowed. "Many were called but few chosen"—and so it is apt to be with journalistic scrap.

Some of the methods proposed, arranged in the order of their completeness, may be mentioned as follows:

First. Binding all journals taken and relying on the index in each volume.

Second. Cutting out all desirable articles or items and filing them in cases with subject indexes.

Third. Relying upon the nearest library for files and purchasing standard engineering and technical indexes.

Fourth. Keeping a private, special card index and going to the library for material.

The first plan is too costly and too cumbersome both as to the initial outlay and as to its subsequent use. It is too much of an undertaking to hunt the back numbers of journals for information, no matter how great the need.

The second is better and, if carried out intelligently, may serve the purpose fairly well. The rapid accumulation of material often causes the system to break down the second or third year.

Number three is better yet, if the files of periodicals are readily accessible. It has this one serious drawback—an index, however good, which is prepared by others may not help you to find what you seek.

The last method does away with this difficulty. Every engineer should make his own index in his own way and he will then be able to find what he wants in the same way.

Perhaps it is superflous to make suggestions when so much has been written on this subject, but it would seem that a combination of some of the methods indicated previously might lead to good results.

The firm or the individual should take two or three journals which deal most directly with the business in question. Clippings from these can be filed and indexed but the selection must be carefully made, and by the engineer, not by the office boy. If a periodical is particularly valuable to the business, two copies can be taken, one to bind and one to clip.

For other journals of less immediate interest, the technical indexes can be used. In addition to these, a special card index should be made covering all the important articles, items or paragraphs which bear immediately on the work of the firm or of the individual, so designated as to be readily found.

When in doubt *do not* clip or index, for a scarcity of material is better than an overloaded collection.. Once a year go over the clippings and discard any of doubtful value; in the same way remove from the file, cards which have outlived their usefulness, for only in such ways can the material be kept within such bounds as to be of practical use.

In general, technical information of this character ceases to be of much value after five years or more have elapsed and its presence is a hindrance rather than a help.

Like the cowboy and his revolver, you do not often want an index but when you do, "you want it powerful bad, and you want it quick."

Try to have your own clipping bureau and your own index and you will not regret the time and trouble involved.

When a gas engine commences to miss it is usually the case that the igniter plugs are fouled. Tangye has a ratchet wheel attached to the head of the plug on the outside of the cylinder, and a pawl which rotates it one tooth with each revolution of the lay shaft.

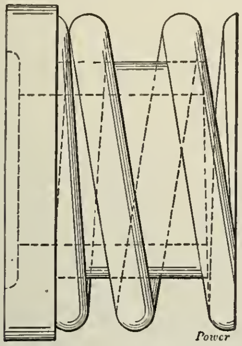
New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

The Weisz Packing Adjuster

This device consists of a steel bushing on which a steel spring is fitted, as shown in the accompanying illustration.

The bushing and spring are placed in the stuffing box and are followed by sufficient rings of packing to fill the stuffing box, as under ordinary conditions. The stuffing-box gland is then screwed up in the ordinary way, which puts a tension on the spring. As the packing wears, the spring keeps it snug against the rod.



Bushing and Spring

This packing adjuster is made by F. G. Weisz, 117 Weldon street, Brooklyn, N. Y.

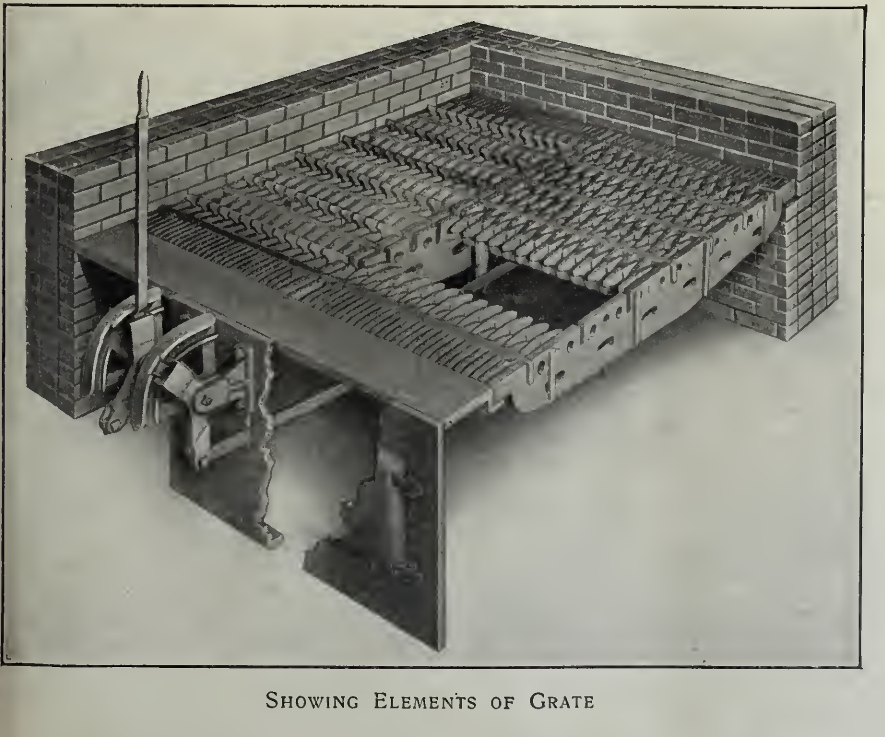
The Twentieth Century Rocking Grate

The accompanying illustration shows the details of this grate, which is made by the Water-Arch Furnace Company, 1252 First National Bank building, Chicago, Ill.

The grate consists of alternate sets of fixed bars between which is a rocking element, the fingers of which work between fixed projections on the stationary parts. The movement of the rocking element produces a powerful slicing and clinker crushing action, thereby forcing open the entire air space of the grate, removing the bottom layer of ash and clinker to the ashpit and allowing a free passage of air to the burning fuel.

The parts of the grate are designed for strength, resistance to wear and have ample cooling surface. The rocking bars are held in their bearings without the use of keys, catches, or other special devices, and by means of a self-locking rocker head it is impossible to leave the moving fingers projecting into the fire to burn off.

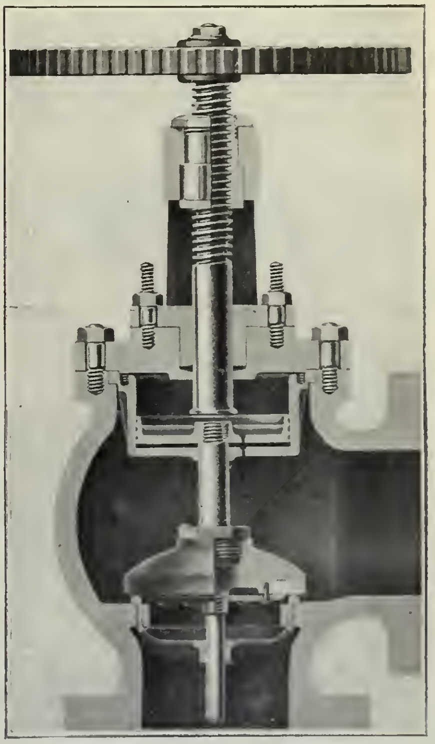
This rocker head compels the operator to center the rocker bars in place after shaking before the handle can be withdrawn, thus preventing any mistake by new or inexperienced help. All parts are interchangeable, built to withstand the hardest conditions of forced firing with clinkering coals, and there are no right- and left-hand parts in the construction, thus simplifying repairs. The air space represents 53 per cent. of the total of the grate and low-grade coal has been burned at a rate of 40 pounds per hour per square foot of grate surface, it is claimed.



Showing Elements of Grate

Automatic Equalizing Stop and Check Valve

This valve herewith illustrated is designed to automatically shut off the flow of steam from the header to a boiler in case a tube should burst or other internal rupture occur, thereby suddenly reducing the pressure in that boiler. It also equalizes the pressure between the different boilers in a battery, preventing one boiler from working at a lower pressure than another. As the valve can only be opened by the pressure in the boiler to which it is attached, it is impossible to accidently turn steam into a boiler which is being cleaned.



Sectional View of Equalizing Stop and Check Valve

The valve is cushioned, to prevent chattering, by an internal bronze dashpot. This eliminates the danger of sticking through corrosion of the dashpot or piston. The valve is designed to be installed in a vertical position and is furnished in extra-heavy pattern only, in sizes of 4 to 12 inches.

The valve is made by Jenkins Brothers, 71 John street, New York City.

A Quick Acting Wrench

The wrench shown in Figs. 1, 2, 3 and 4 will turn a square, hexagon, flat or round-shaped piece which has a slight resistance in the direction of rotation desired, without removing it from the piece. The action is positive and will not slip, it is claimed.

The shank *A* is slotted, the stock *C* is pivoted to it at *E* and is free to move in the slot except when the locking bolt *D* is pushed under the yoke *H,* in which case the wrench can be used in every way the same as the regular monkey wrench with the added advantage of having a slight friction due to the spring *I* in the adjusting screw *F* which tends to keep it set to the size it is adjusted for.

When used as an automatic wrench the locking bolt is pushed back into the stock, as shown in Fig. 2, and the wrench adjusted to size on the piece to be turned.

The action of the wrench when it is pushed backward for a new grip is as follows: First for the shank *C* to lift away from the adjusting screw *F,* then for the sliding jaw to open over the corners of the nut, then to close in again on the next flats of the nut due to the tension of the spring *I* in the adjusting screw *F*.

When the wrench is pulled forward against the shank it engages the adjusting screw *F* and is ready for another pull.

The curved surface of the spring rod *G*, which rolls on the bottom of the groove in the shank *C*, is shaped to give a slight pressure tending to close the wrench.

A detachable pipe jaw fits loosely over the jaw of the wrench and is held from falling off by the hook-shaped arms, as shown. The object of this jaw is to slip freely on the face of the wrench jaw when in use and wedge itself into the pipe or other object being turned, this forming a three-point bearing on the object and giving a grip that is sure to hold but which will release itself easily and catches readily for a new grip.

The pipe jaw is very neatly secured in the hollow handle by the spring, as shown, when not in use.

The wrench was designed and patent applied for by George W. Jessup, Jr., Newton, Mass.

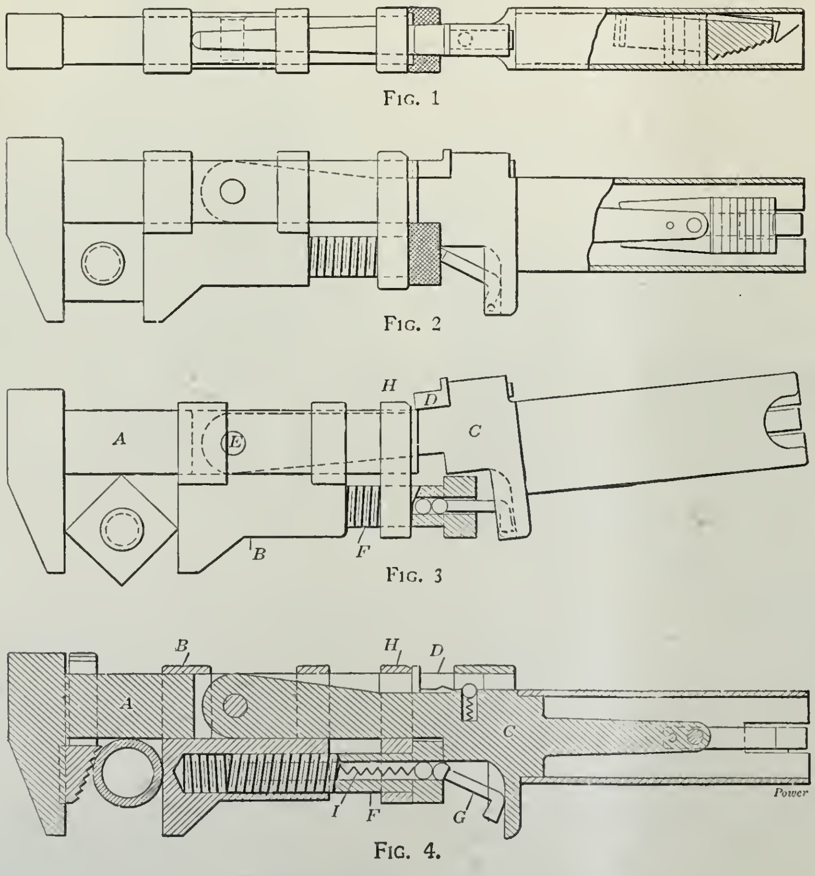


Fig. 1

Fig. 2

Fig. 3

Fig. 4.

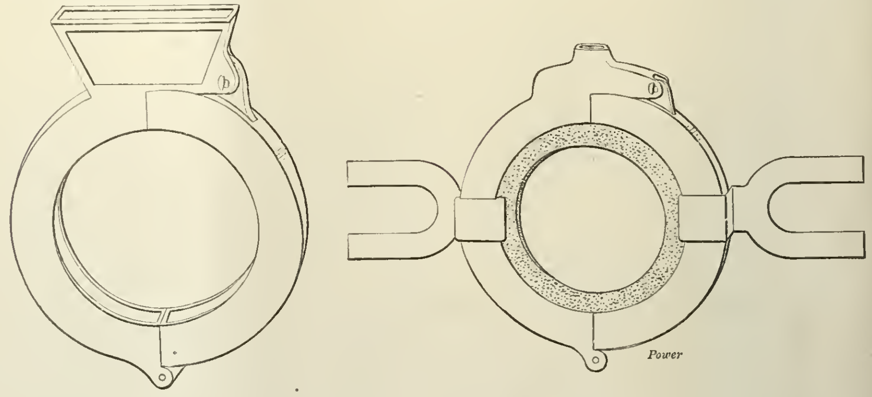
Details of the Wrench

Weiss Rod Oiler and Wiper

The illustration produced herewith shows two types of the Weiss oiler and wiper. One is designed for stationary engines and pumps, air compressors and condensers of the horizontal type, and the second type is designed for vertical or marine engines and vertical pumps. The latter type is bored and tapped to be connected to an oil cup at any convenient place.

The oil-soaked felt which lines the oiler comes in contact with the rod at all points and wipes it clean at each stroke, and at the same time lubricates the packing, which prevents friction and prevents the rod from scoring.

These cages, or oilers, are made of aluminum in all sizes from 1 inch up by the Weiss Brothers-Manufacturing Company, Detroit, Mich.



Two Types of Rod Oiler and Wiper

While engaged in building a mountain road from Banning to Idyllwild, Cal., large pieces of soft coal were discovered in a heretofore unexplored portion of a cañon of the San Jacinto range. An investigation will be made at once in an endeavor to locate the main deposit; such a discovery would mean a large increase of interests in southern California. The coal is of good grade and easy burning.

Keep the brickwork tight so that cold air does not enter, as it will cause fuel losses. Fire lightly and frequently and endeavor by all means to save fuel for your employer. Endeavor to carry a uniform water level when the boiler is in service. Keep the feed pumps, injectors and all other appliances in good working order and as neat as possible.

Convention of the American Boiler Manufacturers Ass'n

The twenty-second annual meeting of this association took place at the Auditorium hotel, Chicago, October 10-13, and in attendance and enthusiasm it is safe to say that no other gathering in the history of the organization approached the present one in any of its details. A steady growth has been recorded among manufacturers interested in the various types of boilers, so that while in the past a great proportion of the activity has been along the lines of marine work, the stationary field is now being represented with a stronger membership than ever before. The convention went on record as favoring specifications calling for not more than 0.03 per cent. of sulphur and 0.04 per cent. of phosphorus in boiler plate, which is in accordance with the original resolution passed on this subect some two years ago. A great deal of discussion was indulged in regarding he quality of boiler plate, those taking part being prominent representatives of he Government, State boiler-inspection ureaus and manufacturers. As a result of the deliberations a standardization ommittee was appointed to report at the ext meeting and it was decided to work oward the adoption in this country of a uniform set of specifications for boilers or stationary purposes somewhat similar to those required in marine work.

All of the old officers were reëlected, as follows: E. D. Meier, New York, president; J. D. Farasey, Cleveland, secretary; Joseph F. Wangler, St. Louis, treasurer; T. M. Rees, Pittsburg, first vice-president; J. Don Smith, Charleston, second vice-president; W. A. Brunner, Phillipsburg, N. J., third vice-president; H. D. MacKinnon, Bay City, Mich., fourth vice-president, and M. A. Ryan, Duluth, fifth vice-president.

Members of American Boiler Manufacturers' Association on Trip of Inspection at Gary Steel Mills

While the entertainment features were of the most liberal character, consisting of theater parties, automobile rides, banquets, etc., which are usual at such meetings, probably the most interesting affair was a trip to the Gary steel mills, by special train, as guests of the United States Steel Corporation. The visitors were even provided with a special train within the grounds and taken to all parts of the large works which were thoroughly inspected before leaving. On the return trip a buffet lunch was served in the Pullman cars.

The association was also the guest of Joseph T. Ryerson & Sons, at a luncheon at Lawndale, Thursday afternoon, and in the evening occurred the farewell banquet at the Auditorium hotel.

Boston was selected as the place of the next annual meeting.

Boiler Explosion

At Union City, Ind., on October 11, a boiler belonging to Witham & Bowen, lumber dealers, exploded. One man was severely injured and considerable property damage resulted. Further details will be given in an early issue.

Steam Meter Presented to University of Illinois

The General Electric Company, of Schenectady, N. Y., has presented the University of Illinois with a recording steam meter. The gift was transmitted on behalf of the company by its sales manager, F. G. Vaughen, to Prof. Ernst J. Berg, in charge of the department of electrical engineering. Earlier in the year the university was presented with a 100-kilowatt Curtis steam turbo-generator which now constitutes a part of the equipment of the electrical laboratory.

Correction

In H. J. Westover's article, "Loss in Coal Due to Excess Air," published in the October 11 issue, page 1808, third column, fourth paragraph, second line, the word *closed* should have been used instead of *opened*, making the sentence read: This is the gas formed when the damper is suddenly closed at a time when a very bright fire is burning.

Head Severed by Compressor Flywheel

At Aspen, Colo., on October 7, while tightening a loose bearing nut on a compressor in the Smuggler Mining Company plant, the foreman of the compressor department was instantly killed by being crushed in the 5-foot flywheel of the machine.

The foreman noticed the loose nut while inspecting the machine and had the compressor stopped. To tighten the nut properly it was necessary for him to put his head and shoulders through the spokes of the wheel, which suddenly turned backward, crushing him against the machine and severing his head.

The body was jammed in the wheel in a manner that necessitated the cutting of several of the neck cords before the body and head could be released.

The Snee Wave Motor

The Snee wave motor has fulfilled our estimate of its abilities and value. Above the big installation of concrete and iron upon the Million Dollar pier, at Atlantic City, there stands a windmill tower with a small windmill, differing from the ordinary form only in the curvature of its blades and bearing the sign—American Power and Manufacturing Company, 410 Union Bank building, Pittsburg, Penn. We learned that Mr. Snee has turned his attention from harnessing the sea to catching the winds, and that stockholders in the Snee Motor Company have been given a proportionate amount of stock in the windmill concern.

George M. Douglass, chief inspector of the Casualty Company of America since its formation, and head of the boiler, flywheel and inspection departments since May, 1909, has severed his connection with the company. Mr. Douglass will take a much needed rest from his recent arduous work in connection with the three departments before again engaging in business.

SOCIETY NOTES

The next meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers will be held at Springfield, O., on November 18 and 19.

BOOK RECEIVED

World Corporation. By King C. Gillette. The New England News Company, Boston, Mass. Cardboard; 240 pages, 6x9 inches. Price, $1.

Applied Thermodynamics. By H. W. Spangler. John Jos. McVey, Philadelphia, Penn. Cloth; 160 pages, 6⅛x9¼ inches; 76 illustrations; indexed.

Machine Drawing. By Gardner C. Anthony. D. C. Heath & Co., Boston, Mass. Cloth; 104 pages, 7½x6 inches; 127 illustrations. Price, $1.50.

Standard Handbook for Electrical Engineers. By a staff of specialists. McGraw-Hill Book Company, New York. Morocco leather; 1497 pages, 4¼x7¾ inches; illustrated; indexed. Price, $4.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

PUMPING ENGINE. Burt O. Gage, Warren, Mass., assignor to Warren Steam Pump Company, a Corporation of Massachusetts. 972,357.

EXPLOSION MOTOR. Charles Francis Jenkins, Washington, D. C. 972,379.

EXPLOSION ENGINE. Charles Francis Jenkins, Washington, D. C. 972,380.

INTERNAL COMBUSTION ENGINE. Geo. F. Swain, Chicago, Ill. 972,409.

TURBINE. George Westinghouse, Pittsburg, Penn. 972,421.

MOTOR WHEEL. James N. Johnson and John H. Johnson, Flomaton, Ala. 972,453.

CONTINUOUS COMBUSTION HEAT ENGINE. Walter F. Brown, Worcester, Mass. 972,504.

GAS ENGINE. George P. Law, Lansing, Mich. 972,547.

ROTARY ENGINE. Vernon L. Capwell, Dorranceton, Penn., assignor to the Rotary Engine and Valve Developing Company, Dorranceton, Penn. 972,598.

GAS TURBINE ENGINE. William A. Reed, Hollywood, Cal. 972,642.

TURBINE. Max Glans, Vienna, Austria-Hungary. 972,762.

ELASTIC FLUID TURBINE. William J. A. London, Hartford, Conn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 972,788.

BOILERS, FURNACES AND GAS PRODUCERS

CRUDE-OIL BURNER. William D. Patton, Dallas, Tex. 972,401.

BOILER. Albert G. Sherman, Buffalo, N. Y. 972,580.

ROCKING-BAR FURNACE GRATE. Edward T. McHugh, Holyoke, Mass. 972,707.

FEED-WATER HEATER AND SMOKE CONSUMER. George C. Miller, Fitchburg, Mass. 972,772.

POWER PLANT AUXILIARIES AND APPLIANCES

DETACHABLE BOILER FLUE. John M. Crozier, Minneapolis, Minn. 972,164.

AUTOMATIC STEAM CUTOFF VALVE. George W. Collin, Bridgeport, Conn. 972,338.

BLOWOFF VALVE. Frank P. Hamilton, Cleveland, Ohio. 972,370.

BOILER-FLUE CLEANER. Robert O. Hodge, Buffalo, N. Y. 972,374.

VALVE SPRING RETAINER. Oscar Blomberg, Muskegon, Mich. 972,434.

STARTING GEAR OF INTERNAL COMBUSTION MOTORS. Paul Daimler and Alfred Vischer, Unterturkheim-Stuttgart, Germany, assignors to Daimler Motorengesellschaft, Unterturkheim - Stuttgart, Germany. 972,512.

VALVE. Wilbur B. Burke, Cleveland, Ohio. 972,677.

LUBRICATING DEVICE. Luther K. Smith, Moberly, Mo. 972,659.

LUBRICATOR. Lewis Jackson Maloy, Copperhill, Tenn. 972,701.

FLUID PRESSURE VALVE REGULATOR. Thomas M. Wilkins, East Randolph, N. Y., assignor of five-sixteenths to W. E. Zierden, Johasonburg, Penn., one-fourth to John G. Whitmore, Ridgway, Penn., and seven-sixteenths to Ziermore Regulator Company, Johnsonburg, Penn., a Corporation of Pennsylvania. 972,744.

CHAIN GRATE. Edward C. Clark, Nutley, N. J., assignor to the Oscar Barnett Foundry Company, a Corporation of New Jersey. 972,751.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRICAL SYSTEM OF DISTRIBUTION. Albert S. Hubbard, Belleville, N. J., assignor to Gould Storage Battery Company, a Corporation of New York. 972,536.

ELECTRIC SIGN FLASHER. Roger D. DeWolf, Pittsburg, Penn. 972,516.

ELECTRICAL INDICATOR. Edward E. Craig, Kansas City, Mo. 972,682.

SYSTEM OF ELECTRICAL DISTRIBUTION. Justus B. Entz, Philadelphia, Penn., assignor to the Electric Storage Battery Company, Philadelphia, Penn., a Corporation of New Jersey. 972,782.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

Vol. 32 NEW YORK, NOVEMBER 1, 1910 No. 44

As a general proposition all of the coal is not taken from the mines. Often that coal which is allowed to remain is left in such condition that to get it out at some subsequent time will entail much expense which by a little foresight in the first place could be avoided.

Some coal is put into beehive ovens and converted into coke. The heat, the combustible gas and the byproducts, tar and ammonia, which result from the conversion of coal into coke, are utterly wasted.

At many large lumber-manufacturing mills, the disposal of sawdust and waste wood is a source of actual expense. At some mills the waste products are burned in immense incinerators. As much of the waste wood as is needed is used in the boiler house to generate steam. The rest goes into the destructor to be burned, in spite of the fact that alcohol could be distilled from the wood and the spent wood utilized to furnish heat for the still.

This is not conservation.

The pity is that manufacturers find it profitable to employ such wasteful methods, for the time is in sight when the scarcity of at least some raw materials will amount almost to a positive hardship.

That the rapid depletion of the natural resources of the nation is a serious question is made evident by the fact that the United States Government has appointed a commission of able men who are devoting earnest efforts to the problem of how best to conserve our stores of raw materials. In 1908 the President convoked the governors of all the States to consider ways and means of furthering conservation. The Second National Conservation Congress has but recently adjourned.

These things are as they should be, and commendable.

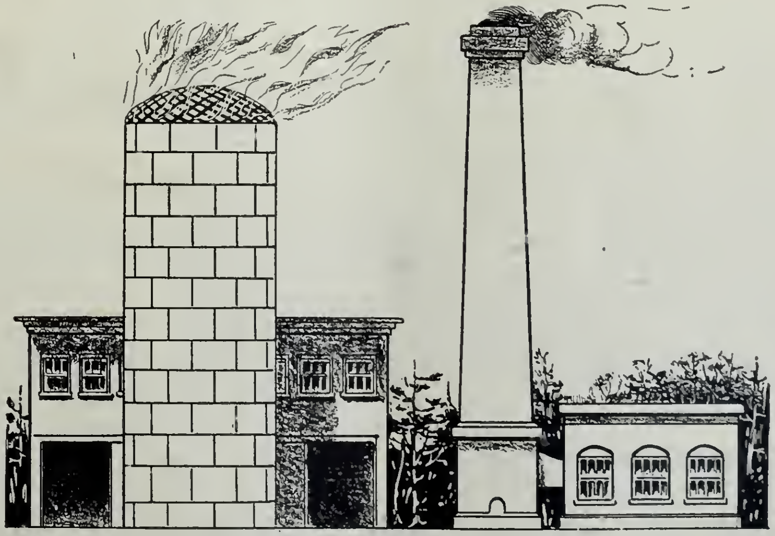
However, true conservation cannot be accomplished by commissions and congresses. It cannot be furthered materially by the withdrawal of coal and mineral lands or by the control of water-power sites by the Government. The people of the Nation must be the conservators in the last analysis.

Genuine conservation can be practised by everyone, for, reduced to the simplest terms, conservation means *best use*. In the instances cited in the opening paragraphs, best use means longer endurance of the supplies of raw materials coupled with greater ultimate profits from their utilization.

With the individual, best use of the materials which are employed and best use of his abilities make a man efficient.

In steam engineering, conservation of the best kind can be practised right at the throttle wheel and the fire door. A properly adjusted and lubricated engine requires less steam than one of the other kind. Less steam means less coal, other things being equal. And with intelligent firing, the required amount of coal is still further reduced. Then there are other factors too—many of them—all of which boil down to man's use of his intelligence.

Therefore you are urged to be a conservator to boost your own as well as the common cause.



Power Plant of McCormick Building

By Osborn Monnett

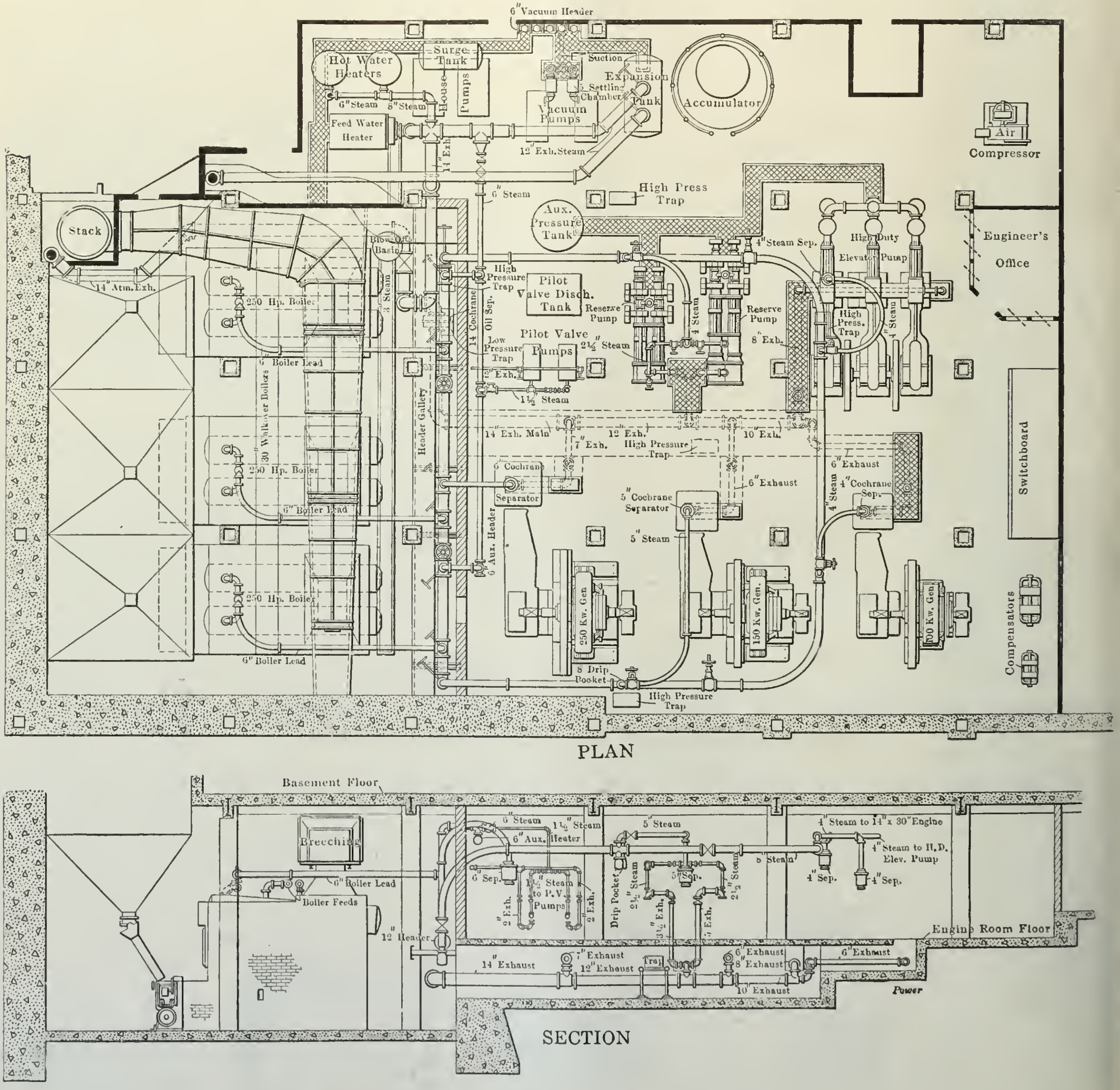
> *This power plant is so designed that continuity of operation is a practical certainty. Ample room, automatic apparatus and instruments and carefully worked out details, such as the piping color scheme, make for favorable operating conditions.*

A visit to the subbasement of the new 20-story McCormick building, on the corner of Michigan avenue and Van Buren street, Chicago, discloses a steam plant of the most modern character. It is evident that no expense has been spared in equipping the plant in the most substantial way and providing everything necessary for the service requirements in a building of this kind. The designers were able to use plenty of room for the power plant, resulting in an absence of the usual crowded conditions found in so many plants. The engine-room floor level is about 35 feet below the street. The decorations are of a plain character. The machines are finished to match the general decoration scheme and neatly striped. The foundations are faced with white enamel brick which, in connection with the "Granito" floor, consisting of a mixture of cement and marble chips, ground and smooth, polished, gives a finished appearance to the room.

In the layout of the plant, as seen in Fig. 1, the electrical generating units have been ranged on one side of the room, while on the other are located the pumping equipment and various auxiliary apparatus.

BOILER ROOM

The boiler room, which is at a level 10 feet below the engine room, contains, as shown in Fig. 2, three 250-horsepower Heine, double-drum water-tube boilers,



PLAN

SECTION

FIG. 1. PLAN AND SECTION OF POWER PLANT IN McCORMICK BUILDING

fitted with McKenzie chain-grate stokers having 60 square feet of grate surface each. The stokers are equipped with individual motor drives and variable speed control. The boilers are fitted with a standard equipment of Bayer soot blowers. A Spencer damper regulator in the main uptake controls the draft according to the steam pressure. It is interesting to note that the damper regulator is to be electrically connected with the stoker drives, in order to vary the rate of feeding the fuel in accordance with the demand for steam. This should result in ideal furnace conditions and act as an additional safeguard against smoke with variable loads. To keep check on the matter of combustion, an Eddy smoke recorder has been installed, to give a continuous record of stack conditions at all times.

Coal is delivered from wagons in the alley, dumped into the bunkers, which have a capacity of 280 tons, whence it is spouted to the stokers in the ordinary way. The ashes are removed from beneath the stokers by hand, wheeled to Illinois Tunnel Company's cars through a connection which taps into the boiler room and removed underground through the tunnel. For boiler feeding there are two 10 and 6½ by 10-inch vertical Blake pumps, installed on the boiler-room wall, feeding from a 1000-horsepower Cochrane open heater under a head of about 10 feet. The feed lines extend to manifolds between the boilers, from which branches lead to the boilers. The feed water is introduced in the front of each head. Gould water columns and feed-water control maintain the water level. There are bypasses on the manifolds so that the feed water may be controlled by hand when necessary.

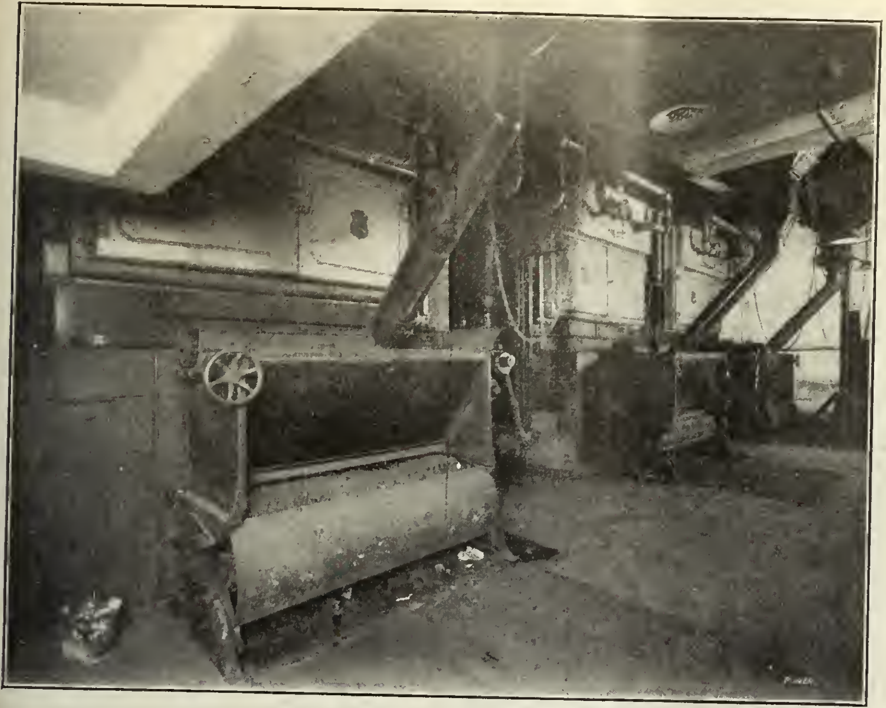


Fig. 2. Boiler Room of McCormick Building

Piping

An elaborate system of steam piping has been worked out to insure against breakdown. By referring to Fig. 1, it will be evident that the high-pressure steam lines have been laid out on the loop system. The main steam header, 12 inches in diameter, lies behind the boilers about 8 feet above the boiler-room floor. An iron platform is conveniently located, making the header easily accessible for repairs. Into the header drop 6-inch connections from the boilers, and out of each end section rises an 8-inch line to the loop in the engine room. Two gate valves divide the main header into three sections, the middle one having a connection direct to the largest generating unit. An auxiliary header extends along the engine-room wall, with pump connections and also a live-steam bypass to the heating system. This auxiliary header taps into each end of the main header and can be supplied from either end at will. By means of the piping arrangement, as shown, the entire engine-room loop may be supplied by either end of the main header or the middle section can be utilized alone for electric power and pumping service. Any section of the header may therefore be cut out for repairs or isolated for testing the performance of any one boiler or engine unit. Similarly, in the engine room, the loop is well provided with gate valves, enabling any section to be taken out of service without interrupting the operation of the plant. Branches to the several engines come out of the top of the loop and are provided with long-sweep bends. The piping is extra-heavy weight with screwed flanges, peened on the inside, the flanges being made up with Tauril packing. Crane-tilt traps take care of the condensation. Cochrane separators are used throughout at the throttle valves and also on the rims of piping.

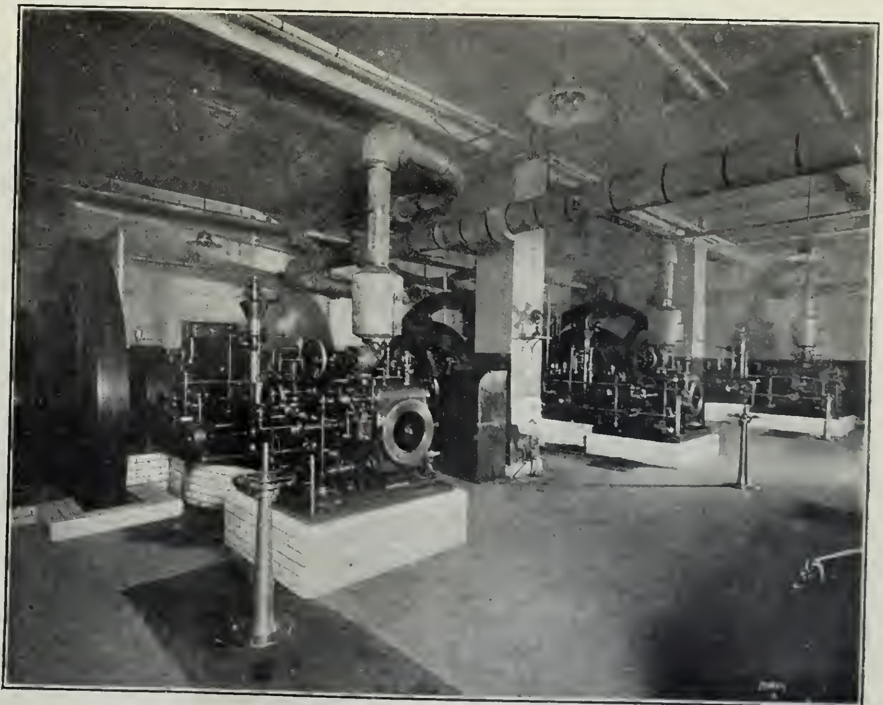


Fig. 3. Generating Units in Engine Room of McCormick Building

All valves in the plant are Lunkenheimer, the large gate valves on the steam-header system being provided with bypasses. All valve stems are packed with Root's high-speed metallic packing. This packing is also used on every steam-piston rod in the plant.

The exhaust piping is unusually accessible; the several exhaust lines from the engines lead into a tunnel under the main aisle of the engine room, where any pipe may be worked on without interfering with any other. The tunnel leads to the boiler room, where, after passing through a 14-inch Cochrane oil separator, the exhaust connection is made to the heater, the heating system and the atmosphere, in the usual manner.

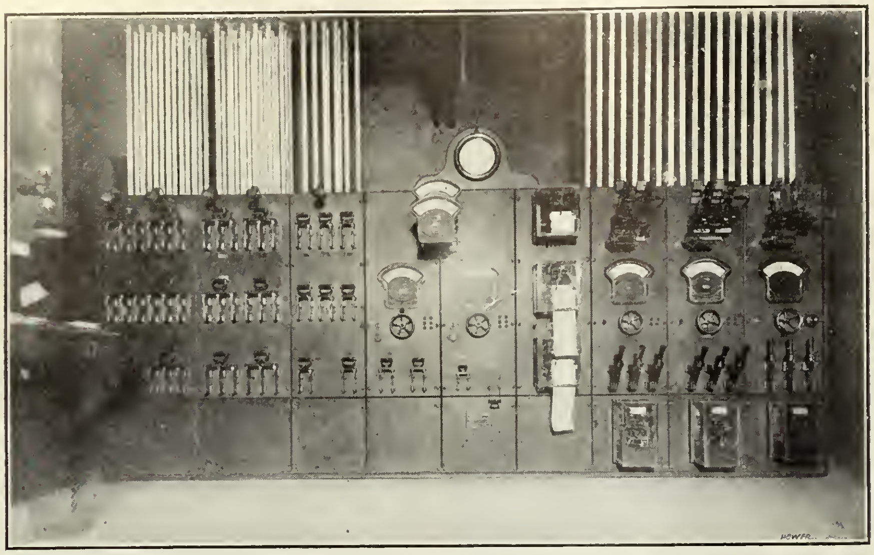


FIG. 4. THE MAIN SWITCHBOARD

GENERATING UNITS

Fig. 3 gives a general view of the electrical side of the engine room. Here are installed three Filer & Stowell Corliss engines, direct connected to Crocker-Wheeler generators. The units have a capacity of 100, 150 and 250 kilowatts, respectively, each running at 110 revolutions per minute. The smallest engine has a 14x30-inch cylinder, the next in size a 16x36 and the largest, a cylinder 18x36 inches in size. Hills-McCanna force-feed oil pumps are used for valve lubrication on these engines as well as on all steam pumps, the main-valve oil supply being kept in a Bowser oil-storage and measuring tank. The engine oiling system, while of the standard gravity-flow type, has some features worthy of note. Waste oil is caught in a settling tank located in the tunnel under the engine-room floor, from where it is drawn to a White-Star filter. The purified oil is pumped to an elevated pressure tank in the engine room, by a small air-operated Knowles pump.

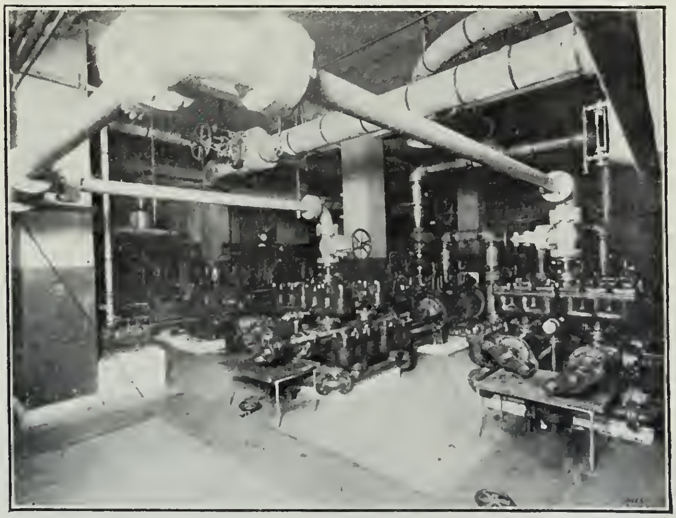


FIG. 5. THE ELEVATOR PUMPS

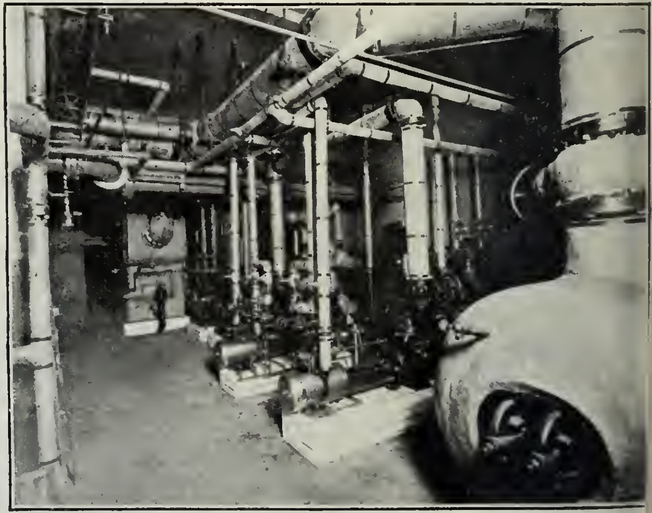


FIG. 6. HOUSE AND VACUUM PUMPS, FEED-WATER HEATER AND EXPANSION TANK

It was found that in hot weather the oil supply was excessively warm, sometimes reaching a temperature of 140 degrees, causing the bearings to run hot. The expedient was adopted, therefore, of passing the oil supply through a double-pipe cooler consisting of a piece of 1-inch pipe inside a 2½-inch pipe through which a small amount of city water is circulated.

From the cooler on, the oil-supply pipes are lagged with regular pipe covering to keep the heat of the engine room out, and the oil now reaches the bearings in hot weather at a temperature not greater than 90 degrees. Nugent telescopic oiling devices are supplied on eccentrics, wrist-pins and wherever necessary.

Current is delivered by the generators at 225 volts and distributed throughout the building by a three-wire system, using balancer sets, the lighting being done at 110 volts while the power circuits are of 220 volts pressure. Each generator panel carries a wattmeter. On the meter panel are carried a Westinghouse graphic recording voltmeter and two ammeters of similar type, one for lighting and one for power. The switchboard itself, a view of which is shown in Fig. 4, is constructed of black slate. The back of the board is inclosed in substantial wire netting, making it impossible to come in contact with the busbars. Every circuit coming to or going from the board is in iron-armored conduits and proof against mechanical injury.

PUMPS

All of the eleven elevators in the building are hydraulic; nine are for passenger service and two for freight. They are of the Otis inverted-plunger type, operating under a pressure of 800 pounds per square inch and having a weighted accumulator and ram floating on the system. For carrying the regular elevator load there is one Laidlaw-Dunn-Gordon high-duty three-cylinder compound pumping engine provided with 16 and 30 by 3 by 24-inch cylinders and 5⅝-inch wate plungers, while held in reserve is a 1 and 20 by 18 by 5½-inch tandem-com pound pump and one simple 16 and 1 by 5½-inch pump, both of the pot-valv type and of the same make as the high-duty unit. The remainder of the elevator equipment is of standard Otis design for this type of machinery. Two 6 and 4 by 6-inch Laidlaw pilot pumps maintain a pressure of 200 pounds on the elevator pilot-valve system, these latter operating at this pressure rather than at the full 800-pound pressure of the main hydraulic line.

Fig. 6 illustrates the remainder of the equipment in the engine room. In this corner are located two 10 and 7 by 10-inch Blake house pumps delivering water to tanks on the roof, and two Knowles 8 and 12 by 12-inch vacuum-return pumps working on the Van Auken system. A 12-inch exhaust riser extends to the roof and the steam is supplied to the radiators on the down-flow principle. Two Baragwanath heaters are also connected to the exhaust-steam line to furnish hot water through the building. On the boiler-room level is a sump in which all drains are collected and elevated to the sewer by means of Yoeman's electrically driven ejectors, one Worthington centrifugal pump direct connected to a 6x8-inch Marine Iron Works vertical engine, being held in reserve.

COLOR SCHEME

An elaborate color scheme has been adopted in this plant following the system outlined in the July 26 issue of POWER. Colors have been selected which will wear well and not change by heat or age sufficiently to be mistaken. The colors used are red, black, green, brown, white and yellow. The idea has been not to paint the entire lines of piping in different colors, but to paint markers on the various pipes in prominent positions where they will show to the best advantage. The marks are made in arrowhead form, the point of the mark indicating the direction of flow.

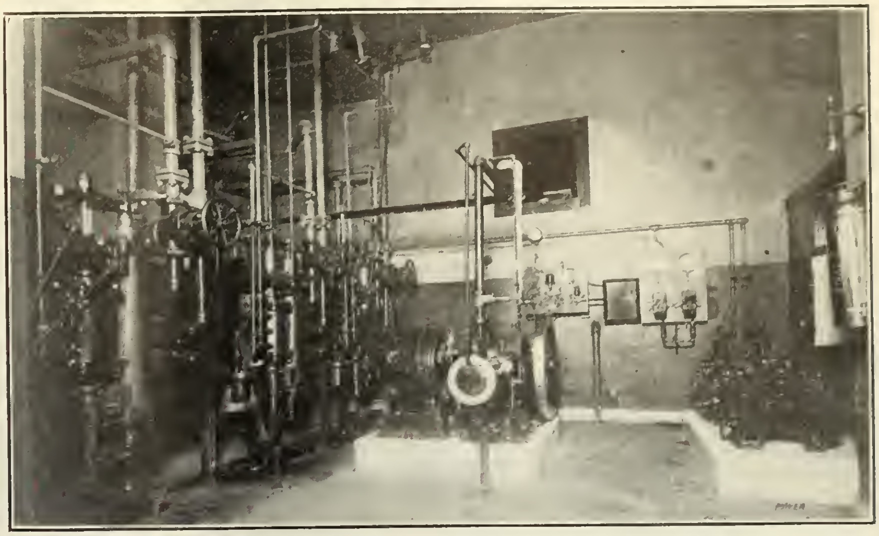


FIG. 7. ELEVATOR CONTROLS AND AIR COMPRESSOR FOR CLEANING PURPOSES AND THERMOSTAT SYSTEM

The high-pressure steam, reduced-pressure steam and exhaust-steam lines, which obviously are the most important, all are marked with black; this color being unmistakable, the distinction in these three cases being a wide mark for the high-pressure lines, two narrow ones for the reduced-pressure lines and a single narrow mark for the exhaust lines. Similarly, pipes for different auxiliary services are marked, as shown in detail in the article referred to previously. With this color scheme it is possible to tell at a glance what is inside of any pipe, and the direction of flow. It is easily and cheaply installed and is a great convenience to the operators.

Do We Waste 95 per Cent. of Our Fuel?

A correspondent writes: "In reading a book which has recently appeared, I came across this statement which astounded me: 'But 5 per cent. of the potential power residing in the coal actually mined is saved and used. For example, only about 5 per cent. of the power of the 150,000,000 tons annually burned on the railroads of the United States, is actually used in traction. Ninety-five per cent. is expended unproductively or is lost. In the best incandescent and electric-lighting plants but one-fifth of one per cent. of the potential value of the coal is converted into light.' This may be true, but it seems incredible that in the conversion of potential into mechanical energy there should be such an enormous waste. What are the facts in the case?"

The correspondent's statement is true to this extent and in this way:

A British thermal unit is the amount of heat necessary to raise a pound of water one degree Fahrenheit, and is equivalent to 778 foot-pounds of work.

A pound of good coal may contain somewhere around 14,000 B.t.u. or the equivalent of

$$14{,}000 \times 778 = 10{,}892{,}000$$

foot-pounds.

A horsepower is 1,980,000 foot-pounds per hour.

An engine which could produce a horsepower for an hour with the consumption of one pound of coal would therefore convert into work,

$$\frac{1{,}980{,}000}{10{,}892{,}000} = 0.18$$

or 18 per cent. of the energy inherent in the fuel.

The ordinary engine takes five or six times this much.

There is a great loss in the conversion of energy, after the engine develops it, into light.

On the other hand, we are limited by our environment and conditions. We can get, with exceptional boiler work, 80 per cent. of the energy of the fuel into steam, but an ideally perfect heat engine would take out of this steam only a fraction represented by

$$\frac{T-t}{T}$$

where

T = Absolute initial temperature or high heat level;

t = Temperature of rejection or low heat level.

Heat is convertible into mechanical energy only by working it from a higher to a lower temperature level. To get all the heat into energy the heat must be worked to the level of absolute zero. To take an hydraulic analogy, each pound of the water in a pond 1000 feet above the sea level has in it, due to its elevated position, 1000 foot-pounds of energy, which it will give up if allowed to fall to the sea. But if the configuration of the country is such that the greatest fall obtainable in the vicinity of the pond is 100 feet, an ideally perfect turbine could only get 100 foot-pounds out of each pound of the water, or 10 per cent. of its inherent energy.

$$\frac{H-h}{H} = \frac{1000-900}{1000} = 0.10$$

or 10 per cent.

The absolute zero of temperature is 461 degrees below the zero of the Fahrenheit scale. We live up on the plane where the temperature is around 70 degrees Fahrenheit, or

$$461 + 70 = 531 \textit{ degrees absolute.}$$

The cooling water which is used in the condensers will average about 35 or 40 degrees in the winter and 70 or 80 degrees in the summer. Unless we use impracticable quantities of it we must let it heat up to 100 or 110 in absorbing the heat necessary to maintain the lower temperature level, which is therefore fixed at not below 100 degrees Fahrenheit, or 561 degrees absolute.

The temperature of 180 pounds steam is 373 degrees Fahrenheit;

$$373 + 461 = \textit{degrees absolute.}$$

The efficiency of an ideally perfect engine under these conditions would be,

$$\frac{T-t}{T}=\frac{834-561}{834}=0.33$$

or 33 per cent.

Even with a 100 per cent. boiler and engine efficiency we could then under our conditions (and with our present knowledge) get, with an engine working between these limits, only 33 per cent. out of the fuel. With 80 per cent. boiler efficiency and a perfect engine we might readily get

$$0.80 \times 0.33 \times 100 = 26.4 \textit{ per cent.}$$

Steam engines are built which get out 60 odd per cent. of the energy due to the fall between attainable temperature limits, say,

$$26.4 \times 0.65 = 17 \textit{ per cent.,}$$

which comes pretty nearly to the 18 per cent. which, as was shown at first, would be represented by the pound of coal per horsepower which is in sight and has been attained by some exceptional engines.

The gas engine with a wider temperature range actually attains between 25 and 30 per cent.

If a *perfect* engine would only realize 33 per cent. there may be some doubt of the propriety of calling the other 67 per cent. a "waste." It is a waste only in the sense that the tailrace flow of a water power is a waste. In the present state of our knowledge we see no way in which it can be saved. But there is this curious fact. The animal organism is evidently a sort of a heat engine in which energy is produced by the combustion of food with a greater efficiency than we are able to obtain and with *no* perceptible temperature difference.

The average efficiencies of actual plants are much less than those quoted and the 5 per cent. which our correspondent mentioned is a very reasonable estimate.

Installing Tubes in Boilers

By H. S. Jeffery

A few hints upon the proper way to remove defective tubes and install new tubes in boilers. Special attention is given to the tools used and the manner of handling them.

The manner of installing tubes in boilers does not differ materially with various types of boilers. The hole in the tube sheet should be drilled as true as possible and should never be punched, as a punched hole does not have the same diameter for its entire depth, and the surface is more or less irregular and rough. It is good practice to drill the hole a little small and then ream out to the desired size, after which the edges are chamfered, as shown in Fig. 1.

In removing tubes from a boiler the head end of the tube is usually split, and the bead is cut from the firebox end of the tube. Then, by means of a maul and special tool, the tube is knocked out of the tube sheet and is removed from the boiler.

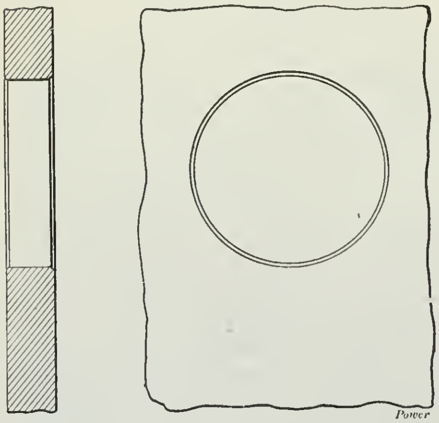


FIG. 1. EDGES OF HOLE CHAMFERED

A ripper, shown in Fig. 2, is used in splitting the tube, after which a slit is cut into the tube, extending back beyond the tube sheet about one inch. This allows the tube to be closed in and removed through its own hole in the sheet. If scale adheres to the tubes to the extent that they cannot readily be removed in this manner, then they may be taken out through a convenient handhole. If this is not in a convenient location, the holes in the tube sheet may be enlarged and the tube extracted through it.

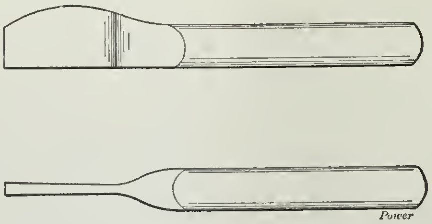


FIG. 2. RIPPER FOR SPLITTING TUBES

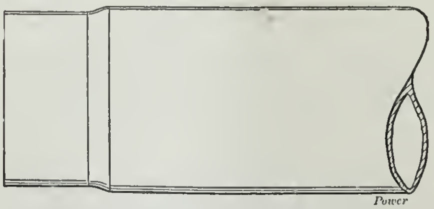


FIG. 3. END OF TUBE SWEDGED TO SMALLER DIAMETER

For this purpose it is necessary usually to enlarge the hole from 1/8 to 1/2 inch, depending upon the amount of scale adhering to the tubes. It should not, however, be enlarged more than necessary, as it must be reduced to nearly normal size when the new tube is put in. The holes of the tube sheet in direct contact with the flames and hot gases are usually cut smaller than the diameter of the tubes, the latter being swedged to a smaller diameter; see Fig. 3. In some cases, especially with the locomotive type of boiler, a copper ferrule is placed between the tubes and the sheets. This usually applies only to the sheets directly in contact with the flames and hot gases, although some manufacturers use a copper ferrule in both tube sheets.

One method of fitting tubes to enlarged holes is by the use of heavy copper ferrules which reduce the size of the holes; see Fig. 4. This practice, however, has resulted in accidents, there being several cases of late where the beads became weakened at the firebox end and allowed the tube to become loose, with the result that the steam pressure, acting upon the tapered section and upon the bead, forced the tube out of the boiler.

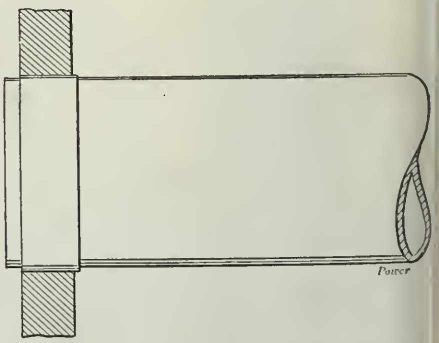


FIG. 4. HOLE REDUCED BY COPPER FERRULE

Another method is to place over the end of the tube a section of tubing about 6 inches long and of larger diameter, expanding to the tube sheet, as shown in Fig. 5. It is usual to have the short section about ¼ inch greater in diameter than the tube. When it is necessary to enlarge the hole to a greater extent a steel bushing is used to reduce it to fit the tube; see Fig. 6. The bushing is expanded into the sheet before the tube is installed and the latter is expanded against the bushing. The use of a bushing for this purpose, although more expensive than the copper ferrule, is justified as the safety of employees and those near the boilers should be considered first, and the expense last.

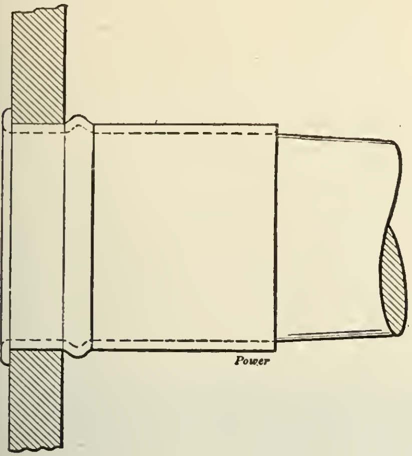


Fig. 5. Short Piece of Tubing Used to Reduce Hole

The replacing of tubes in an old tube sheet must be done with utmost care. The sheets should be carefully inspected and all scale removed from the inside surface around the hole. The edges of the hole should be chamfered, leaving a good fillet to prevent the tube from being cut by sharp edges or burs. The holes are liable to be out of round, in which case they must be reamed. In some types of boilers there is a marked tendency for the firebox-tube sheets to expand more in a vertical direction than in a horizontal. This distorts the hole, the vertical diameter being slightly greater than the horizontal diameter. The holes at the center of the tube sheet usually suffer most from such distortion.

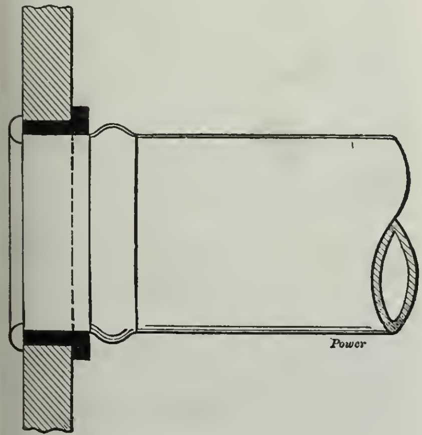


Fig. 6. Hole Reduced by Steel Bushing

Before installing a set of tubes attention should be given to the sheets to see if they are straight, and, if not, they should be straightened as far as possible. This is accomplished by means of bars and bolts. When the sheet has been straightened, a number of tubes, depending upon the size of the boiler, are temporarily fastened to both tube sheets to permit the removal of the straightening bars, after which the balance of the tubes can be inserted and the operation completed.

It is general practice, with tubes which are beaded at one end and the other end expanded only, to cut them the desired length before inserting into the tube sheets. Sometimes there is more or less variation in the lengths of the tubes, in which case the tube sheet is divided into sections, the tubes of each section being cut the same length. They are then marked for their respective sections, and when installed all will project from the sheet about the same distance. If the tubes are to be beaded at both ends, then the measurements should be taken with more care, and the variation between the sections should not exceed 1/16 inch. Some boiler manufacturers design their boilers so as to use a special length of tube. These tubes are usually about the right length and those that project too far beyond the sheets are cut off at the desired point by a special machine, although frequently a flat chisel and hammer are used. Tubes expanded into the sheet and not beaded should not extend beyond the sheet over 3/16 inch; those which extend too far beyond the sheets not only prevent proper expansion, but also permit the surplus part of the tube, which is not cooled by the water, to be burned or wasted away.

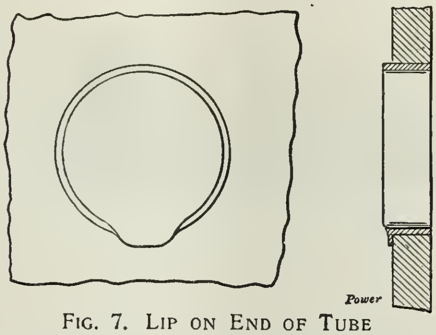


Fig. 7. Lip on End of Tube

Tubes are expanded into the sheets by means of a roller, or sectional expander called a prosser. Some boilermakers use both the roller and the prosser in order to make a neat fit. When the tube is inserted into the hole it should project beyond the sheets from 3/16 to ¼ inch, which amount is allowed for the bead. The tube is held in place temporarily by lipping the tube with a hammer, as indicated in Fig. 7. The lip will hold the tube in position, while it is being expanded. If the tube is expanded by the use of the prosser only, a very common practice, the latter should be turned about four times. This forces the tube out against the sheet, which would not be the case if the prosser were not shifted.

The use of a heavy maul should be discouraged for driving the mandrel. A four-pound maul is sufficient and the blows struck upon the mandrel should not be heavy enough to cause the tube sheet to warp or spring. The tubes ought to be worked to the sheet and if this is done properly the sheet will not be distorted. A large tapering pin should never be used to enlarge the tube. The copper ferrules are annealed before being put into place and must be a neat fit. If too small for the hole they may be stretched and reannealed. In expanding tubes with the roller expander the mandrel should be forced in only a short distance at a time; otherwise, lumps will form and make the rolling unsatisfactory.

In working leaky tubes expand them very lightly when using the prosser, turning it about three times, and use an ordinary hammer. In expanding a leaky tube it is well to remember that heavy pounding is not required to tighten the tube in the sheet, as the tube has already the full contour of the expander. Beading the tube is very important. A beading tool constructed with a sharp heel is very liable to result in the sheet being cut when the bead is calked. In Fig. 8 is shown a beading tool having a sharp heel, also the manner in which it cuts the sheet. In Fig. 9 is shown a beading tool with a square heel which prevents a groove from being cut into the sheet.

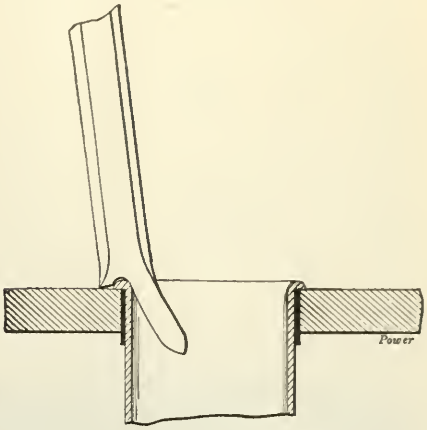


Fig. 8. Beading Tool with Sharp Heel

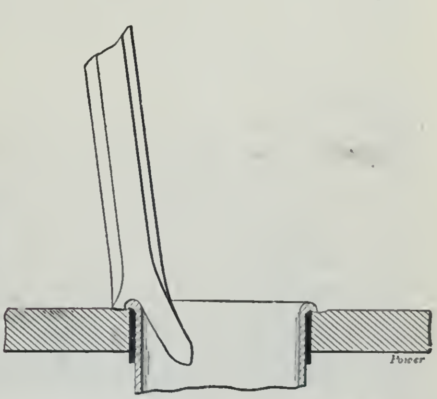


Fig. 9. Beading Tool with Square Heel

Boston Elevated Receiver Explosion

Some additional information on the receiver explosion which occurred in the Boston Elevated, Harvard Square plant. Receiver was 52 inches in diameter and was equipped with safety valve set for 50 pounds pressure.

In the October 11 number a short description was given of the receiver explosion which occurred on the morning of September 16, at the Havard Square power house of the Boston Elevated Railway, located at Boylston street, Cambridge, Mass. Further information is now available. The receiver connected the high- and low-pressure cylinders of a 4000-horsepower vertical compound engine which was direct connected to a 2700-kilowatt railway generator. The engine, as it looked previous to the accident, is shown in Fig. 1, and Fig. 2 shows the wreck.

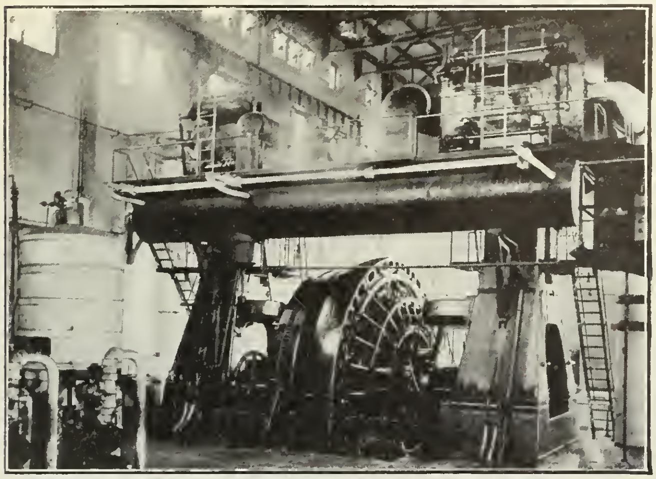


Fig. 1. Before the Explosion

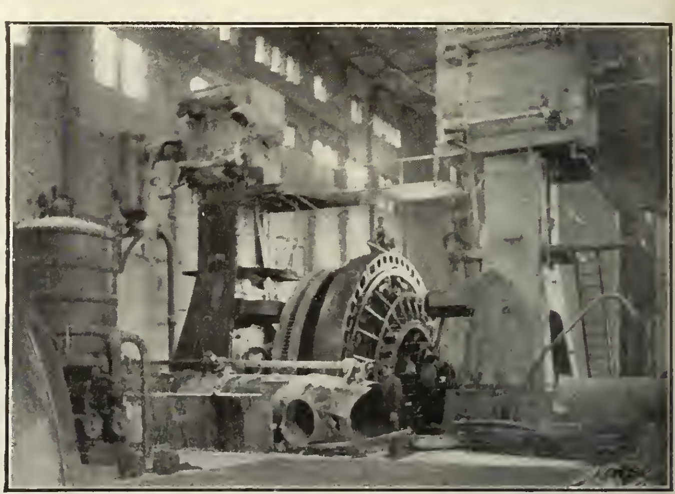


Fig. 2. The Wreck

The receiver which let go was made up of cast-iron sections, 1¼ inches thick and 52 inches in diameter. A 4-inch pop safety valve, set to blow at 50 pounds, had been placed on the receiver.

As the receiver burst, a great cloud of steam and water was carried over to the switchboard gallery which is shown in Fig. 3, and put about 15 lines out of service for a time.

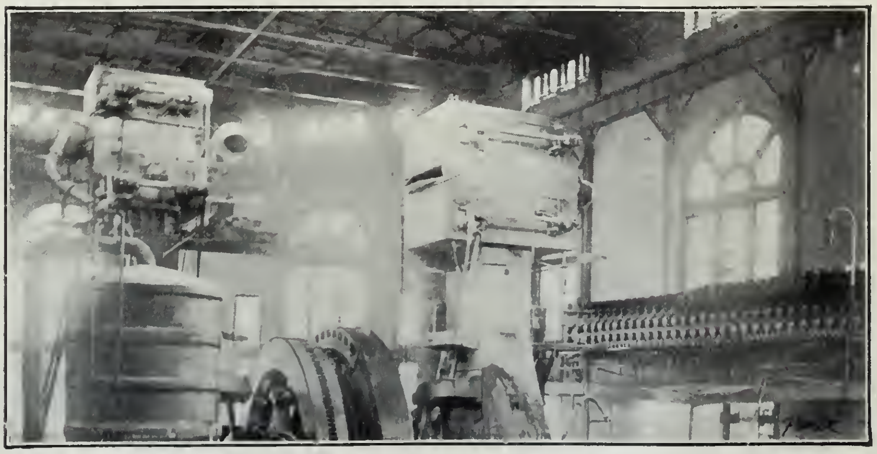


Fig. 3. View of Damaged Unit and Switchboard

A large piece of iron struck the governor of the engine which was carrying the load at the time of the accident and put this unit out of service for a great portion of the day until temporary repairs were made.

Fig. 5 shows the pieces of iron which went up through the roof, which is 40 feet above the engine, putting a 30-ton electric crane out of commission and tearing a great hole through the 6-inch concrete, strengthened by expansion metal. A large ventilator was smashed down and several small pieces were found several hundred yards away.

Fig. 4, a view of the end of the station taken from the Charles river road, shows how the windows were broken; it will cost several hundred dollars to repair the building. It will be an expensive job to repair the engine and generator, and several weeks must elapse before the unit will be ready for commission.

At the time of the accident there were seven oilers and engineers on watch. After the steam and asbestos cleared away, they went to work to see how many were lost, and found Ernest Lyons directly under the wreckage, where he had stood at the throttle on the high-pressure side. He was found to have several cuts and bruises about the body and one leg broken; although he is expected to recover, it will be six months or a year before he will be able to resume his duties as an engineer.

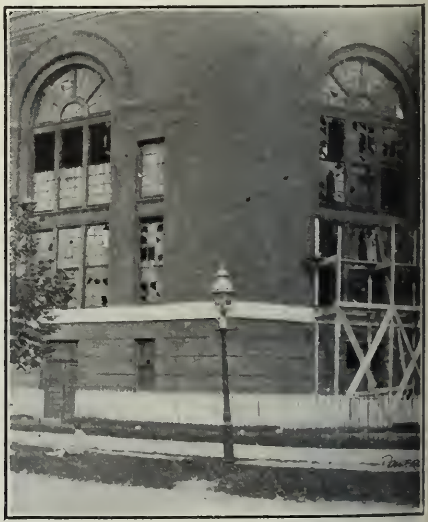


Fig. 4. Damage to Windows

At this writing the exact cause of the accident is not known. An investigation by the company is still pending and some information may perhaps be given out at a later date.

The Marine Engineer and His Work

By Thomas H. Heath

The marine engineer is handicapped by lack of space and adverse operating conditions. Breakdown usually occurs in rough weather. Conditions are being improved by the use of simplified machinery.

Among operating engineers, probably the marine engineer has to meet trouble under the greatest disadvantage. It is not with a fair breeze and smooth sea that he is most liable to "go up against it," but rather under the stress and strain of a heavy sea, when there is the greatest need for every wheel and cog to do its part, that the mishap or breakdown occurs. It is at this time when cool courage and experience count; when to meet the situation right may mean not only saving of time and damage to ship and cargo, but the lives of those on board as well.

This work has to be done generally under the most adverse conditions, in a rolling ship and with the limited space and means found aboard a vessel. It is bad enough to have things go wrong with good old mother earth beneath one's feet, but how much worse it is to have to do things in a tumbling ship that is wallowing around in the trough of the sea. It is hard enough to stand, much less make repairs.

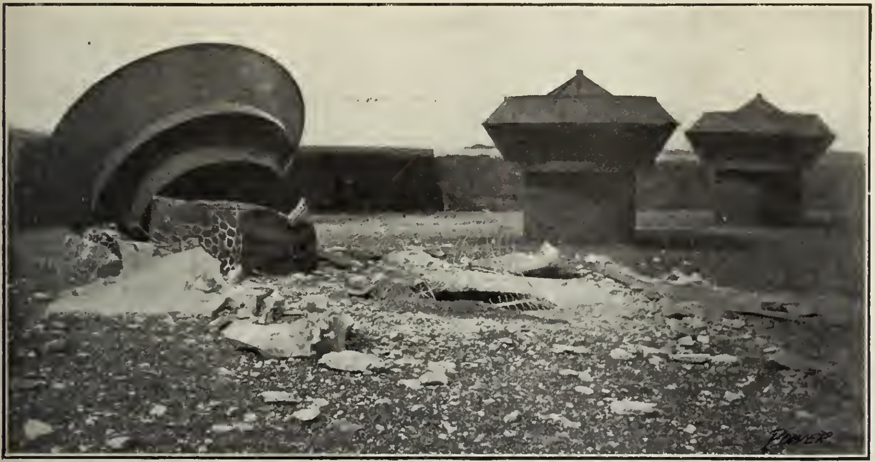


Fig. 5. Holes in Roof and Upset Ventilator—See Page 1828

Our wise old "Uncle Samuel," in his "Marine Regulations," says that no one can become a marine engineer until he has served his three years as oiler or fireman on the water. Then, if the would-be engineer makes good in an examination he gets his first "ticket," probably as a third or fourth assistant. After this it means another three or four years of marine engine-room service before he is allowed the coveted unlimited chief's papers. The man that has shown the ability and tenacity of purpose to go through this course of training is the man hat is generally ready for the emergency.

As in a similar sized power plant ashore, the chief engineer of ocean-going steamers takes no part in the active operation of the machinery while things are going right. He exercises a general supervision, keeps in touch with things below and devotes some of his time to daily reports.

While the vessel is loading or discharging cargo in port, the engine-room crew is busy getting things in shape again for the next cruise. Boilers are washed and looked after, bearings taken up, pumps looked into, repairs made where necessary and everything made ready for the next voyage, which may mean days and weeks without a slow down.

Coal has to be bunkered, supplies got aboard and fifty other things not forgotten; for, after once at sea, it is not a matter of telephoning uptown for something that is required. The tanks are filled with fresh water for human use and boiler feed. The evaporators are used only when a shortage occurs before reaching port.

Ocean-going vessels generally carry well stocked store rooms. Besides repair supplies and tools that would meet most requirements or needs, duplicates of many of the parts of the engines and auxiliaries most likely to give out are carried. Modern marine engines are multiple expansion, from compound to quadruple. When possible, the different parts of the engine are made interchangeable; for instance, crank brasses, eccentrics, crosshead brasses and many others. One or two spares of each of these with their bolts and accessories are carried to replace any one that may give out.

The crank shafts on most marine engines nowadays are made up in sections—a section for a cylinder, each section interchangeable so that in case of a broken crank shaft the part fractured may be taken out and the spare section substituted. For example, the triple-expansion crank shaft is made in three sections, each of which is exactly like the others; an extra section carried in reserve can be used to replace any one of the three in case of failure.

One of the most prolific causes of serious breakdown in a rough sea is the racing caused by the propeller being partially cleared of the water due to the plunging of the vessel. This, of course, is hard on the engines and it is at such times that things are liable to let go. Various governors have been devised in attempts to stop this racing but with engines that expand steam in three or four cylinders no scheme yet devised will come up to the old-fashioned way of handling the throttle.

One of the things that the marine engineer is constantly on the watch to prevent is sea water getting into the boilers. A little leak through some of the sea connections on the feed-pump lines or cooling water getting through into the steam space of the condenser may make trouble. Salt raises the mischief with the tubes and plates as well as the pipes and fittings. It also causes priming. A salinometer is a part of the equipment of all marine-engine rooms.

Bilge water is also one of the cares of the engineering department. Besides the regular bilge pumps, the circulating-pump suction line is arranged to take water from the bilge in case of an emergency.

Besides the main engines and boilers with their auxiliaries, there are the electrical plant, the refrigerating plant, the steering engine, the wireless apparatus and many other pieces of equipment. Most of these are huddled into about half the space or less that a similar equipment of equal capacity would occupy ashore.

Changes are taking place with the marine engineer as well as with his brother on land. The complicated reciprocating engines are giving way to the simpler and safer turbine. The safety of the ship does not now depend on a single propeller, for twin screws and even triple screws are replacing the single. Future requirements on water seem to point toward more power, greater speed and increased tonnage. Whatever the requirements may be, given a reasonable show, the marine engineer will be on hand to "deliver the goods."

Safety Stops for Steam Engines

By W. H. Wakeman

Types of safety stop consisting of special designs of valve, some of which are operated by steam and others by electricity.

A number of safety stops are made in the form of special valves connected to the steam pipe above the throttle valve. Several of these stops are herewith described.

In Fig. 1 is shown a butterfly valve located just above the engine throttle valve. The rod shown at the right is attached to a special governor run by a belt from the crank shaft. When the regular governor fails to control the engine speed, the rod is detached from the valve stem *A* and the weight *B* closes the valve.

Stops which close the throttle valve, or a special valve in the steam pipe are especially useful in connection with high-speed engines that are not fitted with disengaging valve gears. They are also applicable to engines that operate at lower speeds, but do not permit the application of a device that operates on the steam valves. However, they can be used on a Corliss engine, as illustrated in Fig. 2.

This device resembles the steam valve of a Corliss engine except that it is made shorter and the port is accordingly wider. If the speed of the engine is increased above a certain point a knock-off cam trips the latch and the weight attached to the valve-stem crank closes the valve. If the governor belt breaks, the same result is secured, for, although the movement is downward instead of upward, another part of the knock-off cam is rolled into contact with the latch, which is tripped, and the valve is closed. The governor must be blocked before the throttle valve is closed when the engine is to be shut down.

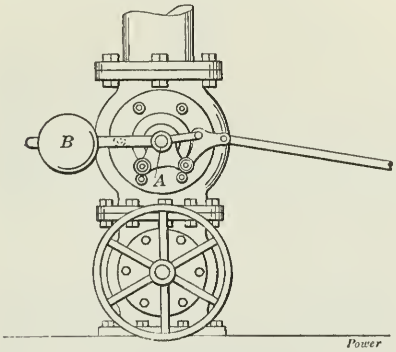


FIG. 1.

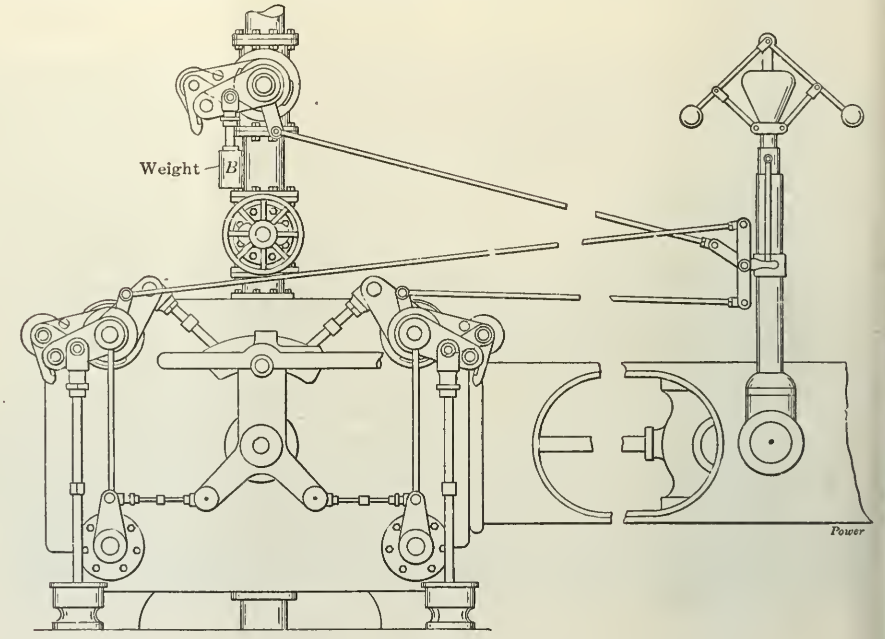


FIG. 2.

The difference between a stop that operates for both increase and decrease of governor speed, and one that acts for increase only, is that the latter stops the engine if the valve gear is so adjusted that the steam valves admit enough steam to run the engine at a dangerous rate at the shortest possible point of cutoff, when the load is very light. This condition can be produced by setting the collar on the governor spindle so low that the center weight cannot go as high as the engine builder intended. It is an easy matter to make this mistake, and it has been made in many cases where the engineer does not suspect it, therefore every engine ought to be tested for this defect. If the governor belt breaks the engine will first run faster for a few seconds and then stop.

In the case of a stop that operates for increase of speed only, as long as the governor belt is in good order and stays on the pulleys, there is no danger of a flywheel wreck from excessive speed, even if the steam valves remain open after they ought to be closed, because steam will be shut off by the independent valve, but if this belt fails and the engine begins to race, there is nothing to stop it.

Fig. 3 illustrates an independent auto-

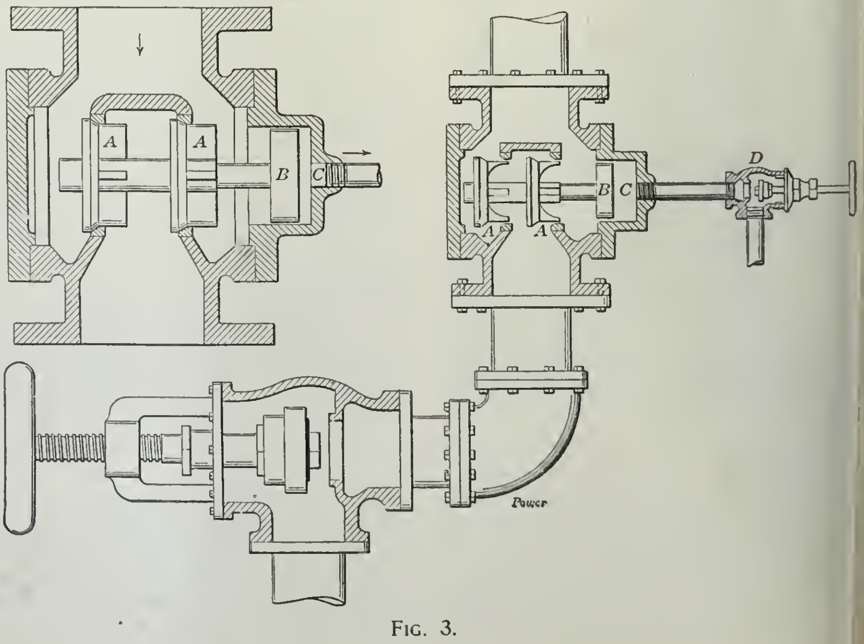


FIG. 3.

matic stop valve located above the throttle valve of an engine. Both are wide open allowing steam to pass freely to the engine. The view at the left of Fig. 3 shows the stop valve shut. Fig. 4 is an external view of the device which closed it and Fig. 5 is a sectional view of it. In Fig. 3 the stem and poppets *A* are in equilibrium, and, so far as they are concerned require but little external force to move them. They are held in the position shown by the difference in total pressure acting on opposite sides of the piston *B* as it allows steam to pass it until full pressure is secured in the chamber *C*, also in and through the angle valve *D* which is left open for this purpose. Valve *D* is used to shut off pressure when repairs or adjustments are required. The pipe from the valve *D* leads to the device shown in Fig. 4.

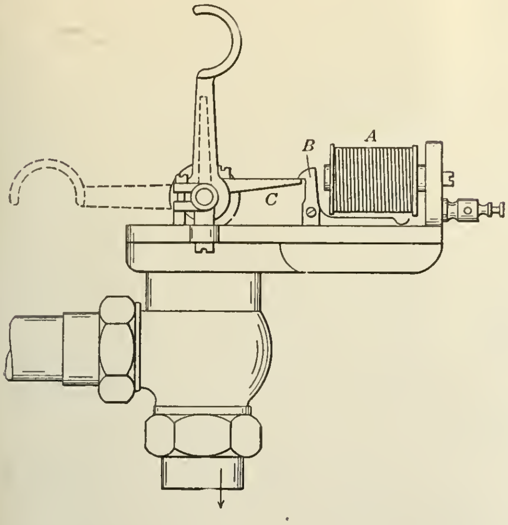


Fig. 4.

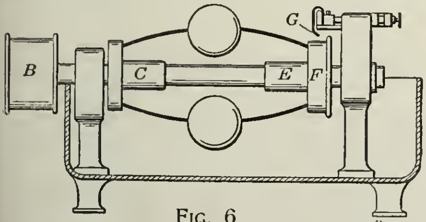


Fig. 6.

Pressure acting on *B*, Fig. 3, holds the valve open because the full area on the outer face is exposed to pressure, while the rod occupies a portion of the inner face, thus reducing the effective area. For illustration, suppose that the piston *B* is 4 inches in diameter, and the rod 1 inch. The steam pressure is 100 pounds on both faces of *B*. The effective area is 12.56 square inches on the outer face, giving a total force of 1256 pounds. On the inner face it is 11.78 square inches, and the total pressure is 1178 pounds, therefore, the difference is 78 pounds which operates to open the valve and hold it open.

The figure shown at the left of Fig. 3 illustrates the stop valve after steam is exhausted through the outlet *C*, thus removing all pressure from the outer face of *B* and, therefore, the full force of 1178 pounds operates to close the poppets *A A*, and promptly shut off the steam.

Fig. 4 is an automatic angle valve which is closed under normal conditions, but when an electric current is sent through the armature *A* it attracts the pawl *B* and releases the latch *C*, which is moved by a coiled spring through one-quarter revolution, as represented by the dotted lines.

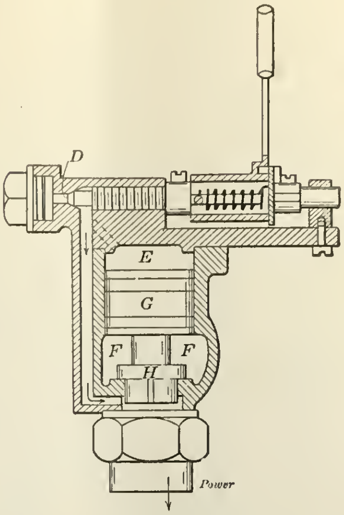


Fig. 5.

Fig. 5 is a sectional view of the same device taken at right angles to the preceding illustration, consequently the inlet does not appear. When the pawl releases the latch it opens the small valve *D*, allowing steam to escape from *E* through a passage shown by the dotted lines into the vertical space below *D* and thence into the exhaust pipe, as shown by the arrows. Full pressure acting at *F F* raises the piston *G*, also the poppet *H* and exhausts all steam from the space *C*, Fig. 3, and closes the automatic stop valve.

Current from a battery is sent through *A*, Fig. 4, when the circuit is closed by the speed-limit governor shown in Fig. 6. The pulley *B* is driven by a belt from the crank shaft. The part *C* cannot move endwise on the shaft, but *E*, at the opposite end, is free to slide. The flyballs are in position to represent normal conditions of speed and service, but a slight increase of engine speed throws them out further until the spring holder *F* comes in contact with *G*, completing the circuit and closing the automatic stop valve as already described.

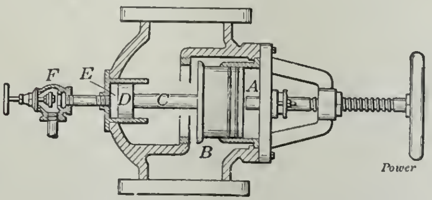


Fig. 7.

A hand-operated throttle valve combined with an automatic stop valve which does not utilize electricity in its operation is shown in Fig. 7. When the wheel is turned from right to left hand, the valve is opened, although the stem *A* and disk *B* are not connected mechanically. When this valve is closed by turning the wheel from left to right, steam pressure acts only on the left-hand side of the disk and as the valve stem is withdrawn by turning the wheel the disk follows it. The stem *C* and piston *D* are connected to the disk, but steam at full pressure fills the space *E*, therefore, under normal conditions piston *D* is in equilibrium and has no effect on the valve action. When steam is exhausted from *E*, however, pressure acts on the other side of *D*, closing the valve. The angle valve *F* is for the purpose of shutting off the closing attachment, but it should always be open when the engine is running.

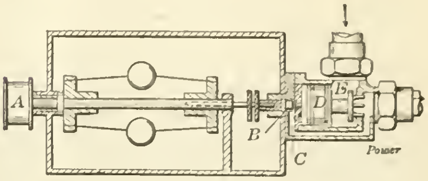


Fig. 8.

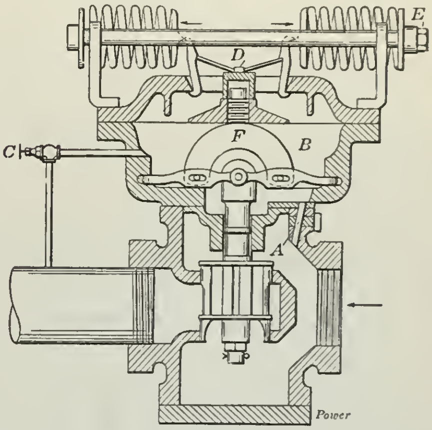


Fig. 9.

Fig. 8 shows a device for closing the valve by steam pressure. The pulley *A* is run by a belt from the crank shaft. The flyballs are mounted on flat springs and are shown in normal running condition. A small valve *B* closes the exhaust passage *C* and the steam pressure is equal on both sides of the piston *D*. The inlet receives steam from *F*, Fig. 7, but it cannot pass out when the valve disk *E*, Fig. 8, is closed. If the crank-shaft speed is increased the governor balls are spread by centrifugal force, causing *B* to open and let steam escape to *C*. As pressure is removed from one side of *F* it is forced toward the left, which relieves the pressure in the chamber *E*, Fig. 7, and the valve is closed. The automatic stop valve is usually located over a steam chest on a

Corliss or other long-stroke engine, while that shown in Fig. 8 is near the crank shaft, but this is not essential, as it is only necessary to reduce pressure in the exhaust pipe, and that can be done from a distant point. By using a tee in this pipe and running a branch line to a convenient point with a globe valve in it, the stop valve may be tested at pleasure. If the automatic device is wrecked by an accident the engine can be shut down without going to the throttle-valve wheel.

Fig. 9 illustrates a self-contained automatic stop valve that will shut off steam whenever the pressure is reduced on the outlet side. Steam enters at the right, as indicated by the arrow, and as the valve is closed, the only outlet is the small passage *A*, which allows steam to pass into the chamber *B* until full pressure is secured at this point, because the angle valve *C* is closed. This valve and the connecting pipes must always be larger than the passage *A*. Pressure acts on the diaphragm above *B* and raises *D* against the two spiral springs, which are thus forced apart as indicated by the arrows. The tension of these springs is regulated by the nut *E*. This tension determines the point at which pressure shall be shut off automatically. When nearly full pressure is realized in chamber *B*, the main valve is drawn upward by the diaphragm and connection *F*, allowing steam to pass through and equalize the pressure, after which *C* is opened. Assuming that this pipe supplies steam to an engine, and that on account of failure of the governing mechanism, the engine begins to race, taking steam so rapidly that pressure is reduced in the supply pipe and also in chamber *B*. The springs are then stronger than the steam pressure and the main steam valve is closed.

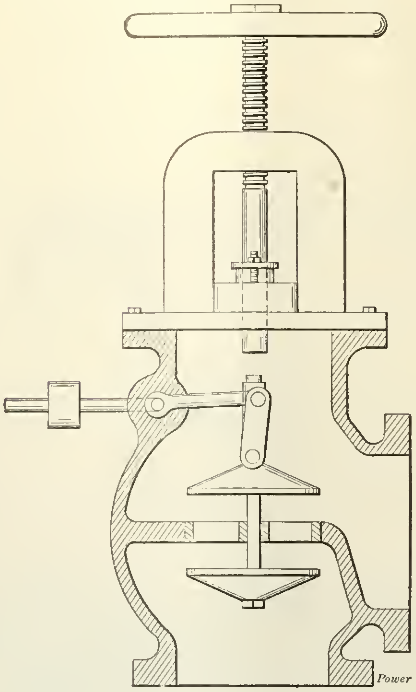


Fig. 10.

If the engine is heavily loaded and pressure is reduced in the supply pipe, the main steam valve will shut down the engine even when there is no danger. To overcome this objection the springs must be set to operate at a lower pressure. Care must be taken to avoid excessive difference in this respect, or else the valve will not operate when the engine "runs away." Suppose that the usual boiler pressure is 100 pounds and this valve is set to operate at 80. If the boiler pressure is allowed to fall to 80 the engine will be shut down. While this is an objection from the fireman's standpoint, it may not be an unqualified evil from the engineer's point of view, as it will be a great inducement to maintain a steady boiler pressure, thus securing good results in the use of steam. If the pressure falls below 80 at night or whenever the engine is shut down, the automatic valve will close, making it impossible to start until the boiler pressure is raised, or the tension on the springs is reduced.

Fig. 10 illustrates an automatic cutoff valve which does not depend on a reduction of pressure for its operation. The internal parts are balanced by a weight on the outside, and a dashpot prevents chattering when steam is taken rapidly and cut off quickly. Under normal conditions, this valve remains open and will deliver a reasonable quantity of steam, but if the engine "runs away" the extraordinary flow of steam caused by excessive speed, closes the valve and shuts off the supply of steam.

Darky Engineer Had a Hot Crank Pin

Somewhere in the "good book" it says, "Let there be light: and there was light." This quotation brings to mind a little experience one of the editors of POWER had some time ago when visiting a manufacturing town in the sunny State of North Carolina.

Desiring to observe usual formalities, the office of a small cotton mill was hunted up, but the door was locked and the management was evidently allowing things to take their own course, although it was long past the regular time for starting up after the lunch hour. Wishing to view the power plant and trusting that no unpardonable sin would be committed in doing so without a permit, the visitor passed, in through the boiler room to the engine room, where he found the coal-black darky in charge studiously perusing his Bible. The darky, by the way, was a substitute for the regular engineer who happened to be sick.

Whether "Sambo" was reading the above quotation or not will never be known, but he was deeply absorbed in whatever he was reading, so much so in fact that he did not at first notice the arrival of a visitor or that the crank pin of the 200-horsepower Corliss engine was smoking. Upon his attention being called to it, the darky closed his book and grabbed an oil can, making the remark as he did so, "I done smell dat 'ar smoke, but I dunno whar it cum frum."

The visitor, not feeling in a position to advise slowing down the entire mill in order to loosen up the brasses, suggested that he get the oil cup to feeding, which he had neglected to do upon starting the engine at the end of the dinner hour, to copiously flood the pin and to hustle for instructions as to what had better be done. Frantic appeals were then sent to the residence of the mill manager, who came in on the run a short time afterward.

Sometimes good intentions are thrown to the winds and are not appreciated; it was so in this case. While the darky was hunting for the manager, the visitor was industriously squirting oil on the hot crank pin. Upon his arrival, the manager proceeded simultaneously to cuss the darky and to shut down the engine. Now engineers know that the babbitt in brasses and a hot crank pin, particularly when as hot as this one, have a strong affinity for each other, especially when the engine is stopped.

As the evident intent of the manager was to shut down the engine at once, the visitor timidly suggested that he keep the engine turning over slowly until he had his brasses loosened up a trifle and cooled down. The suggestion was not kindly received, for the manager, quickly turning, said: "Who the —— are you! What the —— do you want?" Then the visitor was emphatically informed that he, the manager, was a graduate from two or more universities, and that he had also made good with a large electric-manufacturing company. Feeling properly squelched, the visitor humbly withdrew, wondering as to the fate of the darky.

The next day while visiting a neighboring machine shop, one of the machinists stated that he had a Sunday job at the ———cotton mill, which would indicate that the manager had again "made good."

Tests on a 12x30 Corliss Engine

By J. A. Polson

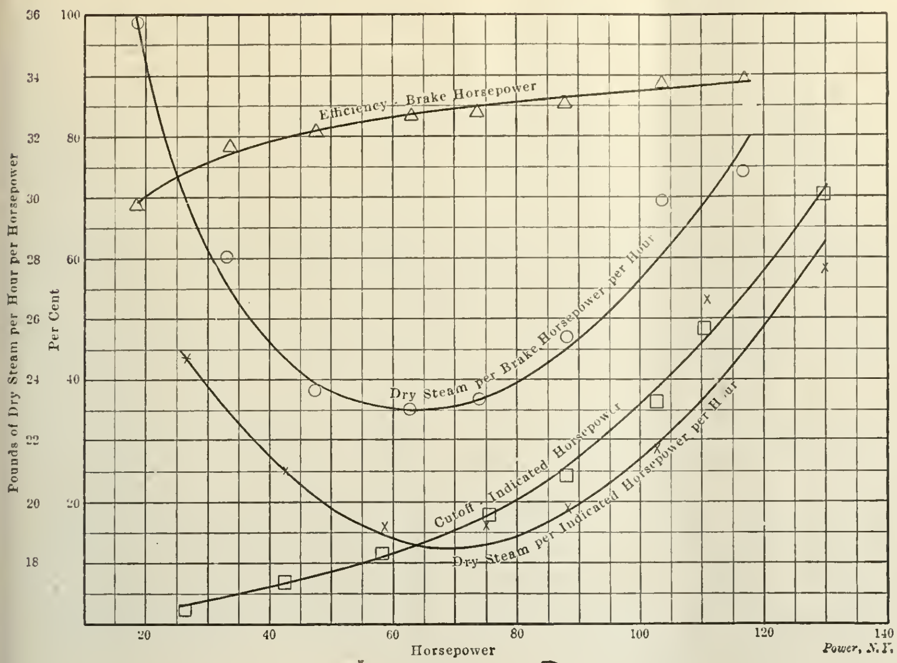


Fig. 1

The engine on which these tests were run is a standard 12x30 Corliss engine built by the Nordberg Manufacturing Company, of Milwaukee, Wis. It is of the long-range cutoff type and has three eccentrics, two of which operate the steam and exhaust wristplates and the third operates the cutoff mechanism in connection with the governor.

The engine is a part of the permanently installed equipment of the engineering laboratory of the Michigan State Agricultural College at East Lansing, and is used for testing purposes and for instruction for engineering students.

The power developed by the engine is absorbed by a friction brake of the prony type, the pressure being determined by weighing on a platform scale. The brake wheel is 6 feet in diameter with a 24-inch face, and provided with internal flanges for water cooling.

The data from which these results have been worked up by the writer were taken some time ago by the students in the laboratory.

In running these tests the conditions were kept as nearly uniform as possible for an average period of 50 minutes, with 10-minute readings. Indicator diagrams were taken simultaneously from each end of the cylinder, using Crosby indicators and reducing motion. The revolutions were taken by a continuous revolution counter. The condensed steam was weighed on platform scales, two tanks and scales being used to facilitate weighing. The results have been worked up in the usual way and averaged, and are given in the table on following page.

The quality of the steam was nearly constant at 96.8 per cent. The low quality of the steam is accounted for by the fact that the engine is about 200 feet from the power house. The quality of the steam was determined by a throttling calorimeter at the engine throttle. The steam pressure given in column 2 of the table was also taken at the throttle valve.

The series of tests given cover pretty well the range of the engine under the given pressure, as can be seen by a glance at the sample diagrams given in Fig. 2. Those marked *a*, *d* and *h* are average diagrams taken under the tests marked by the same letters in the table.

In Fig. 1 the results given in the table are plotted on a common base of horsepower and plainly show the behavior of the engine under increasing load.

The lowest steam consumption is 23 pounds per brake horsepower-hour and occurs when the engine is developing about 62 brake horsepower. The steam consumption is 19.2 pounds per indicated horsepower at this load. The diagrams

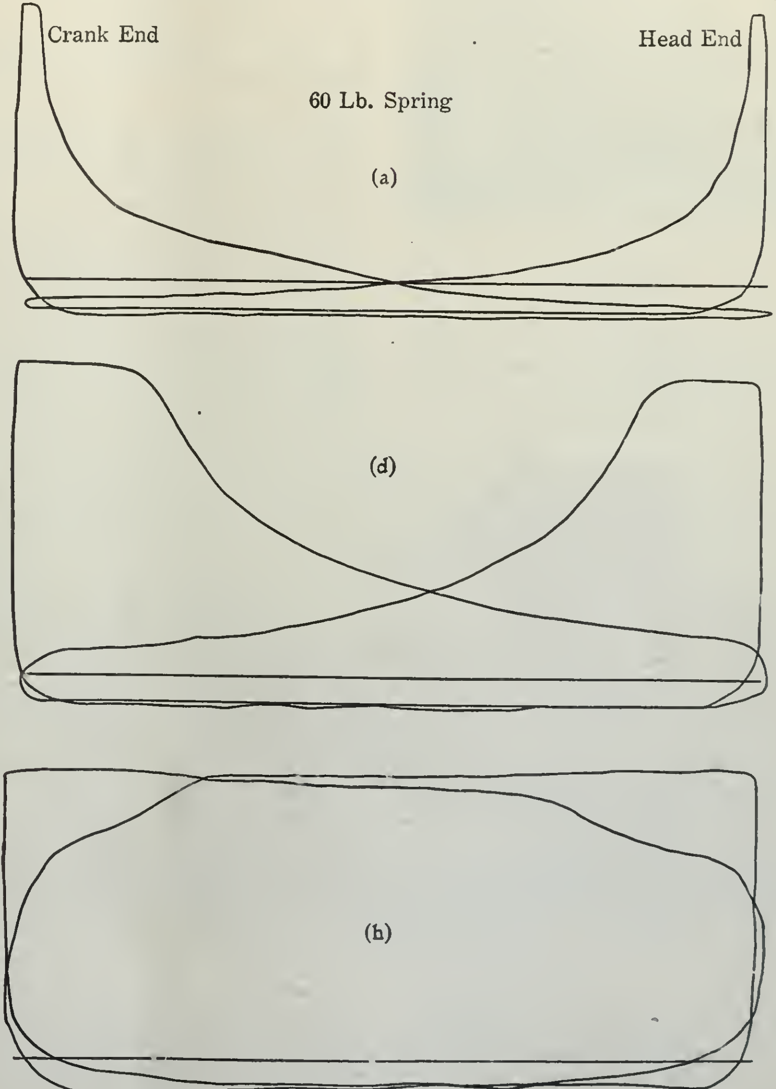


Fig. 2. Sample Diagrams

marked *d* in Fig. 2 are representative diagrams from this test.

As the long-range cutoff is a special feature of this engine, it is interesting to note how the cutoff increases with increasing load. This is the average cutoff for both ends and is shown by a single curve. At a load of 25 horsepower cutoff takes place at 3 per cent. of the stroke, at 80 horsepower cutoff has increased to 20 per cent. of the stroke and at 130 horsepower cutoff occurs at three-tenths of the stroke. The mechanical efficiency ranges from 69 to 89.9 per cent.

RESULTS OF TESTS.

No. of Test.	Av. Steam Press., Gage.	Av. Vacuum, Inches Mercury.	Rev. per Minute.	Horsepower. Brake.	Horsepower. Indicated.	Horsepower. Friction.	Mechanical Efficiency, Per Cent.	Dry Steam Per Hour. Brake Horsepower.	Dry Steam Per Hour. Indicat'd Horsepower.	Average Cutoff, Per Cent.
1	2	3	4	5	6	7	8	9	10	11
a	94.5	20	89.56	18.25	26.48	8.23	69.0	35.7	24.6	2.5
b	94.8	20	89.32	33.17	42.6	9.43	78.2	28.0	21.0	6.4
c	94.1	20	89.0	47.25	58.38	11.13	81.0	23.6	19.1	11.2
d	95.4	20	88.74	62.44	74.8	12.36	83.4	23.0	19.2	17.5
e	96.0	20	87.88	73.75	87.1	13.35	84.6	23.3	19.7	24.1
f	96.0	20	86.64	87.78	102.6	15.82	85.5	25.4	21.7	36.3
g	93.5	18	85.4	103.1	115.6	12.5	89.2	29.9	26.6	48.0
h	91.5	18	84.14	116.4	129.6	13.2	89.8	30.8	27.7	70.5

Locomotive Boiler Explosion

By F. Krebe

Low water given as cause of explosion which killed the engineer and fireman and, with the exception of the trucks, practically destroyed the engine.

The accompanying photographs show the remains of a locomotive which exploded on October 1, about three miles west of Tennessee Pass, on the Denver & Rio Grande railroad. The photographs were taken in the yards at Salida, Colo., immediately after the wreck was brought in. Three locomotives, two pulling and one pushing, were taking passenger train No. 2 up a steep mountain grade, when the front engine, at the head of the train, exploded. The boiler proper, as shown in Fig. 1, was blown 200 feet forward, and about 80 feet to the side of the track. The firebox sheets, or lining, were blown 600 feet to the side, except the back head, in which the fire doors are placed. This was blown forward 600 yards. It is a mystery to all concerned how the back head could have been blown forward, unless the boiler turned around before the explosion occurred. The engine trucks remained on the track, and it required only about an hour's work to get them in shape to be pushed on a siding close by. Fig. 2 shows the trucks just as they stood after the explosion, with the exception that the side rods were taken off to permit bringing the trucks into the yards. Fig. 3 shows the firebox sheets as they were picked up and placed in a gondola car.

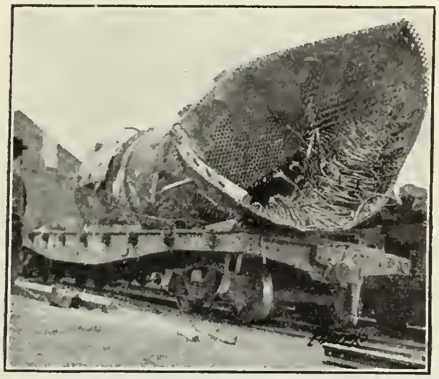


Fig. 1. The Boiler Proper after the Explosion

The Government inspection following the explosion resulted in a verdict of low water as the direct cause of the explosion, and this seems to be the consensus of opinion of others who examined the plates, and who were competent to know.

The body of the engineer was found 250 feet up on the side of the mountain, with both legs completely gone; these were found close to the rear end of the train. The fireman's body was blown across the river 500 feet from the train, and was frightfully mangled.

The second engine escaped with but little damage. The head light was pushed off, and the engine-cab windows were broken. Neither fireman nor engineer of the second engine were hurt, and the two engines proceeded with the train.

The engine which exploded was only two years old, and had just been put in good repair. The life of an engine o these mountain roads is about five years at the end of which time they must b practically rebuilt.

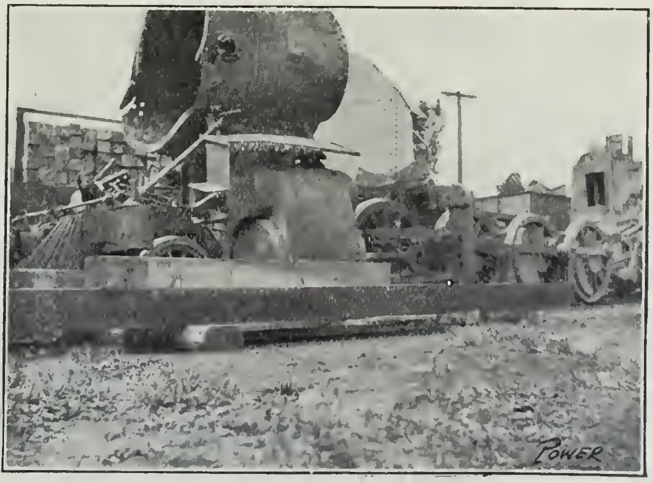


Fig .2. Trucks and Front Course of Boiler

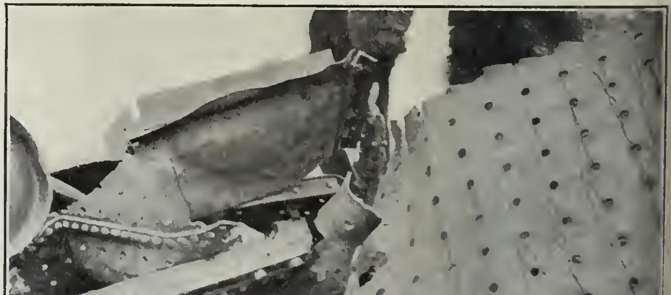


Fig. 3. Firebox Sheets Loaded onto a Gondola

Notes on Causes of Boiler Failure

By A. A. Adler

> *Among the causes due to operating conditions are overpressure, overheating, scale, lamination of the metal and, possibly, the spheroidal state of water. Structural causes include imperfect methods of bending the sheets and driving the rivets and the weakness characteristic of the lap joint.*

Some boilers explode clearly from overpressure, due to erroneous gage readings and defective or erroneous setting of the safety valves. Overpressure may also be wilful. There is no remedy for this state of affairs further than the elimination of the cause.

Overheating of the plates and tubes is another cause. Such a condition is evinced by the bulging of the steel due to the internal pressure, and the plastic condition of the steel at the high temperature. Scale, lamination in the steel, low water and possibly the spheroidal state of water may be the causes. The spheroidal state of the water may be defined as that state where the water is separated from the plate by a film of steam through which the transmission of heat is somewhat retarded, sufficiently so to cause a rise in temperature of the plate. This latter condition is probably due to forcing the water to evaporate faster than the circulation can bring new water surfaces to the plate. The remedy for the various conditions is evident.

There are cases on record where the boiler exploded on the opening of a valve after the boiler had been lying idle for some time. Aside from the structural defects, this is the most dangerous condition that there is to contend with.

It is a known fact, that the temperature of water can be raised several degrees above its boiling point, when it is in an absolutely quiet condition and holds no entrained air in it. The temperature may likewise be lowered several degrees below the freezing point. Suppose a boiler to be full of water in a superheated condition due to its quiescent state. If this state be disturbed by drawing off some steam, the temperature of the water will immediately drop a certain amount and the surplus heat will convert some water into steam, and thereby raise the pressure suddenly. The temperature of the water will then be the same as that of saturated steam at the new pressure.

What the extent of this pressure rise is I am not prepared to say very definitely. As the condition exists, it seems as though it is not a negligible quantity. A case will be taken, however, and the extent of the superheating of the water necessary for a certain rise in pressure will be estimated.

Assume a boiler 5 feet in diameter and 16 feet long. The weight of water contained in this boiler when two-thirds full, is

$$5 \times 5 \times 0.7854 \times 16 \times 62.4 \times \tfrac{2}{3} = 13{,}067$$

pounds of water, neglecting the space occupied by the tubes. The steam space for this boiler being ⅓ its capacity is,

$$5 \times 5 \times 0.7854 \times 16 \times \tfrac{1}{3} = 104$$

cubic feet.

Heat Content at 100 Pounds Gage Pressure above 32 Degrees Fahrenheit

$$\begin{pmatrix}\text{Total heat}\\ \text{of water}\end{pmatrix} = 13{,}067 \times 305.8 = \quad 3{,}996{,}000 \text{ B.t.u.}$$

$$\begin{pmatrix}\text{Total heat}\\ \text{of steam}\end{pmatrix} = 104 \times 0.257 \times 1188.8 = \quad 31{,}774 \text{ B.t.u.}$$

Total heat of boiler................ 4,027,774 B.t.u

Heat Content at 150 Pounds Gage Pressure above 32 Degrees Fahrenheit

$$\begin{pmatrix}\text{Total heat}\\ \text{of water}\end{pmatrix} = 13{,}067 \times 333.9 = \quad 4{,}363{,}000 \text{ B.t.u.}$$

$$\begin{pmatrix}\text{Total heat}\\ \text{of steam}\end{pmatrix} = 104 \times 0.3626 \times 1195 = \quad 45{,}063 \text{ B.t.u.}$$

Total heat of boiler.... .... 4,408,063 B.t.u.

The difference in boiler heat content is,

$$4{,}408{,}063 - 4{,}027{,}774 = 380{,}289$$

B.t.u., and the superheat necessary to produce this result is,

$$380{,}289 \div 13{,}067 = 29$$

degrees approximately.

The question now arises, is such a rise in temperature probable? Water may be heated as high as 290 degrees Fahrenheit at atmospheric pressure, which is 78 degrees above its boiling point.* It is, of course, questionable whether this same rise occurs at the higher pressures, but as I have not gone to the extreme in the example, the reader may admit its probability.

To show the effect of this sudden pressure rise, assume the boiler pressure to be as before, 100 pounds. Then, assume the pressure to rise suddenly to 150 pounds. This sudden increase of 50 pounds produces a momentary strain in the shell equivalent to a steady pressure of 100 pounds, and the resulting strain in the boiler shell is that due to 200 pounds. Should the boiler be near its limit of strength due to the 100-pound pressure; this suddenly applied increase of pressure might be sufficient to cause rupture.

Some people seem to think that water-hammer caused by the sudden opening of a valve is a cause of explosion, but I do not give much credit to this, as the steam that is drawn off may be less in amount than the fluctuation caused by an engine under a variable load.

Possible Structural Causes of Failure

Tubular boiler plates are made by putting a flat plate in a bending roll and rolling it until it assumes a circular shape. This is done with the material cold, and hence strains are set up in the plate which exceed its elastic limit, otherwise the plate would immediately return to its original shape. In the ordinary process the bending roll is unable to bend the end of the plate with the result that this portion of the plate does not have a circular shape but remains straight. As this portion forms the joint and, as will be shown later, the joint is a very important place, special precautions should be taken to have it conform to a true circular shape. Possibly cold rolling does not injure the boiler very much, but it must, nevertheless, be treated as an uncertain element in the design. Cold rolling increases the tensile strength of the plate at the expense of its ductility.

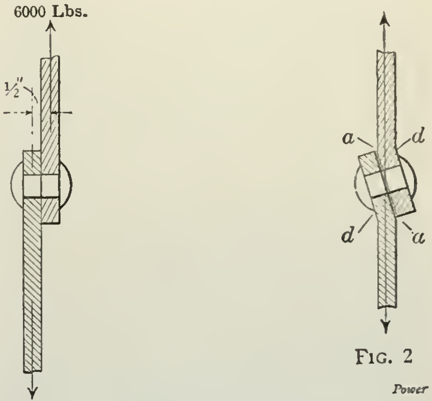


Fig. 1 Fig. 2

Lap-riveted Joint and Bending Effect Produced by Stress

The cooling of the rivets sets up unknown stresses, drifting, if permitted, has its attending dangers, and overpressure on the rivet produces strains in the metal at the sides of the hole. Even the method of supporting the boiler and the introduction of diagonal bracing, set up strains that are usually lost sight of in the design. In fact, all of the shop operations tend to impair the value of the finished product.

When a piece of lap-riveted joint is put in a testing machine the position of the plates with respect to each other will change from that shown in Fig. 1 to that in Fig. 2 on failure of the test specimen. The line of stress tends to become straight and when the forces are not in a straight line, there is a bending tendency, which increases the strain in the joint.

For the purpose of computing the amount of this strain assume a plate of

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*Ennis—"Applied Thermodynamics."

60,000 pounds ultimate tensile strength. Using a factor of safety of 5 and neglecting the so called efficiency of the joint, the allowable stress on the plate will be 12,000 pounds per square inch. If a plate ½ inch thick is used and a strip 1 inch wide is considered, working stress will be 6000 pounds on the strip, as its area is ½ square inch. When one piece of plate is riveted to another, the distance between the forces acting on the center of gravity of the plates is ½ inch and the bending moment for this condition of loading is,

$$\tfrac{1}{2} \text{ load} \times \text{length of lever arm},$$

or

$$\tfrac{1}{2} \times 6000 \times \tfrac{1}{2} = 1500$$

inch-pounds. This bending tendency must be withstood by the internal resistance of the plates. Since,

$$\text{Fiber stress} = \frac{\text{Bending moment}}{\text{Section modulus}}$$

and the section modulus for a strip 1 inch wide and ½ inch thick is $\frac{1}{6}\, b h^2$, where $b = 1$ inch and $h = \frac{1}{2}$ inch, or it is

$$\tfrac{1}{6} \times 1 \times \tfrac{1}{2} \times \tfrac{1}{2} = 1/24.$$

That the joint bends in two places must also be taken into consideration. Hence, the resulting stress is half of that which would be caused by bending in one place and the section modulus as 1/12 instead of 1/24.

The bending stress in the plate is, therefore,

$$\frac{1500}{\frac{1}{12}} = 1500 \times 12 = 18{,}000$$

pounds at the outermost fibers. Since in a plate that is bent, there exists a tension on one side *a*, Fig. 2, and a compression on the other side *d*, and in addition there is an initial stress of 12,000 pounds to the square inch, the resulting total stress in the joint is

$$18{,}000 + 12{,}000 = 30{,}000$$

pounds tension at *a* and

$$18{,}000 - 12{,}000 = 6000$$

tension at *d*.

The plate is then up to the elastic limit, except for the reasons that will be noted. Had a thinner sheet been assumed, the stresses would not have run up quite so high; they are, nevertheless, very much greater than is sometimes supposed. The material when stressed up to its elastic limit does not indicate failure, but its reliability is impaired. The boiler may be somewhat out of round which circumstance also may cause the stress to become still higher.

The corrosive action of water seems to be much more effective at places where the stress is high, due probably to the increased porosity of the material when under stress. As the corrosion reduces the section of the metal, the stress per unit of area increases and a more vigorous corrosive action follows with the result that the dangerous grooving takes place which is so often seen in lap-riveted boilers.

The result of the foregoing analysis is sufficient cause to reject this type of joint for boiler uses at least.

The problem is somewhat complicated by the strengthening effect of the heads of the boiler. The strengthening effect is considerable for short drums, and gets less as the boiler gets longer. Again, as the plates bend the forces have their lever arm decreased and thus the bending tendency is somewhat diminished. Some tension is also put on the rivet.

The stress as it is commercially figured for the shell, is not the exact stress that exists in the plant. A stress of 6000 pounds on the longitudinal seams is accompanied by a stress of 3000 pounds on the girth seams, and as these stresses are at right angles to each other the actual stress will be nearer to the resultant of the two. In the case in hand it will be

$$\sqrt{6000^2 + 3000^2} = 6700$$

pounds. We should proceed very cautiously when it comes to advocating a reduction of the factor of safety, or what amounts to the same thing, the carrying of a higher pressure on the boiler than the one assumed in its original design. It can be seen that the computation of stresses in a boiler under internal pressure is only approximate, and the only reason that exists for figuring boilers as we do, is because we have had a moderate degree of success in so doing.

From the foregoing the following conclusions may be drawn:

First. In order to prevent a superheating of the water, the blowoff should be opened for a short time or the safety valve should be lifted when the boiler is considerably below its working pressure. A sudden rise in pressure will then probably not exceed the actual working pressure.

Second. Boiler shells should be formed hot and bent to a true circle and thoroughly annealed before final riveting so as to eliminate as much as possible all initial internal strains.

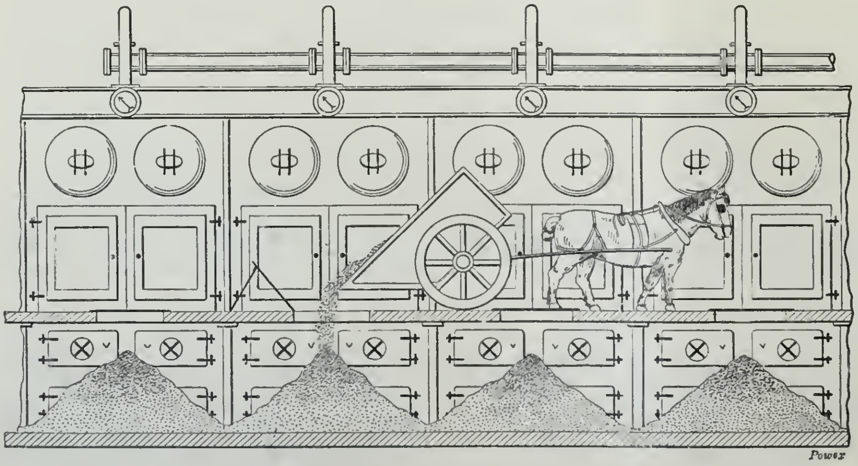
Third. Rivets should be driven hot, and with a pressure not more than experience shows to be sufficient to entirely fill the hole when cold.

Fourth. Lap-riveted joints should not be used for longitudinal seams on boilers. The double butt-strap joint is essential, and should have inside and outside plates of equal thickness to prevent eccentric loading. The plates should be bent to their respective contours on the inside and outside of the boiler shell.

In conclusion, it might be mentioned that there are a number of other abuses that a boiler may be subjected to, but as they are self-evident, they need not be taken up here.

Novel Method of Delivering Coal

The problem of coal delivery to a boiler room is sometimes difficult to solve, and frequently novel devices and peculiar arrangements are used.



COAL-UNLOADING PLATFORM IN BOILER ROOM

At one power plant the boiler-room floor is about 12 feet below the street level. The coal is delivered in wagon loads and owing to the surrounding buildings can be delivered to the boiler room only at each end. This necessitated wheeling the fuel to the central boiler.

In order to deliver the fuel in front of all the boilers, of which there were twelve, a trestle platform was built, extending the entire length of the boiler room, and strong enough to safely hold a two-horse team and load of coal. In front of each boiler a trap door was hung on strong hinges.

The wagon is stopped at the proper place and the trap door lifted. The coal is then dumped and falls through the opening to the floor below. Then the trap is closed and the wagon pulled off at the opposite end from which it entered.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Induction Motor Hand Starters

By R. H. Fenkhausen

The subject of induction-motor control may be divided for convenience into four divisions:

1. Hand control for constant-speed service.
2. Hand control for varying-speed service.
3. Automatic control for constant-speed service.

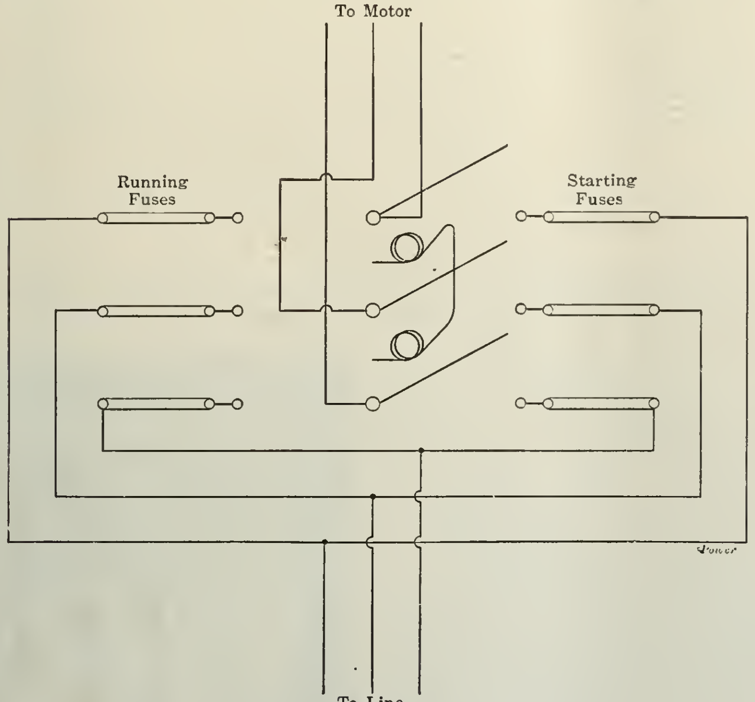


Fig. 1. Safety Spring on Starter Switch

4. Automatic control for varying-speed service.

The present article is confined to the consideration of apparatus commonly used for hand control in constant-speed service.

Squirrel-cage Motors

As the type of constant-speed induction motor most widely used is the squirrel cage, the control apparatus adapted to this type of motor logically comes first in the list. Squirrel-cage motors not exceeding 5 horsepower in size require no other starting device than the main line switch and may be started with full voltage on the primary winding. When started in this manner five or six times the full-load current is required to exert full-load torque. As the fuses required to carry this heavy starting current would be far too large to protect the motor when running, it is advisable to install a double-throw switch and two sets of fuses, one for starting and the other for running. The side of the double-throw switch equipped with the heavy fuses for starting should be provided with a spring placed so as to prevent a thoughtless operator from leaving the motor running with the starting fuses in circuit. This arrangement and the switch connections are shown diagrammatically in Fig. 1. When circuit-breakers are used instead of fuses, a tap may be taken off the overload coil of the circuit-breaker as in Fig. 2, the connections being such that fewer turns are in circuit during the starting period.

Squirrel-cage motors of more than 5 horsepower are almost invariably started with reduced voltage on the stator winding in order to prevent taking such a heavy starting current as to cause objectionable voltage fluctuations in the supply system; and also to reduce the heavy strains that would occur in shafting, belts, etc., were the motor forced up to speed suddenly.

Starting with Reduced Voltage

Four methods of reducing the voltage applied to an induction motor are available:

1. Special low-voltage taps from the supply transformers.
2. Star-delta or series-parallel connection of the stator windings, as represented in Fig. 3.
3. Noninductive resistances in series with the divisions or "phases" of the stator winding.
4. Autotransformers (commonly called compensator starters when used for motor starting).

The first method is applicable only when the motor is supplied from transformers situated nearby and as a 50 per cent. tap is the only one likely to be available in standard transformers, only one magnitude of starting torque can be obtained.

The second method requires that several extra leads be brought out of the motor frame, and, besides, is open to the same objection as the first method that only one starting voltage (50 per cent. in a two-phase and 57 per cent. in a three-phase motor) is available. Where the required starting torque is not excessive this is not a serious objection, but in service where the starting load is liable to be heavy, some other method of starting should be adopted; otherwise an increase in the load may prevent the motor from starting and leave no other recourse than the installation of a large motor.

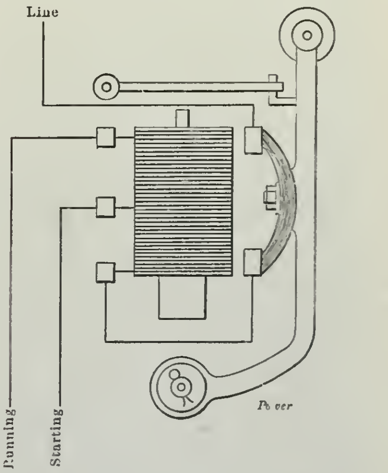


Fig. 2. Starting Tap on Circuit-breaker

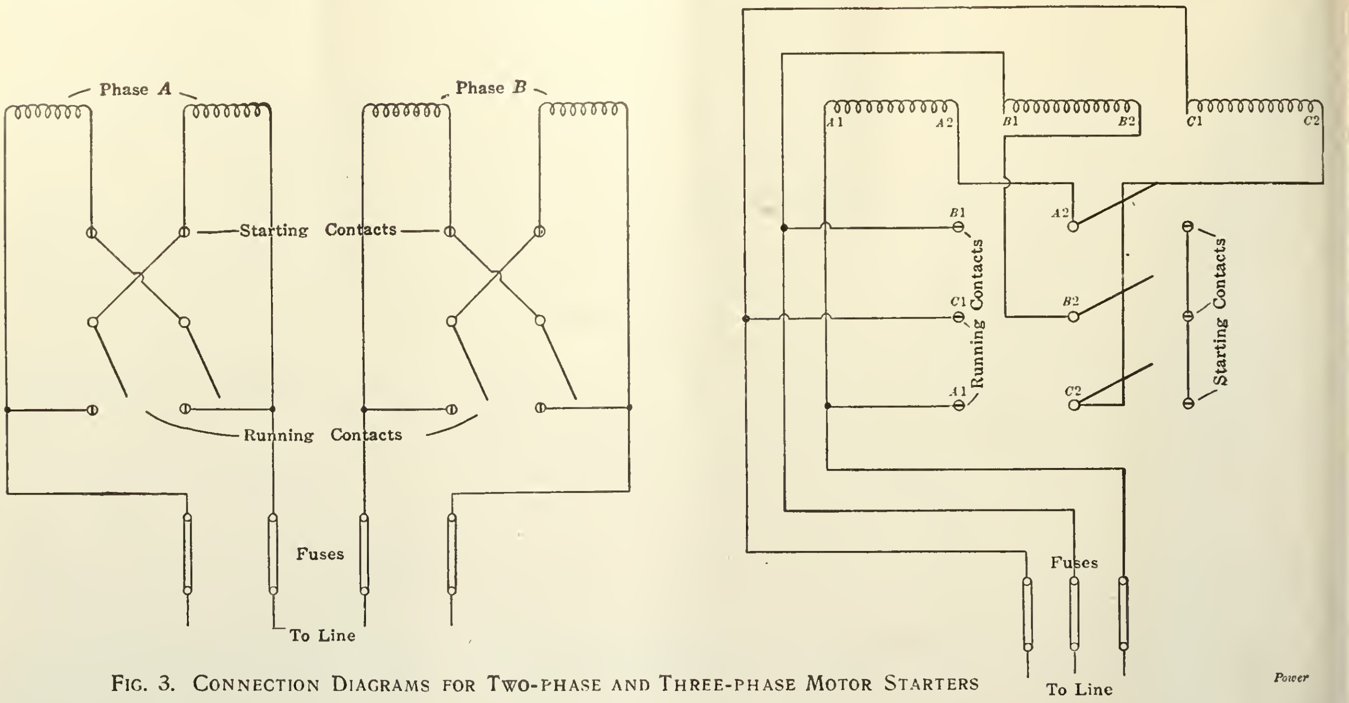


Fig. 3. Connection Diagrams for Two-phase and Three-phase Motor Starters

Resistor Starters

The third method has been the subject of much argument in the technical periodicals. Its advantages are light weight, simplicity and a greater number of starting points than any other method possesses, resulting in more gradual starting and uniform acceleration of the load. Fig. 4 illustrates the Allen-Bradley resistance starter which has recently been brought out. This starter utilizes the property possessed by graphite or carbon of widely varying its electrical resistance under different degrees of pressure. Columns of graphite disks inclosed in an enameled-iron tube are gradually compressed by the starting lever until their resistance is negligible and a slight further movement of the lever short-circuits the columns and connects the motor directly to the line. As the available resistance variation is about one hundred to one and there are no "steps," a very sensitive control of the motor current is possible during the starting period. This feature is particularly valuable when large belts are being put on or where for any reason it is desirable to move the motor through a definite fraction of a turn. The principal disadvantage of the resistance starter is the amount of energy wasted during the starting period, but this disadvantage is more a theoretical than a practical one, as a little consideration will show. The power wasted in a resistance starter is in watts or useful power, whereas the current taken by an autotransformer is largely wattless; therefore, although the power in watts taken by a resistance starter is much greater than that taken by an autotransformer, the current in amperes is nearly the same, and the line disturbance is practically equal in both cases. As the amount of power wasted while starting is negligible in all ordinary cases, the resistance starter is not objectionable on that score and in many cases it gives satisfaction where no other type of starter would.

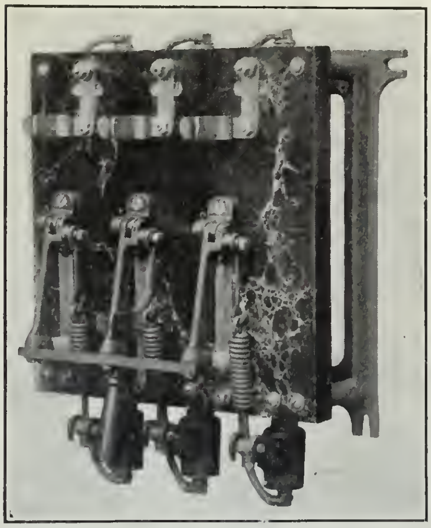


Fig. 4. Starting Switch with Graphite Resistors

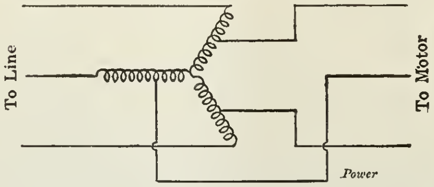


Fig. 5. Three-phase Starter with Three Autotransformers

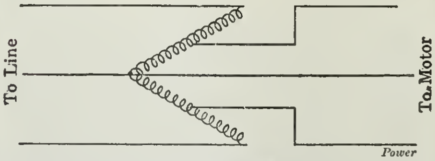


Fig. 6. Three-phase Starter with Two Autotransformers

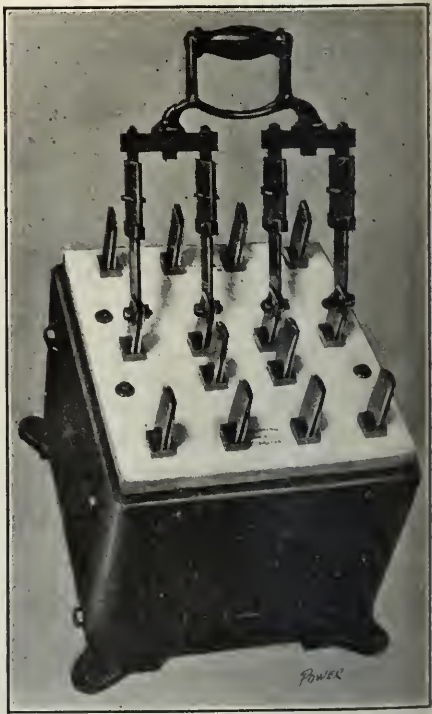


Fig. 7. Old Style Autotransformer Starter

Autotransformer Starters

The fourth method is most extensively used at present. Most of the autotransformers now built for starting motors of 50 horsepower and less have but one starting point. Several taps are brought out from the autotransformer, however, so that several starting voltages are available. These taps are accessible by taking off the starter cover, and by trying successive connections the voltage best adapted to the case in hand can be determined. Care must be used that the transformers are connected alike on all phases. For two-phase work two autotransformers are used. For three-phase work either two or three may be used, as desired.

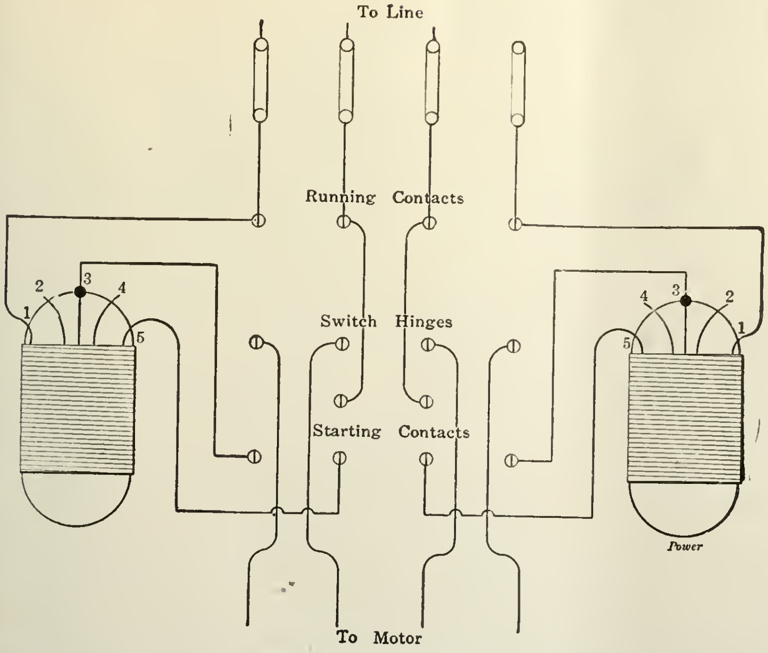


Fig. 8. Connections of Two-phase Autotransformer Motor Starter

The standard practice of the General Electric Company is to use three autotransformers for all three-phase starters up to 50 horsepower, connected as in Fig. 5. The Westinghouse Company, on the other hand, uses but two autotransformers for both two-phase and three-phase motors (Fig. 6), a standard two-phase starter being connected in V or open delta for three-phase motors. The first starter of the autotransformer type that came into extensive use is the form shown in Fig. 7. This has long since been superseded by autotransformers equipped with oil-break switches, which are now furnished by all motor builders; but as many thousands of the old starters are still in use, the internal connections of this form are shown (Fig. 8) for the benefit of those who are still operating this type of apparatus.

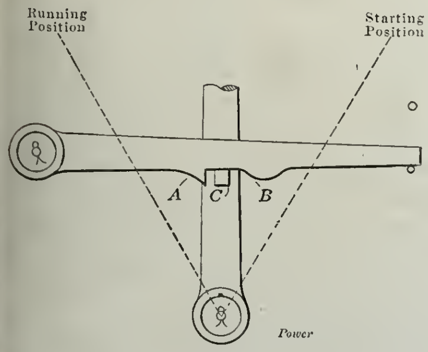


Fig. 10. Safety Latch

Most of the types of autotransformer starter now manufactured are so arranged that the operating handle may be moved from the running to the "off" position without passing through the starting position, thereby reducing the sparking at the switch contacts. The simplest method of accomplishing this is by locating the "off" position in the center of travel and the starting and running positions one on each side. This method is open to the objection that a careless attendant can throw the handle directly to the running position, with consequent danger of injury to the motor and its connected shafting, etc. On the General Electric starter, Fig. 9, there is an ingenious device to prevent this. A notched locking bar, Fig. 10, rests upon a projection attached to the operating lever of the starter. The square shoulder at *A* prevents the lever from moving toward the running position. The rounded projection at *B*, however, offers no obstacle to the movement of the lever toward the starting position. Once the motor is up to speed, the operating lever may be thrown to the running position by a quick movement, because the projection *C* on the operating lever will, in passing under *B*, throw the notched bar upward, and the operating lever will have passed under the shoulder *A* before the bar falls again. If the operator attempts to move the lever slowly from the starting position to the running position, the shoulder *A* will prevent it from passing the "off" position and a new start will be necessary.

The internal connections of a two-phase starter of this type are shown in Fig. 11. It will be noted that an extra set of fuses are cut into circuit when the handle is moved to the running position. When this type of starter is used on a two-phase three-wire system, great care must be exercised in selecting the proper pairs of leads to be connected together for the common return. If this is not done there is great danger of a short-circuit being formed and consequent destruction of the starter. As previously stated, a different starter for three-phase motors is made, but for emergency use the two-phase starter connected for three-wire operation may be used with three-phase motors. In this case, however, the running fuses will all be of the same capacity, whereas for two-phase three-wire circuits the common-return fuse must be 40 per cent.

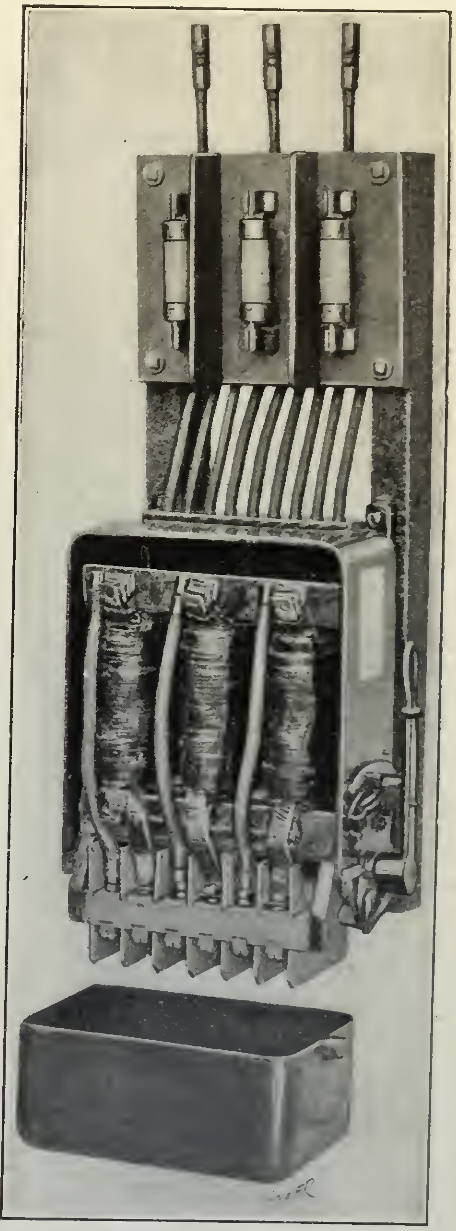


Fig. 9. Modern Three-phase Starter

larger than each of the other two. In Fig. 11, the proper leads to connect for three-wire service are designated *LX*,

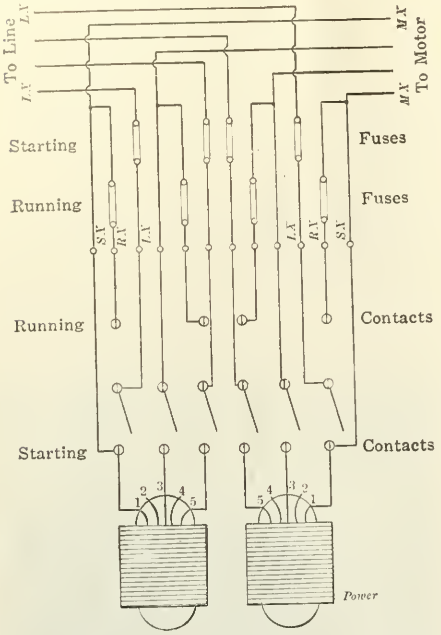


FIG. 11. CONNECTIONS FOR STARTING AND RUNNING FUSES

SX and *RX*, meaning "line," "starting" and "running," respectively.

The starter shown in Fig. 12 is the

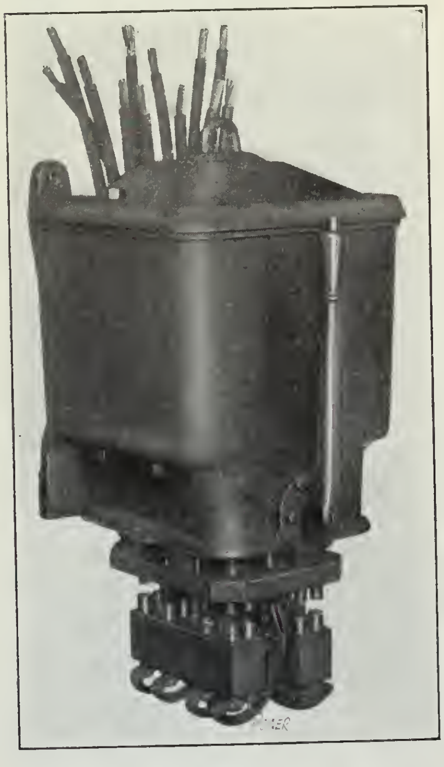
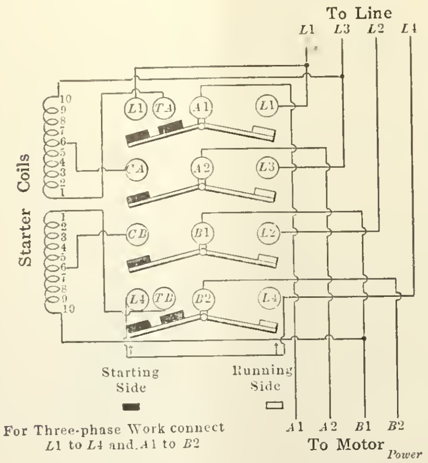


FIG. 12. TWO-PHASE STARTER WITH BUTT SWITCH

latest type produced by the Westinghouse Company. The "off" position is in the center of the lever travel and in addition to a device similar to Fig. 10 to prevent movement to the running position first, there is attached to the operating lever a spring tending to return it to the "off" position. The lever is held in the running position by a latch but, there being no latch in the starting position, there is no chance for the motor to be left running with the lever in the starting position. Spring-operated butt contacts of copper and brass, shown near the bottom of Fig. 12, are used in this starter instead of the familiar drum and finger contacts, and the makers claim greatly increased life of contacts due to their use. The internal connections of this starter are shown in Fig. 13.



For Three-phase Work connect L1 to L4 and A1 to B2

FIG. 13. CIRCUIT DIAGRAM FOR FIG. 12

Generators with Commutating Poles

BY ALVIRTIS S. SALKELD

Commutating poles are generally used on direct-current machines designed to carry a widely variable load. They furnish the necessary flux to neutralize the effect of armature reaction, which would otherwise shift the electrical neutral point between no load and full load. This flux also assists the reversal of the current in the coil during commutation.

A certain number of ampere-turns is required on the intermediate poles to secure proper commutation, and in order to make it possible to get exact excitation, a shunt is provided across the terminals of the interpole winding. This shunt is usually a noninductive strip of german silver, which is unsuitable for rapid load changes. For instance, suppose the load suddenly increased to full load or more; the heavy current, instead of going through the commutating field winding, would be forced through the noninductive german-silver shunt, on account of the inductance of the field winding, allowing the electrical neutral to shift and cause bad commutation. To prevent this, an inductive shunt should be used on all machines of 200 kilowatts or over, and with a voltage of 500 or more. An inductive shunt consists of insulated copper strip wound around a laminated iron core divided into two parts so that the inductance of the shunt may be changed by separating the two core members more or less.

As a test is required to determine the amount of current to be shunted from the commutating-pole winding, it is best to give the inductive shunt a greater inductance than that of the commutating pole, a low resistance and an ample current-carrying capacity. If any extra resistance is needed, a german-silver strip may be connected in series with the inductive shunt, and by changing the length of the german-silver strip the total shunt resistance necessary to give the desired commutation through the range of load for which the machine was designed may be obtained.

When satisfactory commutation has been obtained, try throwing the load on and off suddenly. If violent sparking or flashing occurs, the airgap of the inductive shunt should be readjusted. If an ammeter connected in the shunt circuit shows that the current quickly falls to zero when the load on the machine is suddenly thrown off, there is too little inductance in the shunt, and the airgap should be decreased.

LETTERS

The Sweeper Should Have Been Promoted

It happened at one of the power plants in a coal-mining town. Two alternating-current generators were running in parallel to supply power to operate the mining machinery. The alternators were driven each by a horizontal cross-compound engine. One of the low-pressure cylinders became disabled, making it necessary to stop the engine, and the engineer hastily shut the throttle without disconnecting the generators at the switchboard. The generator of the disabled engine kept revolving, of course, taking current as a motor from the busbars. The engineer sent his assistant to shut off the valve in the steam line in the boiler room, but the engine would not stop. Then the engineer sent for some bars and timber, intending to use them to brake the flywheel in order to stop the engine and generator. An Italian, who was employed as sweeper in the engine room, was watching all the time but said nothing until he saw the men about to "brake" the engine; then he touched the engineer on the arm and said: "Mister, pulle de switch, pulle de switch." This corrected the trouble.

The chief engineer has moved.

W. H. KELLER.

Charlestown, W. Va.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Willans & Robinson's Two Stroke Cycle Engine

We illustrate herewith a design of internal-combustion engine, working on the two-stroke cycle, which has been evolved by the well known English firm of Willans & Robinson, pioneers in high-speed steam-engine design. The piston acts as a valve with respect to the inlet ports, as usual, but it is supplemented by a concentric sleeve reciprocating between the piston and the cylinder wall. This sleeve is perforated by two sets of ports which at the end of the working stroke coincide with the inlet and exhaust ports in the cylinder. The sleeve is actuated by means of eccentrics on the crank shaft. The piston works within the sleeve and the length of the travel imparted to the sleeve is about half the length of the piston stroke. The stroke of the sleeve and piston are in phase, or nearly so; that is, the sleeve and piston arrive at the ends of their downward strokes and similarly at the ends of their upward strokes at nearly the same time; preferably the eccentric actuating the sleeve is set 15 degrees in advance of the piston crank, so that the exhaust ports are opened just before the opening of the inlet ports.

Referring to Figs. 1 and 2, which show sections taken at right angles to each other with the various parts in different positions, *A* is the cylinder in which a sleeve *B* is reciprocated by means of eccentrics *C* on the crank shaft *D*, a rod *E* connecting each eccentric to the sleeve. The piston *F* reciprocates within the sleeve *B*. The cylinder head is made with a cylindrical projection *G* which extends within the sleeve *B* and is provided with packing rings. In the sleeve *B* are two rings of ports, one ring *H* toward the inner and one ring *I* toward the outer end, and there are in the cylinder wall corresponding ports *J* and *K* which coincide with the ports *H* and *I* at the end of the working stroke. At the end of the upward stroke, the outer ring *I* of ports has passed above the packing rings of the projection *G* and the packing rings of the piston *F* have passed above the inner ring *H* of ports in the sleeve, as shown in Fig. 2.

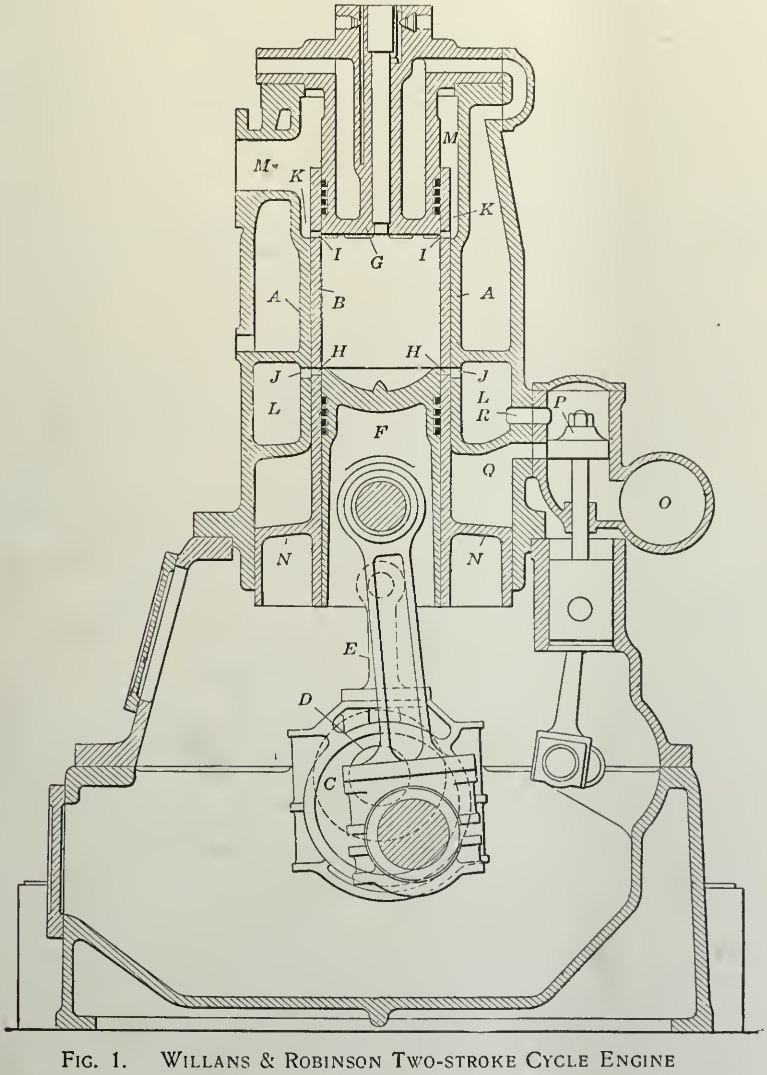


FIG. 1. WILLANS & ROBINSON TWO-STROKE CYCLE ENGINE

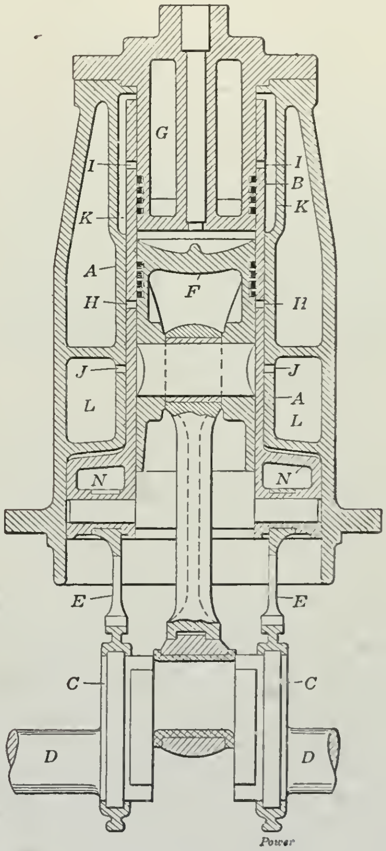


FIG. 2.

The explosion having taken place, the piston *F* and the sleeve *B* move downward, and toward the end of the stroke the outer ring *I* of ports in the sleeve passes beyond the end of the cylindrical projection *G* just before the piston *F* passes the inner ring *H* of ports in the sleeve, as shown in Fig. 1; at the end of the stroke the ports *H* are uncovered, and the ports *J*, leading to an annular receiver chamber *L*, are opposite the inner ports *H*. The ports *I* in the upper part of the sleeve *B* communicate always with an annular passage *K* leading to the chamber *M*, whence the exhaust gases pass out. Scavenging air is blown through the annular space or receiver chamber *L* and through the ports *J* and *H* into the interior of the sleeve, which is really the work space of the cylinder, and the products of combustion are blown by the scavenging air through the ports *I* and the annular passage *K* into the exhaust chamber *M*.

On the return stroke the air contained in the sleeve is compressed as the piston approaches the cylinder head. With an engine of the Diesel type the fuel is injected in the usual way at the commencement of the following or working stroke; with a gas or gasolene engine, the fuel is introduced with the scavenging air at the end of the downward stroke.

For providing the scavenging air an annular piston *N* is formed on the inner end of the sleeve *B*, working in an enlarged part of the cylinder bore. Air enters from the pipe *O* and is admitted to the enlarged part of the cylinder by the piston valve *P*, controlling a port *Q*. On the upward stroke, air is driven by the annular piston *N* through the ports *Q* and *R* into the receiver *L*, the piston valve *P* being then below the port *Q*; and at the end of the next downward stroke of the piston *F*, air from the receiver *L* will be blown through the sleeve for scavenging and charging purposes, as already described.

We are indebted to *The Mechanical Engineer*, of Manchester, Eng., for the illustrations and the foregoing particulars.

The Gas Turbine

By F. R. Low

The gas turbine about which most is known in the United States is perhaps that of Armengaud & Lemale, of Paris. I had the privilege recently of an interview with M. Marcel Armengaud, who, since the decease of his brother René, in 1907, has been the managing director of the Société des Turbomoteurs. He is a consulting engineer and patent attorney with a large practice, a graduate of the University of Zürich, where he studied under Stodola, and was one of the French commissioners to the St. Louis Exposition in 1904. Besides continuing the researches of his brother with regard to the internal-combustion turbine he is devoting attention to aviation.

"As you know," he said, "we attained a considerable success with the turbines which we built, but our little company has not the capital necessary to undertake their manufacture, nor even to continue to experiment upon a large scale. We have, however, furnished two turbines, one of 150 and one of 200 horsepower to different builders of torpedoes for the propulsion of which their extreme compactness and lightness make them particularly adapted. The 150-horsepower size with all its fittings and fuel weighs not more than 100 kilos, less than a pound and a half per horsepower. Its speed is 15,000 revolutions per minute, reduced by gearing to 1500 turns for the shaft."

The turbine is operated by injecting kerosene, air and water into a closed chamber where combustion is maintained continuously. The 150-horsepower machine, when developing its full load in a shop test, used:

$$\left.\begin{array}{l}122\text{ grams of water}\\349\text{ grams of air}\\17\text{ grams of oil}\end{array}\right\}\text{ per second.}$$

This is $\frac{17 \times 3600}{150} = 408$ grams (0.9 pound) of oil per brake horsepower. While this is considerably more than is required by high-compression reciprocating oil engines [Sulzer Brothers guarantee Diesel engines for 180 to 250 grams (6.4 to 9 ounces) per brake horsepower-hour] the turbine offers many advantages in reduced size and weight in continuous high-speed rotary motion, in silent action, absence of exterior moving parts and ease of operation, and Monsieur Armengaud has an abundant faith in its future. One of its advantages for applications such as that to the torpedo, where a supply of compressed air must be carried, is that it can use the air away down to a pressure of seven atmospheres or less while the high-compression engine stops when the pressure is still some 30 atmospheres. The turbine, too, can get along with a smaller quantity of air. The present practice with the Armengaud & Lemale turbine is to maintain the pressure in the combustion chamber at about 70 pounds above the atmosphere and to maintain the temperature of the gas after expansion at about 650 degrees Fahrenheit. More or less water may be used according to conditions, the desideratum being to use only enough to prevent a destructive temperature. The efficiency increases as the supply of water is restricted and the temperature allowed to rise.

The principal incubus of the gas turbine at present is the air compressor. To furnish the large volume of air required under considerable pressure requires a machine more massive than the turbine itself. The recent development of the centrifugal air compressor by Professor Rateau and others points to possibilities in this direction which may result in the reduction of the compressor to the simplicity of the turbine itself and permit its being made integral with it. Monsieur Armengaud has proposed and intends to develop a compressor upon the impulse principle which ought to result in the same reduction in number of stages and thus in weight and bulk that the impulse turbine offers over that of the reaction type and which has led to the very common use of impulse wheels for the initial stages of reaction, or Parsons, turbines, cutting down their length and weight enormously.

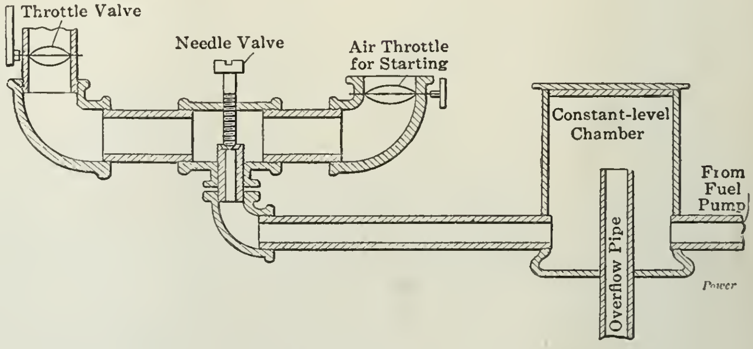
CORRESPONDENCE

A Homemade Carbureter

The accompanying sketch illustrates the construction of a homemade constant-level carbureter for a gasolene engine. The apparatus is made up of pipe fittings. The overflow chamber is a 1½-inch pipe cap; the fuel pipe, overflow and pipe to the needle valve are of ⅛-inch pipe. The "standpipe" is a short nipple with the mouth at the upper end bored out tapering for the point of the needle valve. The suction and main throttle valves are round disks made of sheet brass; the main valve is 11/16-inch diameter, set in a ¾-inch pipe, and the suction throttle is 1 inch, in a ¾-inch elbow. All of the material did not cost more than half a dollar. The carbureter was made by a youngster who got his idea for the design from the U. S. Geological Survey Bulletin No. 277.

James E. Noble.

Toronto, Can.



A Homemade Carbureter

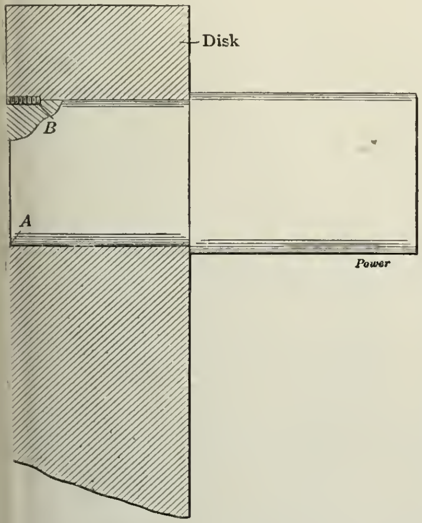
Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

A Loose Crank Pin

One evening when our engine was carrying about one-half load, the crank pin suddenly loosened and turned in the disk, breaking off the centrifugal oiler. Before the engine could be stopped the pin had worked out nearly ½ inch. As the engine was needed at once, the pin was repaired, as shown in Fig. 1, and was run until a new pin arrived. A hole was drilled and tapped out ⅜ inch at *B* and a plug screwed in to prevent the pin from urning. The crank disk was then countersunk at *A*, and the pin peened over to prevent it from coming out.

When the new pin arrived it was inserted in the following manner: The old



ANNER OF SECURING CRANK PIN IN DISK

n being driven out, the hole was cali-red and found to be true and the new n was put in a lathe and turned down til it was 0.005 inch larger than the le in the crank (0.001 inch being alwed for each inch of diameter of the n). The crank was then put on the ter center and a brick forge built under and heated with a charcoal fire, which s forced by compressed air until the nk was hot enough to melt a piece of der put into the crank-pin hole. At s heat the pin was driven in square h a few blows from a sledge and when d it was peened at *A*.

After a pin has been entered it must quickly forced into place, as it is ble to expand and stick in the hole, ich would mean drilling it out. If the e in the crank is not round, it should bored out, for if the pin is inserted and hole is not true, the former will not ain tight.

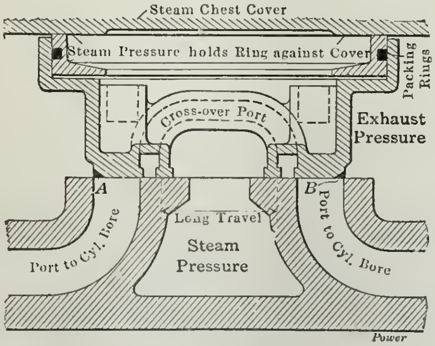
After this experience, I have seen to hat all crank pins were peened at *A* to prevent their coming out in case they should work loose. Generally when a crank pin works loose, it will cause a pound and give warning to the engineer, but this engine ran quietly until the pin began to turn in the disk.

R. L. MOSSMAN.

Tampa, Fla.

Curing a Knocking Slide Valve

A horizontal center-crank automatic engine, 19x18 inches, designed for 100 pounds steam pressure at the throttle, direct-connected to a 150-kilowatt 250-volt generator, gave the owners considerable trouble on account of knocking in the steam chest. The owners were so afraid to operate the engine that they were considering whether or not to shut it down until the trouble could be corrected. The chief engineer and two other engineers who were called in recommended a new valve in the steam chest, but I found upon examination that a new valve would not have corrected the trouble.



SECTION OF VALVES

It seems that when the engine was purchased there was no feed-water heater in the exhaust line and the owners did not express any intention to install one. The engine was built, therefore, to give the best possible economy with a free exhaust to atmosphere. Later, an open-type feed-water heater was installed and some additional ells were put in the exhaust line from the engine. Then when the engine ran "light," the steam valve lifted off the seat near the end of every stroke of the piston, and reseated again with a clack as soon as the pressure under the valve was reduced.

The valve was removed and ⅛ inch filed off at the exhaust edges, as indicated at *A* and *B* in the accompanying cross-sectional sketch, and there has been no more knocking in the steam chest since then.

W. H. KELLER.

Ansted, W. Va.

Failure of Boiler under Test

The soundness of the Massachusetts boiler rules and the necessity that existed for their enactment were recently demonstrated at the Atlantic Works, East Boston, where a boiler that did not comply with the requirements of the rules was, at the request of the owner, subjected to a hydrostatic test to ascertain the pressure required for its destruction.

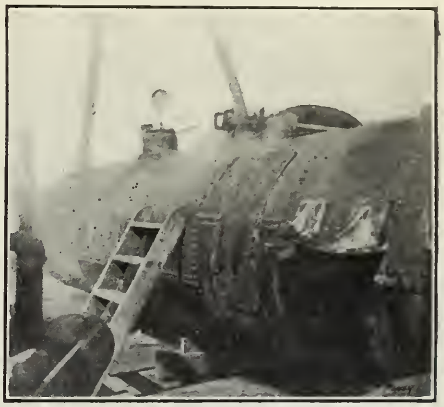


FIG. 1. BOILER JUST BEFORE FAILURE

The boiler was thirty years old and had been in service up to about three years ago, at which time the allowable pressure was 65 pounds. The specifications were as follows: Type, horizontal tubular; length, 19 feet 6 inches; diameter, 66 inches; thickness of shell plates, ⅜ inch; thickness of heads, ½ inch; type of joint, double-riveted lap seam. Pitch of rivets, 2 1/16 inches; tensile strength of plate, 60,000 pounds per square inch. The shell was made up of seven courses and was patched in several places, and in addition to the prohibited lap joint, the boiler embodied a number of features which are considered bad practice in Massachusetts. The nozzle and manhole ring were of cast iron, the manhole cover being bolted down to the outside of the flanged ring; the blowoff was located in the front head about 3 inches above the bottom of the boiler; and there was only one handhole, that being located in the front head. Also, the fusible plug was placed in the third sheet on the side of the boiler.

The test was conducted under the supervision of chief inspector McNiel, assisted by district inspectors Evans and Luck, and resulted in distorting the third sheet to such an extent, when the pressure reached 350 pounds, that the cast-iron nozzle was unable to withstand the strain. Parts of the nozzle and the test gage were blown several feet in the air, falling into the harbor. The sheet cracked each side of the nozzle opening, the cracks being about 3 inches long and having a somewhat crystallized appearance. The heads and tubes remained intact, showing no appreciable effect from the pressure.

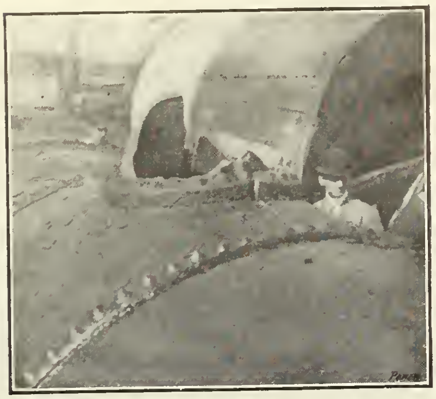


Fig. 2. Broken Nozzle and Cracks in Sheet

C. F. Adams.

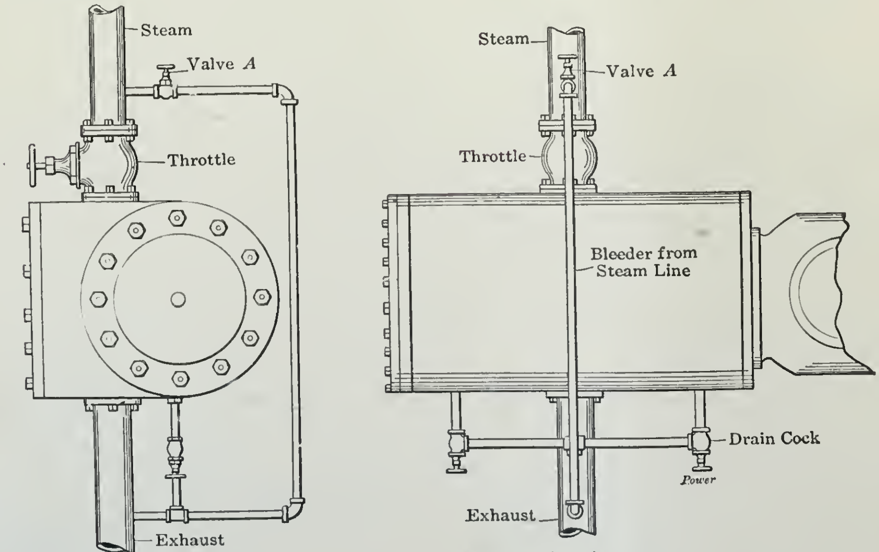
Lynn, Mass.

What Causes the Engine to Run?

We have a simple high-speed engine direct connected to a dynamo. The engine has an indirect balanced slide valve and the bleeders are connected as shown in the accompanying sketch. When the valve *A* on the bleeder is opened the engine starts and continues to run in the right direction as long as the valve is open. Will someone explain how this causes the engine to run?

E. R. Teer.

Anderson, Ind.



End View

Side View

Cylinder Showing Bleeder Connections

Trouble with Metallic Packing

I am in charge of a 100-horsepower, 12x12-inch, high-speed engine, which was packed with metallic packing when I took charge, over a year ago. The packing, which was a soft metal in small grains, containing considerable graphite, was put in sacks and placed in the packing box with a copper wire around the piston inserted before it and a ring of fiber packing put in afterward.

The packing gave no trouble during all last year and the engine ran nicely, but after being idle for about three months this summer and started again this fall, the packing began to leak and was tightened only slightly, because, if it had been tightened much the piston rod would have become hot. This only partially stopped the leak but as there was no more packing on hand, we just let the engine run with the small leak. In a day or so we began to notice a knocking inside the cylinder which sounded as if the piston were hitting the head. This kept getting worse until upon stopping the engine and taking off the head we found a small handful of the packing in the back end of the cylinder. This was taken out and the head put back into place, after which we expected to see the engine run all right again; but not so. At times it would knock quite hard, then run smoothly for a while, then begin the knocking again. Finally it became noticeable that while the knocking was going on, the leakage through the packing was greater. Hence we stopped the engine and prepared t take out the old packing and put in new Another handful of the packing was taken out of the back end of the cylinder, this being mashed in the ring grooves which hold the rings in place, and much of it was stuck fast to the piston. The piston and rings were cleaned and new packing put in and the engine ran al right.

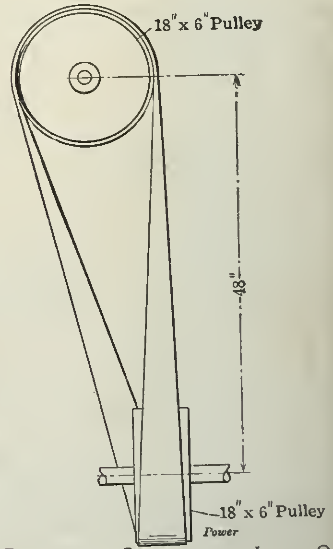
I believe metallic packing of this kind should not be tightened after it is once set. If it begins to leak badly, new packing should be put in.

E. V. Chapman.

Decatur, Ill.

A Quarter Turn Drive

Recently I was assigned the work o laying out and constructing a drive for swing saw, a circular saw, a boring mil and a nigger head, to be used by a contracting firm which was erecting a reinforced-concrete building. The manner i which the contractor desired the machinery placed necessitated a quarter-tur drive. A "mule stand" was out of th question on account of its cost, the lac of space, and the fact that the drive wa wanted for three months' use only. we made a layout of a direct quarte turn drive, as shown in the accompanyi sketch.



Drive as Originally Laid Out

When the drawings were completed, was found that the distance between ce ters of the 18x6-inch pulleys was just feet. On the drawing this did not lo small, but when I turned the drawi over to the master mechanic for approv he studied it for a while, and after a vising me not to have a high belt spe finally approved the layout, and we beg to erect it.

When the construction was nearly co pleted the distance between cent ooked so insignificantly small that I was n doubt as to whether the drive would run satisfactorily, so I looked up various hand-ooks on the subject and found only one vhich contained any definite information; his advised a minimum angle of 25 de-grees for the belt, or a distance of twice he diameter of the largest pulley, which, n this case, would have been 36 inches; ut we already had 48 inches. However, t gave no information relative to the ize of belts or belt speeds. Ours was 6-inch single belt running at 1600 feet er minute. Upon showing this informa-ion to the master mechanic, he decided to isk it, but advised the use of a 4-inch louble belt instead of 6-inch single belt. Jpon going out to the job, however, I ound the contractor skeptical; he did not ike the idea of running the risk. So he rdered that the shed be raised and a ew layout substituted.

When I went back to inform the master echanic of the changes that had been rdered by the contractor, I was unable o get in touch with him, and the next orning he noticed the changes being ade. I was called in and was censured or allowing the contractor to alter our riginal plans, as it made us look like a ot of amateurs in the eyes of the con-racting firm.

My object in writing this is to ascer-ain first, what is the minimum allowable istance between centers of pulleys for quarter-turn drive of this sort; second, hat is the maximum belt width in ches; third, what is the maximum al-owable belt speed?

WALTER J. BITTERLICH.

Watertown, Mass.

eeding Boilers through the Blowoff Connections

An incident recently came to my atten-on which illustrated what a resource-l engineer can do to prevent a shut-own. It was at the plant of a large dustrial concern, where the boiler-room quipment consisted of four 150-horse-ower return-tubular boilers, which were d by an ordinary duplex pump, dis-harging through a manifold into the dif-rent feed lines entering the boilers.

About the middle of the forenoon one y, the fireman accidentally twisted one the valve stems from its seat, leaving e valve, which was of the gate type, ide open. This brought about an awk-ard condition of affairs as there was no eans of regulating the water level in at boiler and it could not be spared ng enough to be cut out of service for placing the broken feed valve. Water se rapidly in this boiler while the water vel in the other boilers was so low at it was not possible to stop the pump.

Fortunately, the chief was a resource-l man and did not permit himself to come excited. A shutdown was next impossible; even the loss of one boiler would practically necessitate it. Water was already too high in the one boiler and too low in the others. Therefore, he disconnected the main line of the blowoff pipe and put a globe valve on it, which he closed. The blowoffs from all the boilers were connected with this line, and by opening all of them all the boilers were connected together through the medium of the main blowoff line. The fire under the boiler with the broken feed valve was then forced considerably harder than the rest, and no more difficulty was experienced in maintaining the proper water level as water was easily forced into the other boilers.

Of course, the needed repairs were made when the plant was shut down for the night. It might be stated that the plug cocks in the feed lines between the check valves and the boilers were located on top in a practically inaccessible position for such a purpose as regulating the water supply.

EDWARD T. BINNS.

Philadelphia, Penn.

Metal for Cylinders

The question arises from time to time as to the most suitable metal for steam cylinders, and it appears that, although foundrymen agree on a certain grade, they do not always produce a metal which wears as well as desired. It is agreed that cylinders should be of hard, close-grained gray iron, but in some cases, although the grain appears fine, more lubrication is required than with others of apparently the same grade. As a general thing, the cylinder which requires considerable lubrication does not acquire the high polish necessary to economy in operation. It is not uncommon for different cylinders on the same engine to vary in grade; therefore, their length of service, before reboring is necessary, also varies. It may be answered that there are other elements influencing wear, not due entirely to the composition of the metal, but experience has shown that the principal factor is the metal.

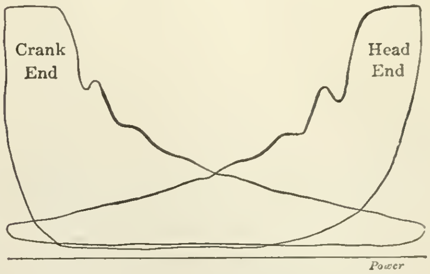
This leads to a consideration of a suitable metal for "bull rings" of pistons, and as to whether they should be of harder or softer composition than the cylinder. Some urge the use of a metal containing a small percentage of manganese; but what is the percentage? Can such a metal be machined readily and will it take a high polish? Others advocate the use of a mixture of steel and iron in casting; and still others claim that charcoal iron is the best. This would indicate that no fixed standard is used, but as every engineer has to renew pistons it is desirable that he should know and be able to specify the composition of the metal most suitable for his purpose. As an instance of this a certain foundry cast a set of rings for a large engine; the rings wore about 1/16 inch a week, and were replaced by rings of apparently the same composition and cast by the same foundry, which have been in service for four years without any perceptible wear.

W. E. RECTOR.

Brooklyn, N. Y.

Diagrams for Criticism

Will some reader of POWER kindly point out what the jagged expansion line in the accompanying diagrams indicate? I am reasonably sure that there was no trouble with the indicator and think the trouble must be in the engine.



DIAGRAMS SHOWING JAGGED EXPANSION LINE

H. T. FRYANT.

Memphis, Tenn.

A Problem in Hot Water Heating

In the building of which I have charge, there is a hot-water heater located at the extreme end of the building, resulting in long horizontal runs before the water reaches the risers. Consequently, I have experienced much difficulty in obtaining a good circulation in the lines farthest from the heater.

I suggested to a heating engineer that if a pump were placed on the circulating line where it enters the hot-water tank a better circulation would result. But he claimed that this would create a vacuum in the branch lines near the top of the system, due to the fact that the pump would increase the pressure above that which is carried on the house tank, this being a closed tank in the basement of the building, and carrying 90 pounds pressure. He contended that the pressure must be reduced below that of the house tank before the cold water would enter. On this point I agree with him, but I do not believe that the pump would create a vacuum before that would occur.

The proper place to install a pump, according to his idea, is on the lines leading to the house tank from the hot-water tank.

I would appreciate opinions regarding this problem.

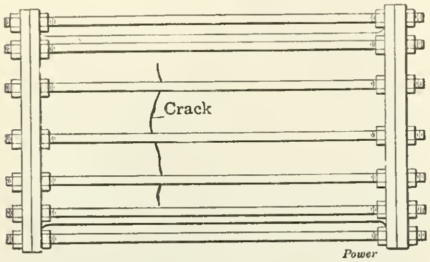
G. L. LYONS.

New York City.

Questions Before the House

Cracked Cylinder

In response to Mr. Harrison's inquiries in the September 20 issue, I herewith indicate how the cylinder of a 30x66-inch engine was once repaired. The steam pressure used in this engine was 185 pounds.



Through Bolts on Cylinder

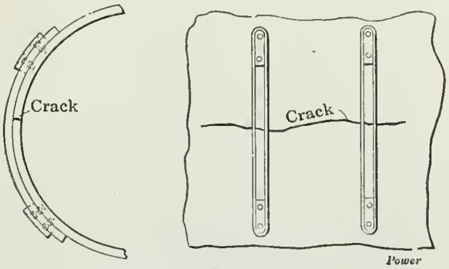
Long bolts were passed through the holes in the cylinder flanges, as shown in the accompanying figure.

H. R. Blessing.

Philadelphia, Penn.

One way in which to repair a cracked cylinder is as follows:

Fit a piece of boiler plate carefully over the crack by heating the plate and hammering it to the proper shape. Cut an asbestos gasket for the patch and then bolt the patch in place with tap bolts. The patch should be 4 inches wide so as to allow room for the bolts on either side of the crack. Space the bolts about 2 inches apart. The boiler plate used for the patch should be about ½ inch thick.



Arrangement of Links and Blocks

Another way in which to repair the crack is illustrated in the accompanying figure. Blocks of wrought iron or steel 3x1x4 inches in size and having rounded ends, are tap bolted to the side of the cylinder, as shown. Links of ¾-inch square iron are then fitted to the blocks and shrunk on. The links must be made of proper size so that they will grip the blocks tightly when shrunk into position. The holding bolts should be ¾ inch in size.

E. F. Tracy.

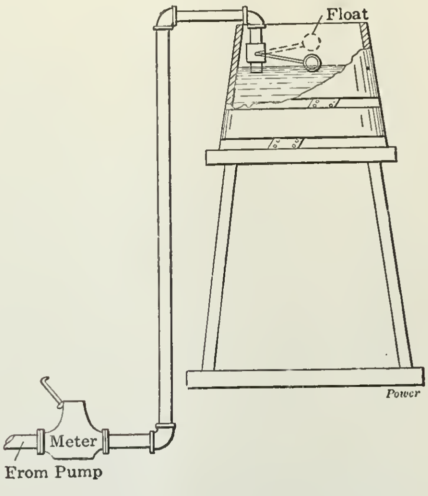
Mechanicsville, N. Y.

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Accuracy of Water Meters

The variation in accuracy of the disk type of water meter is receiving some attention at the present time in the columns of Power. It is quite well known that a disk water meter is just as capable of being inaccurate as is an electric meter. However, all in all, the disk meter is a fairly reliable instrument.

When a meter is small in comparison with the amount of water it is required to handle, it is very prone to read high. The disk meter is designed to work somewhat on the same principle as the plunger meter by measuring off small portions of the flow. I have found the disk meter to be quite reliable under moderate working conditions. But when it is overloaded, an overregister is almost sure to be made. While the plunger meter is quite accurate, at the same time it will not work under all conditions. I recall a case where one of the best known plunger meters was abandoned and a disk meter installed. The water had a peculiar chemical effect upon the linings of the plunger meter and it repeatedly failed to operate. A disk meter that is handling a large volume of water in proportion to its rating, when located in a main which is directly connected to the water-works pump, as is often the case, will work ahead on account of the action of the pump.



Arrangement of Tank and Meter

A case of this kind is shown in the accompanying figure. A large meter is connected into a large main carrying, say one-eighth of the volume of water which the pump will handle. The main delivers the water into a tank controlled by a float valve of the balance type. This meter does not have a steady continuous motion. The lower the float is the more the meter will register in excess of the quantity that is actually flowing. As the float approaches the position shown by the dotted lines, the meter becomes more accurate. This shows that when the water is flowing with the least amount of resistance the meter registers more than is passing. This and many similar experiences with the same type of meter makes it conclusive to me that a disk meter should be large in proportion to the amount of work done. If there are any vibrations in the line due to the working of the pump plungers it is a good plan to close down the delivery beyond the meter so as to afford some back pressure upon the disk of the meter and thus steady the movement of the disk.

C. R. McGahey.

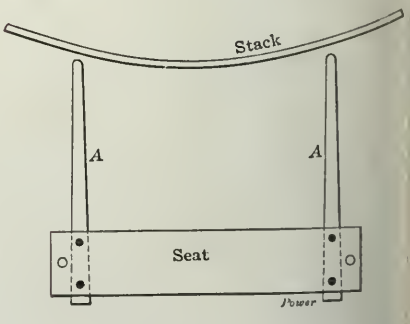
Sheffield, Ala.

Painting Smokestacks

The article on the above subject in the August 16 issue is very interesting, and should prove helpful to anyone who has stacks to paint.

The *modus operandi* of a practical steeple-jack painter who includes this district in his circuit is so simple and effective that his ideas are worth passing on.

Instead of two sets of blocks and tackle as described in Mr. Hawkin's article referred to above, this painter requires bu one, and a handline of ¼-inch rope. Th set of blocks, single and double, are rov with about 200 feet of ⅜-inch rope, an the handline is about 60 to 75 feet i length.



Construction of Seat

To attach the tackle to the first guy-wire band the painter gets as high as possible by means of a ladder placed on the roof, then he uses a few lengths of ¼-inch pipe to hoist the double block and its hook up to its position on the lug of the guy-wire band. A little U-shaped hook stuck into the top end of the ¼-inch pipe makes it easy to hook on the main hook, and also to detach the pipe rod.

Without any more preliminaries, when once this upper block is securely hooked in place, he is all ready to begin painting; for with his method, he paints *up* one side of the stack and *down* the other. There is thus no time or energy lost in initial ascents as in Mr. Hawkin's method.

Also, there is no need for a helper, as this painter has a device whereby he hoists himself easily and safely. It consists of two eccentric grips somewhat similar to wire stretchers; one is fastened to a breast belt, and the other to a pole-climber's leg iron on the painter's foot. The loose end of the fall passes through hese grips. When he wishes to hoist himself a little, the upper grip holds him while he loosens the bottom one and aises his foot. Then the bottom one is ightened in the raised position, and by oosening the top one and standing up he seat is raised.

Although this takes some time to decribe, it takes very little time to oprate, and certainly works nicely.

The swing seat attached to the single lock is fitted with a couple of guide ieces of 1x4-inch wooden strips, as hown at *A* in the accompanying figure. his prevents the seat from hanging too ose to the stack, and also eliminates ie trouble of turning around as menoned in the article just referred to. A ook in the bottom of the seat carries a x- or eight-gallon pail containing the aint.

Having attached his handline to the pe rod, the painter ties the other end to e swing seat so that when he has ainted up to the guy-wire band, he can ull the rod up and transfer the double ock again to either the top of the stack the next guy-wire band, as the case ay be. Of course, while making the ange, he has the swing seat secured to e stack by means of a spare hook.

In painting, he uses a long-handled hoe ush similar to the one described and own by Mr. Hawkins; and by reaching both right and left, one-half of the ack circumference can be painted from e side unless the stack is larger than e average size. As soon as all of e surface within reach is painted, the ush is stuck in the pail, and the painter nply hoists himself 4 to 6 feet and gets sy again.

When the top of the stack is reached, e-half of the painting is done. By ans of the spare hook he shifts himf and tackle around the rim of the ck to the opposite side, and then paints his way in easy stages down to the point where he changed his tackle support on the way up. Then, as before, he changes the double block by the use of the pipe rod which he left tied at this point of the ascent.

When the bottom is reached and the tackle unhooked, the stack is finished. The usual time taken on a stack, say 4 feet in diameter and 80 to 100 feet high, is about 2½ or 3 hours. This time includes getting the outfit up on the roof, etc., and getting the tackle in position.

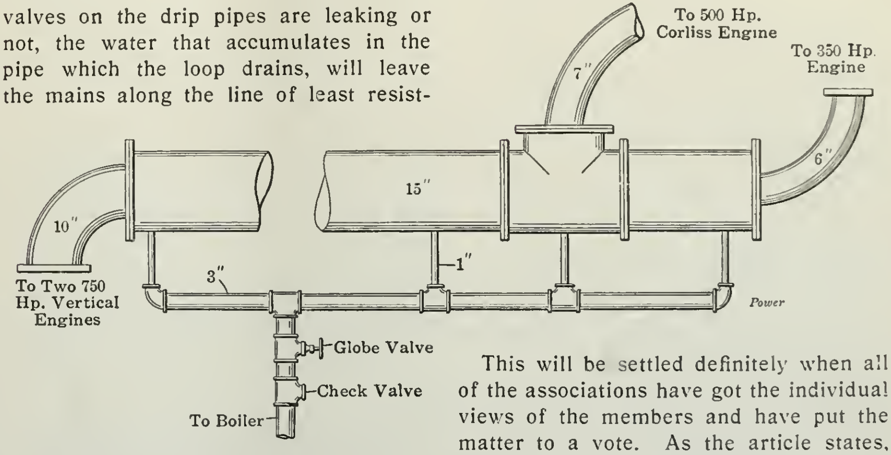
J. A. Carruthers.

Bankhead, Alberta, Can.

Trouble with a Gravity Loop

In the August 30 issue of Power, Thomas Sheehan describes his trouble with a gravity loop. I wonder if he realizes that the cause he gives for the trouble, namely, leaking check valves in the drips from the steam main allowing the engines to pull the water of condensation back out of the receiver when the engines were carrying an overload, is an impossible one?

There are two ways in which this trouble could occur. First, the loop could be balanced. When this occurs the water of condensation does not "loop the loop" but stands still and, whether the check valves on the drip pipes are leaking or not, the water that accumulates in the pipe which the loop drains, will leave the mains along the line of least resistance as it condenses, and do no harm if not trapped in some pocket. And let me say right here that Mr. Sheehan's steam main is well adapted to hold the water back until the slugs get large enough to wreck an engine or two.



Revised Piping Arrangement

Second, Mr. Sheehan's steam main is not large enough for a plant of 2350 horsepower and, as peak load means overload, conditions get worse, and with overloaded engines on one side and boilers working up to their capacity, or a little above, on the other, one can very well see a good excuse for the gravity loop being overloaded by the boilers priming. And conditions could be still further complicated by losing the vacuum on one of the engines or by the steam pressure dropping.

Does Mr. Sheehan know that if he took the pressure out of his system, connected a good pump on at *A* (see Mr. Sheehan's sketch), filled his receiver full of water until it touched the check valves, then removed the check from the valve and started the pump, he could not pull the water out of his receiver?

Here is a suggestion for Mr. Sheehan. Put up a good 15-inch steam drum in place of the pipe of many sizes. Separate the 10- and the 7-inch engine pipes by connecting the 10-inch at the end of the drum, as per the accompanying figure. Drain the drum, not by a steam loop but directly into his boilers at the opposite end from where the steam leaves the boilers. He does not have to extend the drip down inside the boiler to the water. Then the main will be well drained and the steam loop will not be overloaded. Whether he can save the power of an engine with this arrangement I cannot say.

Gilbert Guerin.

Adams, Mass.

National Conventions

In the issue of September 27 is an editorial headed "The Rochester Convention," and ending with the words, "Is it worth the price?"

This will be settled definitely when all of the associations have got the individual views of the members and have put the matter to a vote. As the article states, the national convention is a costly affair for the associations as the *per capita* tax levied draws money from some who can ill afford to pay. The tax could be used to better advantage by buying books which are slightly beyond the reach of many engineers and the educational benefit derived from these would be 100 per cent. greater than that of listening to a delegate's report of the convention, which usually consists of an account of the addresses of welcome; winding up with a statement of who were elected as officers. This may be highly entertaining but it is not one whit educational. The only educational benefit which the delegate gets is at the various booths by looking at the exhibits. And the only educational advantage which the members of an association get is the literature which the delegate brings back with him. The delegate, nine times out of ten, is unable to recall how some little thing or other works, when asked. Decidedly, an annual national convention is not worth the price.

Would it not be better for the various exhibitors to get together and give demonstrations in the different towns? They could send out circulars stating that there would be a demonstration (naming the various articles) on such a day and that it would be of benefit if the recipients or their representatives witnessed the demonstration. The exhibition could also be given in the evening when the engineer could attend, thereby giving greater publicity to the products.

Speaking individually, the national convention, held annually, is not a good thing. Once every three years is sufficient. The officers should be men capaable of performing their duties for that length of time.

On the other hand, I think that the annual State convention is a good thing. The expense is not so great and the conventions are generally held at some large-sized city and the country associations' delegates are there able to discuss matters of mutual local interest and see some of the plants and become acquainted with the engineers who are in charge. Then, on his arrival home, a delegate is able to tell the members of his association about things that he saw that were of benefit.

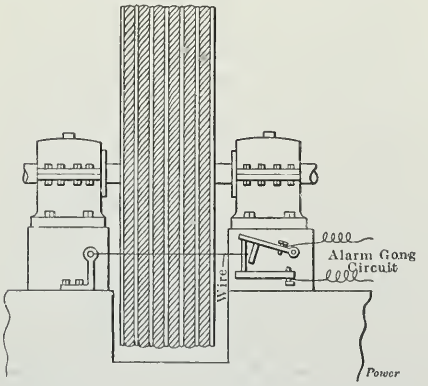
Again, the national convention is not able to recommend the passage of any law to individual States, while such recommendations can be made by a State convention.

George H. Handley.

Newburgh, N. Y.

Safety Device for Rope Drive

In the September 6 issue, Edward Moran described the details and gave a sketch of a safety device for rope transmission. His scheme is good, although I believe that the one which I have is simpler and just as effective.



Arrangement of Safety Device

The accompanying figure shows my arrangement. A wire is stretched across the face of the pulley. At one end it is attached to a fixed post and at the other to a piece of wood which is wedged between the contact blocks of an alarm-bell circuit. When a strand breaks, it hits the wire which results in the wood piece being pulled out of the grip of contact blocks. The upper block then falls onto the lower one, the circuit is closed and the alarm sounds.

E. A. Heiny.

Springfield, Ill.

Engineers' Wages

In the September 13 issue, C. R. McGahey states, "Wages count for nothing." This statement is obviously untrue.

Again, Mr. McGahey states, "Some men are short-sighted enough to say, let us have a union." He then suggests, "An honest board of examiners, which cannot be bought, to find out what a so called engineer does know; find out if he is a $4 man or a $2 man and classify him." Is this not a form of union? Is it not asking the State to enter into a union with you to protect you from your less efficient fellow craftsman? Does it not show a desire to be legislated into a position free from strife, or where strife is reduced to the minimum; where you can rest serene in the knowledge that you are protected and that you need not endanger your own precious being? Does it not show a desire to get the better wage by holding down your fellow craftsman, rather than by the more natural way of lifting him up?

G. H. Kimball, in the September 20 issue says, "In any plant running 24 hours, the matter to be disposed of is only the question of the length of the watches." If you decide on the length of the watches and pay no wages you will find your agreement as to the length of watches will not last long. It is hard to understand how a man is so lacking in self-interest as to make such a statement.

Mr. Kimball again says: "Many men (employers) think that because an engineer does not ask for more pay he does not want it. If they knew how some men feel about this they would realize that a raise gained by asking is considered as given grudgingly." For the purpose of this discussion it is useless to refer to, or have in mind, special cases. In general then, I will state that a raise is always given grudgingly, whether it is asked for or not. The reason the employer pays what he does is because he is not sure that the employee would work for less. If he knew he could reduce his pay roll $50 or $1000 by cutting Tom, Dick and Harry, or by making a general reduction without demoralizing his force, he would do it at once; and this, regardless of the education or efficiency of the employees. This he would do because he is human and an employee would do the same for the same reasons if positions were reversed.

T. M. Sterling, in the September 6 issue, says, "It must be admitted that the wage question is about the first, middle and the last consideration, even though all the other items pertaining to the deal are satisfactory." In this statement Mr. Sterling is absolutely right. He knows what he is working for. If he has a family, he knows he is working for wages to enable him to provide for it. He does not feel that he should apologize to his employer for collecting his wages.

Mr. Sterling then suggests a union for education by means of which the members could be given a rating. We seem to be afflicted with a desire to educate, in proportion to our desire to eliminate ourselves from the wage question. Have we not the best schools and colleges in the world? Our technical schools are the equal of any. If we cannot attend these, we have the best correspondence schools, giving good practical courses. We can secure special books written on any subject in which we may be interested. We have technical papers by the score to enlighten us on modern practice.

We should realize that we are not in the educational field as educators; that is the province of people who are specially trained for the work and who devote their whole time to it. We should not have a morbid desire to cram our education into unwilling subjects. This educational "bug" is too evident with us.

Another quotation from Mr. McGahey, "Consider some of the National Association of Stationary Engineers' meetings; lots of good can come from these, but often one gets the floor who knows less than anyone and the posted man goes away in disgust." This intellectual monstrosity, this self-assumed fountain of knowledge, "the posted man," is anywhere and everywhere that men come together for discussion. He is common to every calling. He makes the most noise in his effort to gain the spotlight and when he succeeds in making himself thoroughly obnoxious, he goes his way bemoaning the stupid ignorance of the rabble. His is not a desire to educate. The applause for him. He is a pest.

This repeatedly expressed desire to be classified, rated or graded is advocated by those who assume that they would automatically become members of class A on the law or rule being adopted; and that the great majority would be relegated to the rear, or degraded and that class A would then be in a position to profit, since competition would be removed. This is the motive. It is useless to deny it.

That you may be disillusioned I ask you to investigate the conditions that obtain in those States where engineers are classified by license laws. There you will find the members of class A as well as B and C working for a lower average wage than the engineers who are not classified.

Through this whole discussion there is

evidence of a hope to attain the ideal. There is everywhere the cry to classify, grade and to eliminate the inefficient, the beer-guzzling, the "bums" and "grafters."

It is very good to have a high ideal, very good to have a subconscious reverie or day dream. It leads man to a purer state. Through evolution he is approaching his ideal; but he is so many centuries in the rear that the practical man is dealing with conditions as they are and trusting to nature and evolution to lay in the lap of posterity the subject of his dreams.

In dealing, then, with conditions as they are and for results during your lifetime, you will grant that it is hopeless to spend time trying to eliminate the inefficient, the beer-guzzling (or the idealist). They are factors in steam engineering as well as in the law, the ministry and every other line of human endeavor.

H. F. Heyrodt expresses my sentiments in the last paragraph of his article in the October 4 issue.

J. J. Nash.

New Haven, Conn.

Do It Right or Quit

In the issue of September 20, there is an editorial under the above title. The title itself is a worthy motto for every engineer. But, unforunately, how few follow such a motto? Yet the engineer is the one who should do things right at all times.

I find that it is just as easy to do a thing right as to do it only half right, and I have always held that anything which is worth doing at all, is worth doing well. Nobody takes pride in what he does unless he is positive that he has done his best.

It has already been said that life's necessities are very easy to get in this country as compared to the older countries of the world. This means that a man in England has to hustle more for what he gets than a man in this country. And, no doubt, this has something to do with this matter of doing things half way.

A job half done would have been better left undone; because it will surely have to be done over again. And it nearly always happens that the job will give way when there is the least time for repair. How much time and labor would have been saved if the work had been well done at first? It is the faculty of nearly all Americans to take things easy. What if something does break down? A fellow has got to work in either case, so what is the difference?

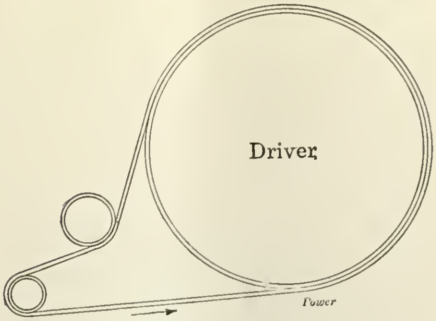
Let us follow the motto of the editorial alluded to, and we will become much better engineers.

J. E. Pochè.

Plaquemine, La.

Useful Belt Drives

In the September 13 issue of Power are shown sketches of belt drives that are said to be useful. Their usefulness is not apparent, although I would not, for that reason, say that they are not useful. The friction load must be considerably increased by their use, although there may be advantages that will overbalance the loss. The drive illustrated herewith has proved its right to be called useful, as by its use we have been able to save over 1500 pounds of coal per day on a daily consumption of six tons.



An Efficient Belt Drive

The driver is 12 feet in diameter and the distance between the centers is about 10 feet.

R. McLaren.

Berlin, Ont.

License Laws

The subject of engineers' license laws has been discussed for many years and from about every angle conceivable. In the issue for September 27, Mr. Nigh gives his opinion of such laws in a manner that would not tend toward the passage of such laws by any legislative body, if his views were accepted.

He acknowledges that if the law tends toward public safety and the making of more competent engineers, the sponsors are public-spirited citizens. As a properly constructed law does tend to safeguard the public and does force the engineer to better equip himself in knowledge pertaining to steam engineering, it would seem that the writer has acknowledged the benefit of such a law. As to the wisdom of forcing the applicant for a license to pay a certain fee, there is a question. It can be only considered when it is the belief that the law is solely for the benefit of the engineer. If it was of no benefit to the engineer and was simply for the welfare of the public the engineer would not stand for it, without a vigorous protest. As engineers have been benefited where suitable laws are in effect, they do not find it a hardship to pay the usual small fee demanded

If no one would accept the position of an engineer, unless he had been trained for the place, and if, no employer would employ an engineer until he had found out his qualifications other than the price he would work for, there might be an excuse for saying an engineers' license law is not a necessity. As the majority of us know that the above is not the case, then it seems to me that laws are a necessity to compel public safety.

No engineers' license law is complete unless it carries with it a clause governing the construction and operation of steam boilers. This particular feature is of the utmost importance to engineers, as they should be the ones most interested in the safety of their boilers. Admitting the ability of an engineer to determine the safety of the boilers under his charge, we must remember that the owner is the one who must pay for repairs and new installations. When he refuses to consider what his engineer may have to say in relation to his plant, is it not wise legislation that compels him to make proper repairs, or renew unsuitable boilers?

Where the owner is willing to take the advice of his engineer and is willing to purchase the best that the boilermaker can produce, it may not be necessary to compel him to comply with a law. However, as such conditions are not too frequent, the engineer, for his own as well as the public's safety, should labor for suitable laws governing the construction and operation of steam boilers.

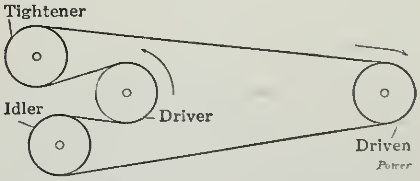
All laws are the outcome of the failure of man to consider his fellow man. If the golden rule was actually carried out by all, the majority of our laws would be of little use. Unfortunately greed is the gravest fault of civilization and the desire of employers to get all they can causes many of them to purchase the cheapest, whether it be steam boilers or the labor to operate them.

T. N. Kelsey.

Lowell, Mass.

Trouble with the Belt

I would suggest to Mr. Hamill, who inquires for a solution of his belt trouble in the September 27 issue, that he run the belt straight instead of crossing it. In order to get a reverse motion it will be necessary to arrange the drive as shown in the accompanying figure. If his pulleys are in line, he will have no further trouble due to the belt running off. With the arrangement shown herewith, the belt should last twice as long as it would with the other layout.



Arrangement of Pulleys to Secure Reverse Motion

S. McRae.

Fulton, Ala.

Progress in Flue Gas Analysis*

By G. H. Kelsay

In the past year the progress made in flue gas analysis has not been as rapid as might be expected. The committee's report is presented in abstract together with the discussion at the meeting and written discussion sent to Power.

As this is the third year this subject has been before the committee, it has been of interest to watch the progress made by gas analysis in boiler-room practice.

Like all other methods that promise to be beneficial in developing higher economies, gas analysis and the automatic recording analyzer were taken up and heralded by many engineers as promising in a way to revolutionize boiler-room economics, all of which has been exceedingly beneficial, as it has been the means of gradually developing among operating engineers an understanding of furnace conditions and some of the most fundamental principles in economical coal consumption.

Inquiries were sent out to all known users of gas analyzers and automatic recorders to obtain as near as possible just what development has been made during the past year on this subject. In addition to inquiries, personal letters were directed to those whom it was thought had done the most during the past year along the line of experimental work, with the idea to determine as nearly as possible what progress has been made.

Replies were rather disappointing in that very few new ideas or expressions were received, which indicated that flue-gas analysis has not made such radical strides as might be expected.

As was shown in previous reports of this committee, there still seems to be considerable trouble in getting successful operation from recording apparatus, as most all the recorders now in the hands of operating engineers seem to give considerable trouble and lack a degree of accuracy, such as to discourage interest by the operating engineer and the boiler-room force. This may be due partially to a lack of diligence on the part of operating engineers and probably this objection is gradually being overcome.

There are different opinions among operating engineers as to the importance of continuous records and intermittent records. Some believe in occasional analysis of the flue gases by means of the Orsat analyzer, so as to establish the conditions that give best economy, and then endeavor to keep the firing up to these conditions. Others believe that continuous and recorded analysis of the gases is necessary for best results.

Two methods of obtaining continuous analysis are used. One is the use of the sampling bottle which collects the average gas for a day's run, and the other is by means of the automatic recorder which intermittently or constantly records the CO_2 at the time the gas is being made at the furnace. There has been some very good evidence presented against the sampling vessel, in that it does not indicate to the engineer the time in which he is getting good or poor results in his furnace. It only gives the average for the day and the fireman is unable to judge at what times he is getting best results, whereas he can watch the continuous recorder and determine what condition gives the highest per cent. of CO_2 in his gases. In power stations where the load is a fluctuating one some seem to have difficulty in obtaining good CO_2 records.

It has been the opinion of some that the percentage of CO_2 could be taken as practically a direct indication of boiler efficiency. That is, a high percentage of CO_2 always meant a high equivalent of evaporation, and low percentage of CO_2 meant a low equivalent of evaporation. However, it has been proved quite essential to occasionally analyze the gases for CO, so as to guard against the likelihood of excess CO, as it has been quite conclusively demonstrated that the highest percentage of CO_2 does not mean the highest boiler efficiency and that a very high percentage of CO_2 may be obtained with very low boiler efficiency or a very low equivalent of evaporation. This may be on account of throttling the air supply on the furnace to such a very low point that there is not sufficient to burn the coal, thereby producing such a high percentage of CO in the flue gases as to reduce the equivalent of evaporation, and at the same time give a very high percentage of CO_2.

Gas analysis as applied to boiler-room practice will, no doubt, gradually adjust itself into methods that will be very beneficial in developing and maintaining higher boiler economies, but it is possible that many overdrawn ideas have been entertained by many engineers as to its value. Also many engineers have failed to give the subject the attention that it deserves and would probably find more merit in the analysis of flue gases than they now think.

It is the practice with a few operating engineers to pay firemen a fixed sum with a bonus of variable amounts, depending on the per cent. of CO_2 obtained from their respective boilers while they are on duty. The following is a sample of what one company is doing in this respect:

When products of combustion show CO_2—

	Extra per Day. Cents.
15 per cent. for 10 hours......	30
14 per cent. for 10 hours......	25
13 per cent. for 10 hours......	20
12 per cent. for 10 hours......	15
11 per cent. for 10 hours......	11
10 per cent. for 10 hours......	10
9 per cent. for 10 hours......	9
8 per cent. for 10 hours......	8
7 per cent. for 10 hours......	5
6 per cent. for 10 hours......	0

It is the argument of the engineer that the extra amount paid the fireman is a small percentage of the amount the fireman is saving the company through his efforts to obtain the increase in his wages. This method is not generally practised by operating engineers, and it appears to be somewhat doubtful as to whether a bonus should be paid firemen on the basis of CO_2 when there are other important elements entering into the proposition. Particularly so when extremely high CO_2 fails to necessarily indicate the most economical furnace operation and when it may be possible for a fireman to obtain very high percentage of CO_2 with very low equivalent of evaporation, because it may be possible for him to obtain these results by throttling the air supply on the boiler to such an extent as to make the work of the boiler very light and his work very light. Thus he might be obtaining the highest percentage for doing the least amount of work and obtaining the poorest efficiency in his furnaces.

Discussion

H. G. Stott, vice-chairman of the committee, presented the report, which in addition to the material upon flue-gas analysis here given, and that upon forced draft in our issue of October 25, treated of Low-Pressure Turbines, Steam Meter and Peak Loads. The other subject will be presented later. The paper wa unfortunate in coming before the meeting just previous to adjournment, compelling the continuation of its discussion until the next morning, when its encroachment on the business assigned fo that session impelled the chairman t cut the discussion off long before th subject was exhausted and while ther were several with good things yet to say We shall be glad to offer in the column of Power an opportunity for the continuance of the discussion and preser in this number discussion which has bee sent to this office for that purpose.

Mr. Stott remarked in his presentatio that "With bituminous coal and force draft there is no difficulty in getting 225 per cent. rating with furnaces as ordinarily built. Better evaporations have been obtained with forced draft than with natural draft. The explanation seems to be that with forced draft the combustion takes place on the surface of the coal. The flames are shorter and do not shoot up into the tubes and become extinguished. In other words, the combustion is complete before the gas has reached the chilling surface. With natural draft the pressure inside the setting is less than that of the atmosphere and the infiltration of air is a very serious matter."

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*Portion of the report of the Committee on Power Generation of the American Street and Interurban Railway Engineering Association, Atlantic City, October 10-14.

B. F. Wood, of the Pennsylvania Railroad, said that boilers were capable of doing much more than was now expected of them, and a suggestion of the possibilities might be taken from locomotives burning from 100 to 150 pounds of coal per square foot of grate area per hour. Some form of stoker that would effect rapid firing was essential for stationary boilers.

G. H. Kelsay, the author of the paper, said he did not wish to convey the impression that he recommended paying the fireman on the basis of CO_2 in the flue gas, although he knew of some plants that paid a bonus on the CO_2 record.

A member whose name was not announced said that he had looked very carefully into the subject of CO_2 analysis and had visited a number of plants where CO_2 recorders had been installed, invariably finding them out of working condition.

Charles Hewitt, of Philadelphia, invited the last speaker to visit his plant where he had two flue-gas analyzers in almost continuous operation for three years and now giving very satisfactory service.

C. O. Mailloux said that the days of tall chimneys were fast passing away. The proper way to supply air to a boiler was by a combination of suction and pressure. He had built, not the tallest, but the longest chimney in existence, a tunnel over 2000 feet long, having been driven to convey the gases to a point where the 100-foot vertical stack would be inoffensive. He brought out strikingly the fact that the change of temperature between the furnace gases and the boiler-heating surface takes place within an infinitesimal surface, less than one-thousandth of an inch, in which there is a fall of temperature of some 1500 degrees. The problem of handling forced-draft systems efficiently in railway-power service comes back to getting firemen to fire well at the varying rates which are required by the varying loads. It is much easier just to meet the demand for steam when the load is steady than when it varies widely, and that is one reason why the CO_2 recorder is more valuable in the station carrying a lighting load than in a railway station. In the effort to maintain a constant steam pressure the usual practice is to open the damper when the pressure begins to fall and put on more coal. Very often this results in holes in the fire which reduce the efficiency and do not increase the output. If the pressure gets too high, the first thing which is done is to close the damper, which gives an insufficient supply of air and the hot coal in the furnace goes on producing CO. Under practical operating conditions the percentage of CO_2 is far from being a true index of boiler efficiency, taken without due consideration of other factors; but efficiency after all is a secondary consideration in boiler operations, subordinate always to continuity of service. Improper control of the damper is responsible for more loss of efficiency than any other detail of operation of boilers. Automatic damper-controlled devices are not satisfactory. The best method is to have a system of signals connecting the boiler room with the switchboard, so that the firemen can control the dampers under instructions from the switchboard operator who sees the load fluctuations. The boiler room is entitled to receive as much attention in this respect as the engine room. In the boiler room, 90 per cent. of the energy is converted, and any percentage of saving which can be effected in that conversion is much more important than the same percentage of saving upon the 10 per cent. conversion which takes place in the engine room. Damper control should not be regulated solely with reference to the CO_2 in the flue gases. The first thing which the damper should control is the steam pressure, and the second essential in their manipulation is to control the volume of steam produced in the boiler. Because a satisfactory steam-flue meter has not yet been developed, it is practically impossible to determine accurately the amount of steam which a boiler is making. The quantity of feed water is not an adequate index, because of a number of boilers fed with the same weighed or metered feed water one may be doing much more than the others. Mr. Stott had told him of instances in which two boilers of 1500 horsepower each had been found to be operating at very different rates of production, one giving 500 horsepower and the other 2000.

The discussion was closed by William Roberts, of Akron, O., who said that progress was being made in the development of reliable and accurate flue-gas analyzers, and spoke reassuringly and hopefully of their usefulness in connection with steam-boiler plants.

Discussion Sent in to Power

Lewis Sanders, vice-president of the Sarco Fuel Saving and Engineering Company, contributes the following written discussion:

We are quite in agreement with Mr. Roehl's statement* regarding the high hopes originally raised by automatic CO_2 recorders and the failure to realize on these hopes. These hopes were based on a great deal of erroneous and misleading statement made on the subject of recorders and the control of furnace operations by means of CO_2 records. One of the prime causes in failing to realize anticipations was the fact that very little stress was laid on the amount of care and attention necessary to maintain high boiler-room efficiency by means of CO_2 control. Consequently, recorders were frequently installed without provision being made for following up their records and enforcing efficiency on the part of the firemen. It was evidently expected that a plant could be operated with less attention by the use of a CO_2 recorder than without, and when this was found not to be the case, the use of the instrument was often condemned. The case is, in fact, exactly the opposite; it takes more attention to run a plant with the aid of a CO_2 recorder than without, and the results repay the effort. It also takes more brains and supervision to operate a compound-condensing engine than a simple slide-valve engine. It should be borne in mind that running a CO_2 recorder and getting records does not constitute using it.

Mr. Roehl's point regarding the decrease in thermal-transmitting ability of the heating surfaces with decrease of gas velocity over it, does not take in all the factors involved in the efficiency obtained with high CO_2. There is another factor and that is the temperature, which rises with high CO_2 and, therefore, increases the heat-absorbing capacity of the surfaces, probably considerably more than it is decreased by the lowering of the velocity of the gases. The loss of efficiency with excessively high CO_2 is not, therefore, explained by decrease of gas velocity.

The entire explanation of the falling off in economy with high CO_2 is covered in Mr. Roehl's second point; that is, the formation of CO. The point at which CO will form varies with different coals and furnaces. We always advise that with hand-fired furnaces and Eastern coal, running as high as 20 per cent. volatile matter, no higher than 13 per cent. CO_2 be carried, and the furnaces be tested once a month with an Orsat apparatus in order to determine if any deterioration in the baffling is resulting in incomplete combustion and formation of CO. With stokers the CO_2 may with advantage be pushed a little higher and with oil and gas fuels may be pushed up as high as 15 to 16 per cent., but we always urge a careful checking with the Orsat apparatus to detect CO when such high percentages of CO_2 are carried. This is particularly necessary under oil-fired boilers where a breaking down of the checker work would cause the production of CO under these conditions.

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*October 25 number of POWER, page 1913.

It was known long before the Geological Survey began its work that the most economical air supply to operate with in well constructed hand-fired furnaces was about 40 per cent. in excess of the theoretical, and that this was indicated by from 12 to 13 per cent. CO_2 with Eastern coals. Apparently, all this old knowledge was lost sight of in the over-enthusiastic literature issued by the makers of CO_2 recorders and no limit at all was placed on the percentage of CO_2 that it was desirable to obtain. A very clear distinction should be made between the efficiency due to design of apparatus, efficiency due to state in which it is kept and efficiency due to method of operation. It is due to the failure to separate these that much of the very noticeable confusion of ideas on boiler operation is due. With defects of design the operating man has little to do; he generally has to make the best of what he has. The most common defect is probably faulty furnace design which tends to produce considerable CO. Defects of condition in the furnace are poor condition of the baffling and air leaks. These result in the formation of CO and in high stack temperature. In the boiler they consist of dirty fire surfaces and scale, shown by high stack temperature. Such points should be investigated at intervals, say once a month, as conditions change slowly. Although we make recording pyrometers, we do not advise their use on boiler plants, as we consider a mercury thermometer all sufficient to take stack temperatures. Defects of operation comprise faulty firing of coal and adjustment of draft and here conditions change continuously and we therefore need a CO_2 recorder as a guide.

Operated with intelligence and with due regard to the engineering facts in the case, the CO_2 recorder is by all means the most reliable operating guide for firing the furnace.

Embury McLean, president of the Engineer Company, submits the following written discussion:

Mr. Roehl claims that the use of CO_2 recorders has not resulted in the improvement in boiler and furnace efficiency expected. Two reasons are given for this unsatisfactory result:

First, that as the percentage of CO_2 increases (or as the excess air is diminished) the velocity of the gases through the boiler is reduced, and consequently the efficiency of transmission of the heat units from the gas to the boiler is reduced.

Second, that the means employed to obtain high CO_2 usually result in the presence of CO, which more than offsets all the advantage gained from the high CO_2.

It is true that the efficiency of the transmission of heat from the gases to the boiler increases with the velocity of the gases, when the amount of heat absorbed by a unit of surface in a given time is the only element considered. On the other hand, when the total number of heat units contained in the gases passing a given surface is considered, the slower the velocity of the gases passing the surface, the greater the proportion of the heat units transmitted.

The loss incurred in attaining an increase in velocity of the gases by dilution far outweighs any gain thereby. Assuming a fixed rate of combustion, if the velocity be increased by increasing the volume of gas by dilution with excess air, the efficiency is very materially reduced, for the temperature of the furnace varies directly with the percentage of CO_2, or inversely with the percentage of excess air, and the rate of absorption of heat depends upon the difference in temperature between the gases and the boiler. Again, still assuming a fixed rate of combustion, the velocity of travel of the gases can be increased by lengthening the path of their travel by increasing the number of baffles.

By applying the laws enunciated by the United States Geological Survey, the increase in the number of baffles will increase the resistance and, consequently, the "pressure drop" between the grate and the chimney, leaving a proportionately smaller "pressure drop" available for the grate. This, of course, would reduce the output of the boiler.

Should we resort to mechanical draft to maintain the output of the boiler, we find that the weight of gas passing through the boiler varies as the square root of the "pressure drop," consequently, the increase in "pressure drop," due to the increase in baffling, will enormously increase the power required to operate the fan. Of course, under these conditions, an increase in output from the boilers would be inexpedient, as it would be attended by abnormal cost of operation of the fan.

It is true that the presence of high CO_2 does not necessarily mean high furnace efficiency, because if accompanied by CO a great loss is sustained, due to incomplete combustion. Therefore, any manipulation of the furnace which will tend to produce CO, is undesirable. Mr. Roehl, however, refers to the balanced-draft system and states that it unquestionably gives a very high CO_2 result, and that the claims of the sponsers of the balanced-draft system are not exaggerated in this respect.

The balanced-draft system differs from other means of obtaining high CO_2 in the furnace (by skilful manipulation, etc.), in that it is impossible, from the very nature of the system of regulation, to produce CO in any appreciable quantity. The fact that in the wide use of this system in the United States and Europe results obtained universally show a material increase in the percentage of CO_2, with practically no CO, substantiates this statement. This result is accomplished in a very simple way by the use of the very simplest apparatus, entirely practical for boiler-room use.

The control of the furnace in accordance with the percentage of CO_2 in the flue gases depends upon the fact that the temperature of the furnace for a given rate of combustion increases directly with the percentage of CO_2 in the flue gases. Again, for a given rate of combustion the rate of absorption of heat by the boiler, and consequently the steam pressure of the boiler, depends upon the initial temperature of the furnace gases.

If we can accurately and automatically control the supply of air to the furnace by the steam pressure of the boiler, it is evident that a rate of supply can be maintained which will give a maximum percentage of CO_2 in the flue gas. It has been found impractical to accurately control the air supply to the furnace where the air is drawn through the bed of fuel, which of necessity means a pressure less than atmospheric above the grate and in the combustion chamber. Under these conditions, air is drawn into the furnace chamber through the fire door when open, through cracks and crevices, and through porous brickwork.

The air supplied to the furnace by a chimney varies with the thickness of the fuel bed, which is constantly changing, with atmospheric conditions, and with the opening of the fire door and other leakages above the grate. In order to obtain accurate control of the air supply, it is necessary to have some more reliable source. For this purpose, we resort to the fan blower, and with the balanced-draft system a special type of blower is used, which delivers, for a given speed, a constant volume at a variable pressure.

As the combined result of two variables cannot readily be controlled, it is necessary to eliminate one of the variables. For this reason the draft is balanced, which means that atmospheric pressure is automatically maintained in the furnace chamber at the fire door for all rates of combustion. Under these conditions all excess suction of the chimney beyond that required to remove the gases from the boiler is eliminated; the function of the chimney now being merely to remove the gases from the boiler and not to supply air to the furnace.

If the special type of blower is operated by an engine, the speed of which is controlled by an automatic valve which follows the steam very closely and varies the speed of the blower inversely in proportion to the pressure of the boiler, giving extreme speeds on a variation of, say two pounds, and intermediate speeds proportional to the variation between these extremes, the air supply to the furnace will at all times be limited to the amount necessary to maintain the maximum temperature in the furnace, which, as we have stated, corresponds with the maximum percentage of CO_2 in the flue gas.

The only condition imposed to maintain the maximum percentage of CO_2 is that a sufficient supply of fuel, evenly distributed over the grate, be maintained. The balanced-draft system accomplishes the rest automatically. Even if this condition is not perfectly maintained, the loss due to thin spots in the fire is less with this system than under ordinary conditions.

The operation is as follows: Assume the steam pressure a little above normal, the rate of combustion is reduced to a point slightly below that required to maintain steam pressure and for a moment there is a deficient supply of air. The temperature of the furnace drops and the steam pressure begins to lower.

As this happens, the speed of the blower gradually increases until the theoretical rate of air supply has been attained. At this point the maximum furnace temperature is attained which means an increase in steam pressure and a repetition of the same process, which will continue to control the air supply to practically the theoretical amount necessary to maintain the rate of combustion so long as the steam pressure remains within the limits of the adjustment of the valve.

If the steam pressure varies more than this amount, due, for instance, to sudden increase of load on the boiler, or to neglect to keep an even supply of fuel on the fire, the blower will run at its maximum speed supplying an excess of air, but increasing the rate of combustion to a point, where, notwithstanding the excess of air, sufficient heat units are developed to bring the steam pressure back to within the limits of the valve, when the air supply is again reduced to practically the theoretical amount required, and vibrates a little above and a little below this amount, so long as the boiler pressure remains within the limits of the valve.

Under these conditions, it is impossible to obtain any appreciable quantity of CO, as the change in temperature due to the presence of CO will immediately be accompanied by a reduction in temperature which will in turn increase the supply of air until the theoretical rate is attained.

We see, therefore, that balanced draft or the maintenance of atmospheric pressure in the furnace chamber, does not in itself control the air supply to the furnace, but it is an essential element in accomplishing that result, as no amount of refinement of regulation of either the blower or of the chimney can accurately control the air supply as long as the chimney remains a variable factor. The balanced-draft regulator is adjusting the air supply constantly and is working every moment of the time, whereas the fireman, having other duties to attend to, at best cannot devote more than 10 per cent. of his time to this regulation.

M. W. Campau, of the Precision Instrument Company, Detroit, Mich., writes:

In making the statement that "high CO_2 with unchanged combustion reduces gas velocity across the heating surface," we think that Mr. Roehl has failed to reduce his high and low CO_2 gas to the same conditions.

Suppose you have a fixed size of opening (represented by the boiler-flue area), through which you are to pass a quantity of gas. To pass in the same length of time, 150 volumes would have to travel at a higher velocity than 100 volumes. Assuming 100 volumes to represent perfect combustion, with just the necessary amount of air, then something more than 100, say 150 volumes, would represent incomplete combustion with excess air. The 100 volumes of gas would be the hottest because the, say, 150 volumes would be cooled down by the amount of heat required to heat the excess air. The gas in perfect combustion would, of course, represent a high percentage of CO_2, and the imperfect combustion, a low percentage of CO_2. This may be expressed as follows:

100 vol. air	Perfect combustion, 100 vol. air burned	100 vol. gas	high CO_2 hot slow velo'y
150 vol. air	Imperfect combustion, 100 vol. air burned, and 50 not burned	150 vol. gas	lower CO_2 cooler higher velocity

When coal is completely burned in the presence of air, and the exact amount of air required is used, the CO_2 will form the same volume in the waste gases as does oxygen in the air (nitrogen remaining constant, and omitting the negligible quantities of water, gas, SO_2, etc.) This well known reaction is:

$$79\ N_2 + 21\ O_2 + C = 21\ CO_2 + 79\ N_2.$$

In complete combustion the volume of the gas passing through the furnace, and therefore the velocity, is unchanged. Size of opening remaining fixed, the velocity of the gas is only changed by adding air.

Mr. Roehl infers that when the gas is traveling at a high velocity, the heat in the gas is transmitted to the water better than with the gas at a low velocity. We think this is open to argument, but it is quite easy to imagine that in a stream of gas flowing quietly, there would be, in the core or heart of the stream some particles of gas which would not come in contact with the heating surface, which they might do if they were tumbled about, so to speak, in a rapid current. It is always considered advisable in transferring heat from one moving medium to another in any apparatus, to break up the currents or change their direction so as to bring all the particles of the heating medium in contact with those being heated, or the heating surface, so that all the heat may be extracted, and this idea is carried out in many well known devices.

The object to be gained, whether by "high velocity" or otherwise, is the complete extraction of the heat in the gas, and we believe that this may be better accomplished in a boiler setting by breaking up the current with baffles, than by introducing excess air to increase the velocity.

It cannot be denied that perfect combustion means high CO_2, and high temperature at the furnace, as in the 100 volumes assumed above, and it should be the aim of any furnace operator to get high CO_2, or high furnace efficiency, and then reconstruct his setting so as to get high combined efficiency of furnace and boiler. If this were done as above outlined, there would be no "rise of flue temperature with increased CO_2," which is not the fault of the combustion, but the fault of the boiler.

However, let us suppose that it is more advantageous to an installation to run at a low percentage of CO_2. Then it should be run at that per cent., and it is just as important to the man running it to know what his percentage of CO_2 is, whether it is 1 or 15 per cent. Therefore we submit that he should have a CO_2 recorder in any case, and we think he would not be disappointed with the information gained therefrom.

J. A. Caldwell, of the Caldwell Economy Company, New York City, writes:

Speaking of Government statements, I have sometimes thought they may not be infallible. For instance, from the report made by the Naval Liquid Fuel Board, 1904 (69 tests), you will find a summary on pages 250 to 253 covering different kinds of burners in which the average CO_2 is stated at 7 per cent. or a little over, and the five tests at which the maximum evaporation per pound of oil was secured read 7.24, 7.68, 7.70, 8.53 and 7.86 per cent., the average for the five tests being 7.8 per cent.

This does not begin to compare with the results obtained by the Pacific Light and Power Company (Frank T. Clarke), on fuel-oil tests which averaged about 14 per cent. on Babcock & Wilcox boilers, the gases drawn from the bottom of the second pass, and the equivalent evaporation from and at 212 degrees per pound of fuel being 15.35 pounds. It would seem that the two oils could not vary so much as this in the constituent that would affect the CO_2.

E. A. Uehling, of the Uehling Instrument Company, Passaic, N. J., sent in the following discussion:

There are very good reasons why the reports on the results obtained from the majority of the early purchasers of CO_2

recorders should be disappointing. First, the earlier CO_2 recorders were complicated and, therefore, difficult to keep in running order. Second, these instruments were generally placed in the boiler room without adequate protection against the dirt and dust, which increased the difficulty of keeping them in order. Third, they were frequently left at the mercy of ignorant firemen, or at best given in charge of the engineers, all of whom may have been good practical men no doubt, but most of whom were entirely unfamiliar with that class of apparatus, and only too often prejudiced against all "scientific frills"; not infrequently with minds made up beforehand that the thing was no good, and that it would not work anyhow, and as a matter of course, had little difficulty in proving it. Fourth, they were imported and pushed into many places where both personnel and conditions precluded successful operation, by very efficient salesmen, whose interest in the apparatus sold ceased on receipt of pay, thus leaving the recorders as orphans in a foreign and often unfriendly camp, among surroundings making useful existence impossible.

Under these conditions it is not at all surprising "that the hopes that the CO_2 recorders aroused as to a solution of the problem of guidance in the attainment of high efficiency have not been realized."

The disappointing reports received by the committee prove nothing against the value of CO_2 as a guide to efficient firing. The economy directly derivable from a high per cent. in the flue gas has, no doubt, been exaggerated by over-zealous venders of CO_2 recorders. Of the fact that they have been pushed injudiciously there can be no doubt. That many of the earlier instruments were faulty in design and badly constructed must be admitted; that not one in ten of the CO_2 recorders installed received the intelligent care and attention which a scientific instrument should receive cannot be denied. The great majority of them were foredoomed to failure from the beginning.

Today all this is changing for the better. The practical men are becoming more and more familiar with scientific methods and apparatus.

The CO_2 recorders have been improved and simplified. They are more easily handled and require only a moderate amount of attention. Intelligent attention, the sledge-hammer type, is still ruinous.

To say that the higher the CO_2 the greater must be the economy, is absurd. To contend that CO_2 is no guide to efficient firing is even more absurd. For every kind of fuel and construction of furnace there is a per cent. of CO_2 which gives maximum efficiency. It may be as low as 10 or as high as 16 per cent. This best per cent. of CO_2 must be ascertained by practical economy tests, and having been ascertained, the CO_2 recorder will keep you informed how well or how badly the fireman has performed his duty. It is important that the fireman should be able to see what he is getting all the time. An indicator at or near the boiler front is, therefore, of the greatest value.

The committee report contains the further statement:

"More light is at present available on the cause of the disappointment than was available after the CO_2 recorders had been in sufficiently extensive use to make the disappointment evident. This light is, of course, that concerning the relation between gas velocities and thermal transmitting ability of the heating surface."

This statement is not very clear but it seems to convey the idea that for some time after it had been discovered that CO_2 recorders were of little or no use, no good reason for the disappointment existed; but that at present more light is available and that "this light is, of course, that concerning the relation between gas velocities and thermal-transmitting ability of the heating surfaces."

That CO_2 recorders have been disappointing is freely admitted, and we have shown that their failure to make good was caused partly by inherent imperfections, partly because of misplacement by over-zealous venders, but mostly because they did not get a fair chance. But all this has no vital bearing on the fact that the per cent. of CO_2 in flue gas is the most available and best guide to efficient firing.

The velocity of the gas is, no doubt, an important factor, and it is quite possible that the reduction in gas volume due to a high per cent. of CO_2 may appreciably affect the rate of absorption and thus slightly increase the stack temperature and hence the loss; but no sane combustion engineer would advocate admitting an undue excess of air for the purpose of obtaining a high gas velocity.

If high gas velocity is of real importance, as it seems to be, it must be attained by other means than that of diluting the products of combustion, if the gain due to more efficient absorption is not to be overbalanced by heat carried off by the greater volume of gas necessary to obtain the higher velocity.

The per cent. of CO_2 in flue gas is a true measure of the excess of air passing up the chimney; in connection with an occasional sample from the rear of the combustion chamber it gives the amount of air infiltration, a most prolific source of heat loss. In combination with the stack temperature it gives the loss due to the sensible heat carried off by the gases. The continuous record is a perfect check on the fireman. It shows how often he fired, how well or how badly he did it. It shows when he cleaned his fire and how long it took him.

An instrument that will make such a record should certainly be a valuable adjunct to a boiler and be amply worth what little extra attention it may require. If, in addition to the above, it will indicate on the boiler front in plain view of the fireman, showing all the time the condition of his fire, all reasonable requirements are fulfilled.

One word more regarding the payment of a bonus to firemen for attaining a high per cent. of CO_2. It seems ridiculous to pay a bonus for CO_2 as low as 7 per cent. Under anything like good conditions and fuel, 10 per cent. should be the minimum set for a bonus. Whether 15 per cent. is economically attainable depends on kind of fuel and operating as well as furnace conditions. As already stated, the most economical per cent. of CO_2 should be determined for each plant and then striven for.

High CO_2 is not incompatible with low CO and low CO_2 is no guarantee against the presence of CO. Except in the presence of very high CO_2, i.e., a very small excess of air CO is due to irregular and uneven firing. If the firemen receive a bonus for high CO_2, they should be penalized for high CO, and particularly so, if it occurs in connection with low CO_2. For the same per cent. of CO the fuel loss is twice as great when it occurs with 7 per cent. of CO_2 as with 14 per cent.

The subject of CO_2 and CO_2 recorders is a large one and the more intelligent discussion it calls out the better.

To coat iron with a covering of lead which will be in absolute metallic contact, states the *Chemical Trades Journal*, is a difficult operation. Valves and small fittings made of iron well galvanized on the exterior may be homogeneously covered with lead in the work shop by any mechanic. The method consists in immersing the piece to be coated in water to which a few drops of sulphuric acid have been added. Then, while in the acid water, the piece is readily amalgamated in the usual way by squeezing mercury through close-woven cloth all over its surface and thoroughly rubbing it in. The excess mercury is rubbed off and the piece carefully dried without heat, and then immersed in a bath of lead, which should be well above its melting point, so that it would not tend to solidify by introduction of the cold piece. The casting may be withdrawn after about twenty seconds, and will be found to be homogeneously covered with lead. This method requires the piece to be galvanized before applying the lead, and the galvanizing must be in good condition, otherwise the subsequent amalgamation will be imperfect and present an unsightly appearance.

Seven Killed in Boiler Explosion

By A. D. Blake

A climax water-tube boiler exploded at the works of the American Manufacturing Company in Brooklyn, killing seven persons and injuring many. The primary cause of the explosion was corrosion of the bottom head of the central drum.

On Thursday morning, October 20, there occurred the most disastrous boiler explosion that has been seen in the vicinity of New York for many years. The accident happened at the works of the American Manufacturing Company, in the Greenpoint district of Brooklyn, and resulted in seven deaths, among which vas that of the chief engineer, and sevral persons were injured. At the time of ne explosion the plant had been running or about three-quarters of an hour and everything was supposed to be operating under normal conditions.

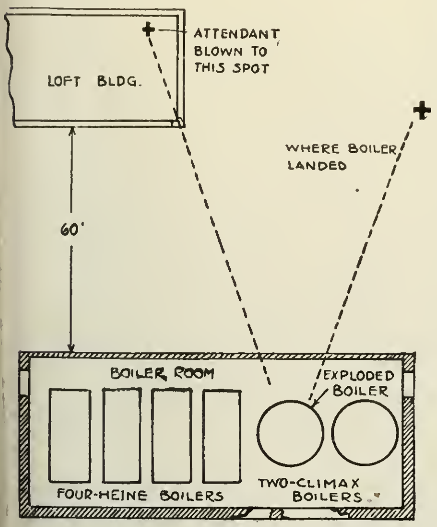

Fig. 1. Plan of Boiler Room

The boiler which exploded was of the Climax water-tube type, operating under 150 pounds steam pressure. In the same boiler room there was another boiler similar to the one which exploded and four Heine boilers. Without any warning the boiler shot straight up through the roof like a skyrocket and landed in a vacant lot about 50 yards distant, the impact being so great upon landing that a hole several feet deep was dug into the earth. Although the whole side of the boiler room was torn out, strange to say, the other boilers were not seriously damaged. It was fortunate that the boiler after being hurled into the air landed as it did in the vacant lot, for if it had turned and fallen in the other direction it would have crashed through the roof of the factory, in which case the loss of life would undoubtedly have been great. An idea of the force of the explosion may be gained from the fact that one of the firemen was hurled through the side of the boiler room, across a 60-foot street, through the side of a wooden building and into a loft where he was found some time afterward. (See Fig. 1).

Fig. 2 is a view of the side of the boiler room, the wall of which was blown out. In this view may be seen the other boilers, the space occupied by the exploded boiler and the hole which was torn in the roof.

Fig. 3 shows the boiler after striking the ground. A curious fact is that the boiler rests partly upon the stack, showing that the latter reached the ground first. It should be noted from this view that the boiler is practically intact, lacking only the bottom head, at which point the rupture occurred: the damage to the tubes was caused by striking the steel roof of the boiler house.

History of the Boiler

The boiler, which was installed in March, 1898, by the Clonbrock Steam Boiler Company, former makers of the Climax type of boiler, was of 600 horsepower capacity and weighed, complete, considerably over 40 tons. A section of this type of boiler is shown in Fig. 4. The central drum was 48 inches in diameter and 29 feet high, it being made up of

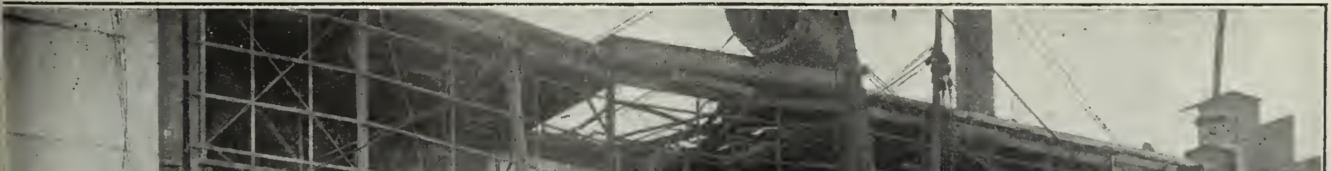

Fig. 2. View Showing Side of Boiler Room Blown Out

a number of sections the circumferential seams of which were single-riveted lap joints, the longitudinal seams being welded. The heads were bumped to a radius equal to the diameter of the drum and were held to the shell by ⅞-inch rivets. Both shell and heads were of ¾-inch firebox steel having a specified tensile strength of 60,000 pounds per square inch. An enlarged sketch of the bottom head and base is shown in Fig. 5. In the later boilers of this type the heads are convex, as shown in Fig. 6, and are made of ⅞-inch metal.

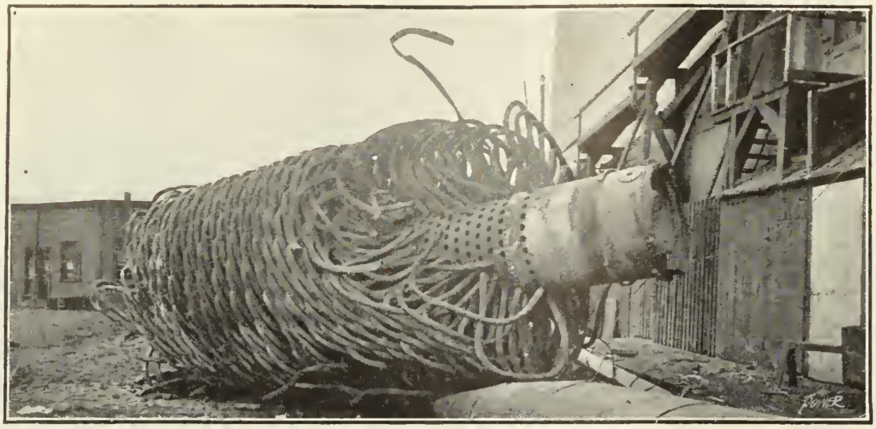

Fig. 3. Boiler After Striking the Ground

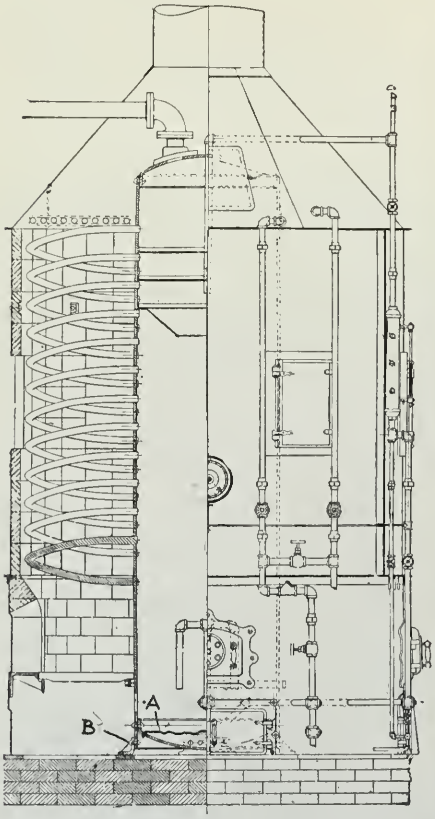

Fig. 4. Section through Boiler Showing Line of Rupture

The boiler had been inspected by the Hartford Steam Boiler Inspection and Insurance Company just a year ago and was to have been inspected again within a week of the date of the explosion. In addition to this, the police department had inspected it last December and had subjected it to a hydrostatic test of 225 pounds per square inch.

Cause of Failure

The cause of the failure is plainly evident. Due to corrosion, the bottom head parted from the rest of the boiler on a circumferential line just below the line of rivets. This line of rupture is indicated by *A* in Figs. 4 and 7. As soon as the head became detached the entire pressure was thrown upon the base flange *B*, which parted at the line of rivets and allowed the boiler to be shot upward. Assuming that the steam pressure did not exceed 150 pounds per square inch, the total pressure on the bottom head, including the weight of the water, amounted to approximately 290,000 pounds. The portion of the bottom hea which was detached was found intac after the explosion.

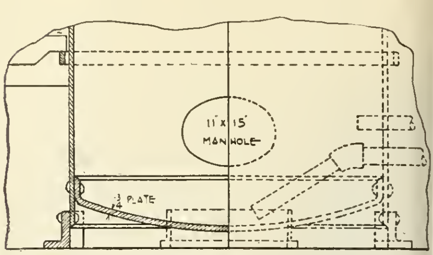

Fig. 5. Bottom Head and Base of Boiler which Exploded

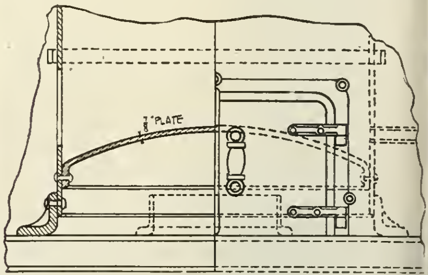

Fig. 6. Bottom Head as Fitted to Late Boilers of the Climax Type

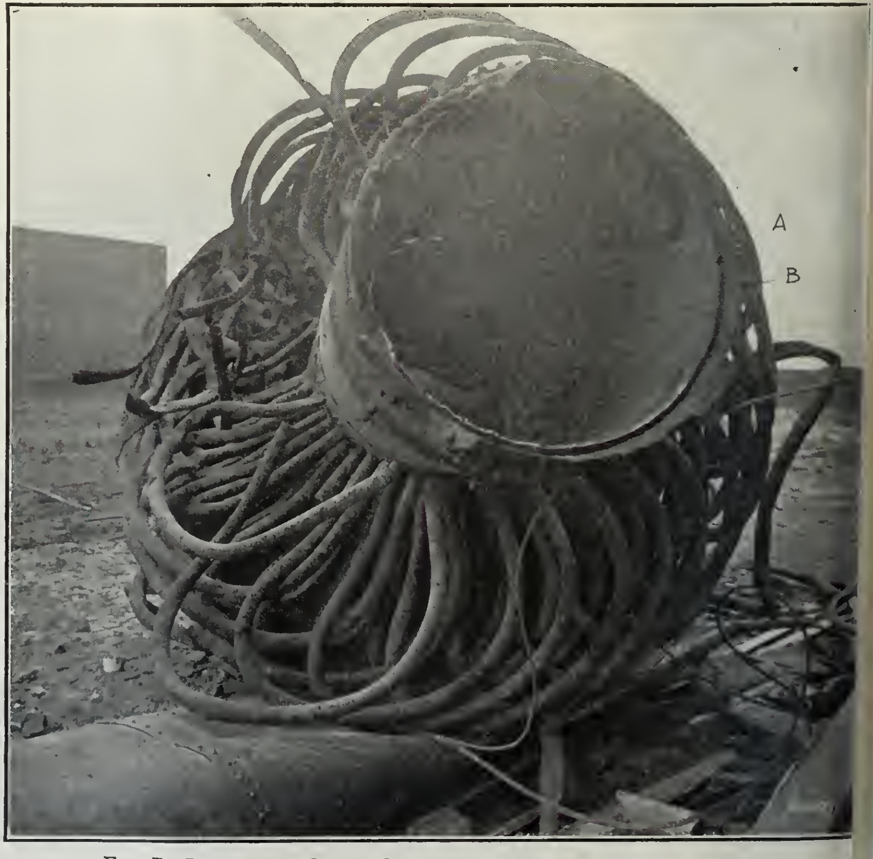

Fig. 7. Bottom of Drum Showing Where Head Was Torn Off

Although the head was originally inch thick, it was found that the me had wasted away to a thickness of a proximately ¼ inch on the line of ruptu and the greater part of this corrosion had taken place on the outside of the head. Owing to the construction of the setting this part of the boiler was inaccessible for inspection.

The whole area of the head was corroded to a great extent but at no part was the metal so thin as at the line of rupture. The fact that this followed almost a true circle around the head would seem to indicate that there was some underlying reason for the corrosion being greatest at this point. One possible explanation of this is as follows:

Owing to the head being bumped to a radius equal to the diameter of the drum, instead of the radius of the drum, the true curvature necessarily had to be deviated from, to a considerable extent, at the flange. With the temperature changes and changes in pressure on the lower head, the latter worked to a greater or less extent, tending with an increase in pressure to assure the form of a hemisphere. With this working of the head the greatest movement naturally occurred on the line of the greatest curvature. This loosened the rust and bared the metal, allowing more rust to form, which in turn became loosened and the action was repeated; hence the decreased thickness of the metal at this point. As the metal became thinner, the effect of the working of the head became greater, until a crack probably developed, resulting in the initial rupture.

At the coroner's inquest undoubtedly many interesting facts will be brought out.

The Career of H. A. Humphrey

Received instruction from Dr. John Perry and Prof. W. C. Unwin. Was closely associated with the late Dr. Ludwig Mond. Inventor of Humphrey pumps and compressors.

Herbert A. Humphrey was born in London and received his technical training in the engineering departments of the Finsbury Technical College and the Central Institution at South Kensington. He had the advantage of receiving instruction under Dr. John Perry, F. R.S., and Prof. W. Cawthorne Unwin, F. R. S., two of the best teachers of engineering in Europe. Some years later, Mr. Humphrey, as the result of original research work, was elected a Fellow of the City and Guilds Institute, being the first to receive this honor.

On leaving college, Mr. Humphrey entered the works of Messrs. Heenan & Froude, Ltd., at Manchester, and was for some time in the bridge-building and constructional iron-work department. He was then made manager of the engineering department, and later was transferred to take full charge of the important engineering works belonging to the same firm in Birmingham. He next entered, as engineer, the large chemical works of Messrs. Brunner, Mond & Co., which company has works in several parts of the country and employs some 4000 workmen. He started and organized several new departments and while with this firm became closely associated with the late Dr. Ludwig Mond, F. R. S. Mr. Humphrey studied Doctor Mond's gas-producer plant and read a paper before the Institution of Civil Engineers, giving for the first time a complete balance sheet of all heat quantities entering into the gasification of bituminous fuel in a large commercial plant. It was at Brunner, Mond & Co's. works that Mr. Humphrey installed the first gas engines exceeding 400 horsepower erected in England, and ever since he has been closely connected with the development of large gas engines and become recognized as one of the leading experts on this subject.

It was at Doctor Mond's suggestion that Mr. Humphrey established himself in London as a consulting engineer, and this enabled him to take up a great deal of interesting work and to closely associate himself with Doctor Mond's group of companies. One of these companies was the South Staffordshire Mond Gas (power and heating) Company which obtained its parliamentary powers to distribute producer gas over a large area after a very severe fight against local opposition in both Houses of Parliament. Mr. Humphrey designed the central-station plant and superintended its erection at Dudley Port and so was instrumental in starting the first public supply of gas for power and furnace purposes.

H. A. Humphrey

Mr. Humphrey's constant connection with large gas engines caused him to endeavor to discover an internal-combustion engine which should be free from the many defects and complications of the existing types and at the same time should solve the turbine problem and give a high-speed rotary motion free from cyclic irregularity. The outcome of several years' work in this direction has resulted in the development of the internal-combustion pumps and compressors which bear his name.

Mr. Humphrey has been consulting engineer to many well known and important works and has been elected to full membership of the Institutions of Civil, Mechanical and Electrical Engineers respectively. He is the author of several papers read before these institutions in connection with which he has received the Telford gold medal and Telford premium, and the Willans gold medal. The latter is only given once in six years, for the best paper in that period, and is therefore specially prized.

The rapid development of the Humphrey pumps and compressors has now made it necessary that Mr. Humphrey should devote practically all his time to the affairs of the Pump and Power Company, Ltd., formed as a parent company to grant licenses for the manufacture of Humphrey pumps and compressors in all important centers of the world.

A new kind of steel, under the name of "cementation steel," is, we hear, being manufactured at Chester, Penn. It is of the high-carbon variety, but contains more sulphur and less manganese than ordinary tool steel. The steel is so dense that it remains, so it is said, unresolved under the microscope with a magnification of 1200 to 1600 diameters, although that of the open-hearth, crucible and bessemer steels can be resolved with a magnification of 100. Its elastic limit is said to be very high, and its ultimate strength 80,000 pounds. It is intended specially for cast gears, crank shafts, connecting rods and so on. The process of manufacture takes from six to eight weeks.

Carbonic acid in the atmosphere arises from respiration of animals, from processes of combustion, from the decomposition of organic substances. Boussingault estimated that in Paris the quantities of carbonic acid produced every 24 hours were: population and animals, 11,895,000 cubic feet; combustion, 92,101,000 cubic feet.

Inquiries of General Interest

Horsepower of Belt

What horsepower will a 12-inch double-leather belt transmit running over a 6-inch pulley making 200 revolutions per minute?

H. P. B.

If the working stress is 60 pounds per inch of width, a belt traveling 550 feet per minute will transmit 1 horsepower for each inch of width. If the working stress is 30 pounds per inch of belt width, it will have to travel 1100 feet per minute to do the same work. Most of the rules given for the horsepower of belting fall between these extremes.

Neglecting the thickness of the belt, it will travel over the pulley at the rate of 320.7 feet per minute, and with a stress of 60 pounds per inch of width, will transmit

$$\frac{320.7}{550} \times 12 = 6.997 \textit{ horsepower}$$

With 30 pounds stress per inch of width the belt will transmit

$$\frac{320.7}{1100} \times 12 = 3.498 \textit{ horsepower}$$

The horsepower which a belt will transmit under proper conditions falls within these limits, but by excessive tension which destroys its elasticity and shortens the period of its usefulness, it may be made to do much more.

Banking Fires

I have two boilers. During the day they are loaded to their capacity but at night the load drops to less than one-half of that during the day. Do you think it advisable to bank one fire every night, or use both, considering the cracking and breaking of tile and for economy?

B. B.

During the day run, while the fires are being forced, the burning of the fuel is going on under the best conditions and each pound of carbon is converted into CO_2, giving up nearly, if not quite, all of the 14,500 heat units it contains.

At night, when the demand for steam becomes less, the heat units liberated must be reduced in number. If this is accomplished by reducing the air supply to the fire, as it must be if both boilers are operated, much of the carbon instead of giving up 14,500 heat units per pound in being burned with plenty of air will give up approximately only 5000 heat units per pound. It is probable that more than twice as much coal is burned in operating both boilers on the light load than is necessary if only one boiler is used. With proper attention given to avoiding sudden furnace temperature changes there is little danger of cracking the tile.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Viscosity and Flash Point

What is meant by the viscosity and the flash point of an oil, and how are they found?

V. F. P.

Viscosity is the resistance a fluid offers to its separation into drops or particles. It is measured by the time required for a given quantity of oil to flow through an orifice. As there is no accepted standard viscosimeter the term means nothing unaccompanied by the name and make of the instrument used in the determination of the viscosity of the liquid.

The flash point is the temperature at which the oil gives off inflammable vapor and is found by slowly heating a small quantity of oil, in which a thermometer is inserted, until a vapor is given off which will flash or burn on the application of the flame of a burning match, candle or gas jet.

Steam Bound Pump

How does a steam pump act when it is steam bound, and what kind of a noise does it make?

S. B. P.

This term is usually applied to pumps which are used for pumping hot water from open heaters or receiving tanks when they fail to operate because the hight of the water above the suction valves is not sufficient to open them, and the pump, having no work to do because there is no water in the cylinder, runs away, clattering or pounding according to the steam pressure and the amount the throttle valve is open.

Air Bound Pump

How does the water end of a pump act when it is air bound, and what does air bound mean?

A. B. P.

A pump is said to be air bound when the air pressure in the cylinder in front of the advancing piston is not sufficient to open the discharge valves and too great behind the piston to permit the water to flow to the pump. It is caused by large clearance space in the water cylinder, by leaking piston and valves and by a high suction lift or by a combination of all three conditions. An air-bound pump will run at a high rate of speed and perhaps knock at the ends.

Relative Diameter of Pump Cylinders

If a pump working against a pressure of 145 pounds per square inch with a steam pressure of 130 pounds per square inch, 16 feet lift has an 8-inch steam cylinder, what will be the size of the water cylinder?

A. G. C.

Assuming that 85 per cent. of the boiler pressure is realized in the steam cylinder and that the piston moves at the rate of 50 feet per minute, the foot-pounds of work done per minute will be

$$130 \times 0.85 \times 50.26 \times 50 = 277{,}686.5.$$

If the friction of the pump and the water absorb 10 per cent. of this, there will be left for the work of lifting the water

$$277{,}686.5 \times 0.90 = 249{,}917.85$$

foot-pounds per minute. The 145 pounds pressure per square inch against which the water is pumped is equivalent to 333.5 feet of head, which, added to the 16 feet of suction lift, makes 349.5 foot-pounds of work for each pound of water delivered per minute and the total will be

$$249{,}917.85 \div 349.5 = 715$$

pounds of water delivered per minute. As the water piston travels at the rate of 50 feet or 600 inches per minute, its area is the quotient of the total weight by the weight of 600 cubic inches of water; 600 cubic inches of water weigh 21.65 pounds.

$$715 \div 21.65 = 33.02$$

square inches area of water piston.

The diameter of a circle whose area is 33.02 square inches is approximately 6½ inches and is the diameter of the water cylinder required by the conditions named. In commercial practice a water cylinder 5 inches in diameter would be selected and the steam pressure reduced at the pump by throttling.

Siphon in a Vacuum

Consider a regular siphon being placed in a closed receptacle and a vacuum obtained, would the siphon cease to operate, or in other words, does the siphon depend alone on the air pressure?

S. I. V.

A siphon will not operate in a vacuum. There must be pressure enough on the surface of the liquid at the higher level to raise it to the highest point in the system.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for October, 1910, *was* 132,000.

November 1............ 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

The Value of CO_2 Indicators

A report of the Committee on Power Generation of the American Street and Interurban Railway Engineering Association did not, to say the least, do the CO_2 recorder any harm by fulsome praise. Both G. H. Kelsay, who handled this particular subdivision of reports, and C. E. Roehl, who wrote that portion of the report dealing with forced draft, say that the hopes, which had been aroused by the introduction of CO_2 recorders, have not been realized. The boiler efficiency is not at all proportional to the percentage of CO_2 in the flue gases and the CO_2 recording apparatus is often found in an inoperative and neglected condition. Mr. Kelsay is hopeful, however. "Gas analysis as applied to boiler-room practice will," he says, "no doubt gradually adjust itself into methods that will be found very beneficial in developing and maintaining higher boiler economies; but it is possible that overdrawn ideas have been entertained by many engineers as to its value; also many engineers have failed to give the subject the attention that it deserves, and would probably find more merit in the analysis of flue gases than they now think."

There is no doubt that the lower the temperature of the gases going to the uptake, the less heat will be carried up the chimney; but one might easily buy a thermometer, put it into the uptake, tell the boiler attendant to keep the temperature as low as possible, and still burn as much coal as before, or more. If the low temperature were produced by letting in an unnecessarily large quantity of cold air, he certainly would burn more. It is quite possible, probable in fact, that his boiler-room expenses will be less, if he lets the gases go from the boiler at a somewhat high temperature, rather than at a temperature not much above the heating surface; for, he would make the heating surface "earn its keep" and get along with less boilers and run with furnace conditions more conducive to perfect combustion.

But, because of these things, the importance of flue-gas temperature as a factor in boiler performance should not be disparaged, its measurement discouraged and the instrument by which it is measured condemned.

It appears to be true that the best overall boiler efficiency is not obtained when the furnace is making the largest percentage of CO_2. Just why this is so has not been satisfactorily explained. It is probably due to the fact that a boiler, to be of any use as a boiler, has to make not only a large *percentage* of CO_2, but a large *amount* of it. If the air were let in so parsimoniously that each atom of oxygen in it must find an atom of carbon in order that the combustion may be complete, and were kept in the furnace until this complete chemical union took place, the percentage of CO_2 would be high, but the process would be so slow that the boiler would do no work. In order to get enough coal burned in a given time, it must be supplied with a surplus of air, such as will allow the combustion to go on rapidly and, with this surplus air, the percentage of CO_2, somewhat over twenty for the ideal case, will fall to from twelve to sixteen for the best usual running conditions. The better means one has for effecting a prompt and ultimate mixture of the gases, the better the furnace is arched and baffled, the higher percentage of CO_2 can be carried without making CO, which is so destructive to the efficiency.

Because the best boiler efficiency is not attained when the percentage of CO_2 exceeds twenty is no reason to decry the benefit of flue-gas analysis. Because a man who can hardly read the figures on the pressure gage cannot gather all the significance of its indications is no reason to condemn the CO_2 recorder. That a rather complicated and very delicate instrument should be often out of working condition when placed in charge of a man whose finest work is usually done with a monkey wrench, is not surprising. To the man who understands its significance, a knowledge of the CO_2 content of his flue gases is of value, and the CO_2 apparatus will give it to him if it has the right kind of care and supervision.

The Hydrostatic Test

There have been during recent years three instances in which disastrous boiler explosions have been prevented by tracing to its source and finding the cause of the leaking steam which appeared through the brickwork of the boiler settings.

It is generally, if not universally, admitted that the lap-seam form of boiler construction which prevents the shell of the boiler from taking the form of a true cylinder under any condition of pressure or lack of it, but still compels a change of shape for every change in pressure, causes a bending of the sheet at or near one of the rows of rivets which finally produces cracks.

These cracks can be seen only by unmaking the joint, and even when suspected to exist cannot be discovered. But the fact that under ordinary working pressure they have made their presence known before the sheet was weakened to the breaking stage points a way to their discovery before the danger line is reached. Periodic intelligent application of the hydrostatic test fifty per cent. in excess of the pressure at which the safety valve is set will in most if not all cases reveal any dangerous weakening of the sheet before it results in failure.

Something like two years ago a boiler used by a contractor in Boston failed under the hydrostatic test when the pressure reached a point three pounds above that at which the safety valve was set. It is, of course, uncertain how many times the varying steam pressure would have bent the cracking sheet before it would have yielded to the regular working pressure and another hidden-crack explosion would have passed into history.

While the hydrostatic test is not considered by all as desirable, it seems to be and probably is the only safe method of determining if a boiler shell or drum will stand a pressure slightly in excess of that for which it is designed. It is certain that it will reveal weaknesses in material and structure that can be detected in no other way and should be applied at intervals to all boilers used for pressures exceeding the ten pounds per square inch usually employed for heating purposes, care being taken not to stress the metal above its elastic limit.

Boiler Explosions

Boiler explosions are becoming almost as common as automobile accidents. The latter have now ceased to excite more than a passing interest, and the average individual upon reading an account of one usually dismisses it from his mind with the chance remark: "Well, if they will take the risks they must be prepared for the consequences." The public, however, cannot afford to regard boiler explosions in a similar light. It is a question of vital importance not only to the engineer and fireman but to the public at large, involving the personal safety of everyone employed in a building in which a steam boiler is in service. When one stops to consider that nearly every large building in New York City is equipped with one or more boilers upon whose safe operation hangs the lives of scores of persons, the gravity of the situation is apparent.

The recent explosion at the works of the American Manufacturing Company, described elsewhere in this issue, strongly emphasizes two points: First, the necessity of having all parts of the boiler accessible for inspection; second, the necessity of a rigid inspection. Failure to comply with the former requirement, however, is no excuse for laxity in the latter. If any part of a boiler is inaccessible, it is the duty of the inspector to demand that the owners provide means of its inspection, and furthermore, he has the power to withhold the certificate until satisfied that the boiler may be safely operated under the particular operating conditions.

It is possible that the coroner's inquest may bring out facts that will more closely fix the blame for the disaster, but whatever the verdict, the two points just mentioned stand out as self-evident facts.

Engineering Carelessness

"William Blank was badly scalded here this afternoon at the plant of the ———— ———— Company, and will die. He entered a boiler to clean it out, and was seen to leave it soon after to get something. The engineer, not noticing that he had returned to the boiler, turned on the steam and scalded Blank so badly that he was removed to the hospital, where it is stated he cannot live."

The above is merely one of a dozen of such newspaper items that appear from time to time. When the article is analyzed, it will be seen that there is something more to it than the mere statement of facts that a man was probably scalded to death because the engineer of the plant opened a steam valve without knowing that a man was in the boiler.

The analysis shows that there was lack of system in the plant. If there had been some method of tagging the stop valve located between the boiler and the steam main, bearing the words "Keep Closed," there would have been no excuse for anyone to open that valve until the tag had been removed.

It is difficult to conceive of any engineer connecting one boiler to another after making repairs or washing out, without first being assured that the boiler was ready for connecting. It is more difficult to understand why any sane man should turn live steam into an empty boiler, as stated, although it is quite possible that the clipping is in error. It may be that the man was in the empty boiler and the engineer decided to blow down another boiler that was under steam, while the blowoff valve on the cold boiler was open.

To the man who was scalded it does not make any difference how the deplorable accident occurred. The engineer was to blame for the accident and the scalded man is in danger of losing his life. An engineer who opens a valve under similar conditions without knowing, as far as human knowledge permits, that he can safely do so, is not only negligent but is criminally careless and not fit to have charge of a steam plant.

The incident furnishes a lesson which the careless engineer might well heed. It is better to climb up on top of a boiler a dozen times and make sure that no one is in it before closing it up or blowing down another boiler without knowing the blowoff of the cold boiler is closed, than to run the chance of scalding a man to death.

The engineer may try to justify his actions when they turn out disastrously, but that will not remove the consciousness that the accident could have been avoided had proper precautions been taken.

A mistake which results in the killing of a fellow workman will never be repeated by the man who made it. By taking proper precautions and using common-sense methods in doing the daily work around the plant, such fatal mistakes may be avoided.

Obsolete Literature

The following statement appears in a recent edition of one of the engineering handbooks: "While our conception of the fundamental principles of electrical science can of necessity have undergone no very considerable alteration, those essential details which in effect constitute the working data of the engineer have so altered and grown that books published only a few years ago are already obsolete." There can be no question as to the truth of this statement; in fact, it applies not alone to the electrical field but to all branches of engineering. For this reason too much care cannot be exercised in the selection of reference books.

In the catalogs of many publishers, as well as dealers, there is nothing in the descriptive matter to indicate the "out-of-dateness" of the book advertised. The same descriptions under which they were sold when just off the press are frequently used in advertising them as long as it is possible to command sales. It would be unfair to accuse publishers and dealers of attempting to obtain money under false pretenses; instead the fault undoubtedly lies in the fact that they do not stop to consider how fast conditions are changing, and in many instances the men who finance the publication of books know nothing of the subjects treated.

The ordinary buyer of technical literature has not as good an opportunity to become familiar with the subjects covered as have the publisher and bookseller. But the latter do not always improve their opportunities. Therefore, after a purchase has been made, and money parted with, it is often found that the book is not only valueless, but if made use of in practice might do infinite damage. In view of these conditions, it is well for a prospective buyer to first make a personal inspection of a book and before accepting its contents as authentic, to check its statements with the uptodate practice brought out in current technical periodicals; also, if possible, to note the opinions as expressed in the "book-review" columns of these mediums.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Induction Oxidizer System

This device has been designed with a view of preventing smoke and producing a high rate of combustion. It consists of an air flue built in the bridgewall, as shown in Figs. 1 and 4. On top of the bridgewall a base of tile is placed in which tile deflecting tubes are set. The base and deflecting tube are shown in Fig. 5; Fig. 2 shows them assembled. The tubes are hollow and are perforated, the holes being so placed that the jets of air converge, as shown in Fig. 3.

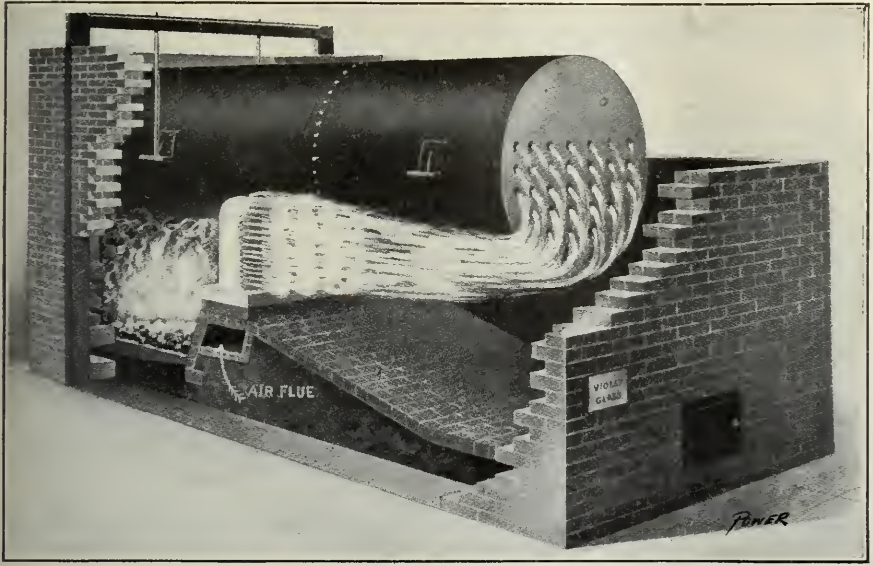

Fig. 1. Showing Oxidizers and Air Flue

Air is admitted to the combustion chamber through the air flue and deflector tube, escaping through the air holes. It is claimed that but 1 per cent. of the air used in the combustion of the fuel passes through these tubes, and in so doing is heated up to 1292 degrees Fahrenheit. Consequently the air is expanded in volume, which causes it to issue from the numerous air jets of the deflector tube with considerable velocity. This heater air, it is claimed, ignites the fuel gases that have passed the bridgewall and deflector tubes.

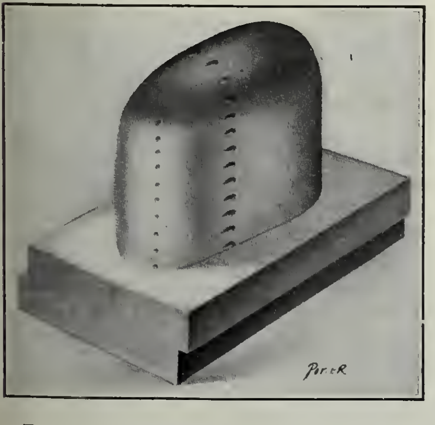

Fig. 2. Base and Deflecting Tube

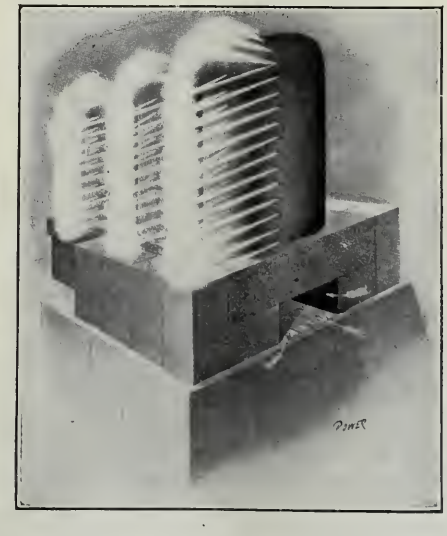

Fig. 3. Showing Air Jets

The builders state that these jets of air set up an induced draft through the grates, which greatly increases the draft and increases the capacity of the boiler.

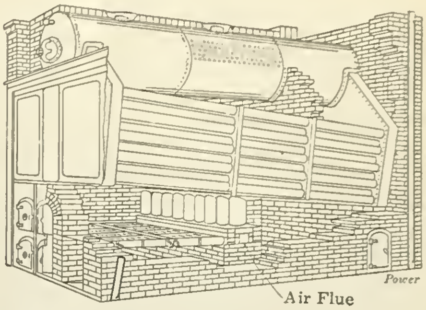

Fig. 4. Oxidizer Under Water-tube Boiler

This system requires no attention after being installed, and it is said that the tile or firebrick deflector tube and base are practically indestructible under the action of the furnace temperatures found in boiler practice.

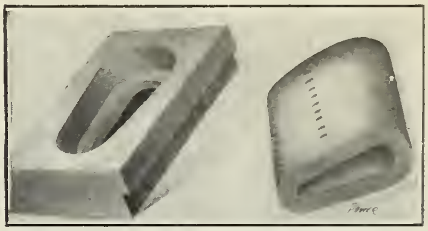

Fig. 5. Base and Deflecting Tube

This system is made by the Induction Oxidizer Company, Lancaster, Penn.

Flexible steel belt conveyers have been recently put on the market by a Swedish manufacturer and a number of installations of this new type of conveyer have been made in northern Sweden. A hardened-steel ribbon, or belt, from 8 to 16 inches wide and 0.03 to 0.04 inch thick, runs flat on the bottom of a trough with sloping side boards. The belts are made in lengths up to 328 feet, and longer belts can readily be obtained by riveting two or more lengths together. The belts are run on wooden pulleys covered with rubber belting. The driving pulleys should be 3.28 to 4 feet in diameter, and the idler pulleys 16 to 20 inches.

It is stated that these belts are suitable for transporting ore, coal, planks, wood stumps, coke, cement slag, ashes, sawdust, etc., and that the power cost and cost of upkeep and repair are inconsiderable. In case of fracture, a riveted piece can be inserted. They are known as Sandvik belts and are made by the Sandvikens Jernverks Aktiebolag, Sandviken, Sweden.

Reliability of Steam Meters*

By R. J. S. Pigott

Anemometer, or Jet-turbine Type

Holly and Lindenheim: Holly's device was a small vane which operated by a small bypass, with entry pointing upstream and discharge pointing downstream; the vane wheel drove a clockwork integrating mechanism through a gland. The errors in this meter are: Friction of the gland or stuffing box on the vane-wheel spindle, variable friction losses in the bypass system and the absence of a correction device for variation of steam pressure.

Lindenheim's machine was very similar to Holly's, but added a correction device for variation of steam pressure, and in this way was somewhat superior; but both are rendered very unreliable by the fact that friction of the vane-wheel spindle bearings and gland has such an enormous influence on the correctness of the reading.

Throttle Disk Type

The Bendemann, Gehr, St. John, Sargent and several other meters are of this type. The means of operation in all cases is some form of throttling disk or cup in a passage or over a seat, loaded so as to offer a slight resistance to the flow of steam through the passage. Consequently, as more steam flows through the meter the disk or cup is forced up from its seat, so as to expose more area for the steam to flow through, accompanied by an increase in pressure drop through the machine. The rise of the disk is some function of the flow, usually not a straight-line function, as the position of the disk is influenced by eddy currents, which vary with rates of flow and with the shape of the disk and seat or passage.

The Bendemann meter consists of a disk floating in an inverted conical passage, in which the steam is flowing upward, the spindle of the throttle disk is carried up into a glass tube (under pressure) and the scale is attached to the tube. Thus the friction error of a stuffing box on the spindle is avoided, but no recording device is attached to the meter. In a later modification, the spindle was extended downward and ended in a long taper plug partially closing an orifice. The amount of steam released from this orifice, therefore, varied with the main flow, and was condensed and passed through a small water meter. The St. John meter has a taper plug, moving in an orifice vertically, the movement being transferred by a system of levers, through a stuffing box, to a recording sheet. There is no device on these meters to correct for pressure variation. The Sargent meter has a conical cup, fitting over a conical seat; the motion of the cup spindle is transferred without levers through a labyrinth stuffing box, and has a pressure-variation correction in the shape of a bourdon gage.

This type of meter has the advantage of large movement of the throttle disk over the range, which gives open scales; but the friction of glands and lever systems introduces an error which may overbalance this. The throttling action tends to erode disk and passages, and must eventually destroy the accuracy of the meter with wet steam. If the meters are spring loaded, or use spring-pressure corrections, they become subject to the additional inaccuracies of springs. The bulkiness of this type of meter prevents its use in anything above 6-inch pipe sizes.

Electric Type

An electric meter was proposed by Professor Thomas, in which the temperature of steam is taken before and after passing an electric-heating coil; the energy sent into the coil is also measured. If the quality and pressure are known the above quantities give a measure of the flow, which is absolutely accurate; but the quantity of electrical energy needed becomes enormous if the steam is 2 or 3 per cent. wet, practically prohibiting the meter. For superheated steam or dry gases it is a very good means of measurement and there is no limit to its size.

Pitot Type

The remaining forms of steam meter are those in which difference of pressure is employed—working on the principle of the Pitot tube and the Venturi meter. These meters have the best opportunities for accuracy, since there are no moving parts to stick or cause friction, and no parts to wear from erosion. They can also make use of mercury and other accurate differential gages.

The advantages of the Pitot tube type are obvious—nothing to do to the piping but put in a small Pitot plug; the recording device may be at any reasonable distance from the point of measurement, and may be made portable. But the total difference of pressure amounts to only ½ inch of mercury at ordinary velocities, so that a very slight leak or bubble of air in the piping to the meter recorder is sufficient to throw it off, and only a most delicate recording device can be used. The Venturi type has the additional advantage of being designed for almost any difference of pressure, allowing of a robust recording device and not being easily affected by small leaks.

One feature which is not taken account of by any meter, and which has a large effect, is the variation of moisture in the steam.

The Pitot tube, Venturi and electric meters are most affected by variation in quality, but their accuracy in actual measurement partially offsets this when the quality variation is not compensated for.

To sum up, the throttle-disk meters suffer, both in measurement and mechanically, from moisture, are usually impaired in value by friction in the mechanism, and are prohibited in large sizes by bulk and expense of installation. In small sizes they are reasonably good.

The electric meter is most accurate, but is too expensive to operate for wet steam, and requires knowledge of the quality to be of any value whatever.

The Pitot meter is simple, durable and easy to install and operate, but is affected by very slight leaks or pulsation in the steam mains.

The Venturi meter has the same advantages as the Pitot tube, with the additional point of much greater differences of pressure and less delicacy. It is probable that the Venturi form of meter will offer the most reliable and accurate means of measurement.

All the meters are affected by variation in quality, and some by variation in pressure (when no compensation device is used). In the present state of the steam meter, it seems as if the safest way to measure the steam output is by means of a Venturi meter placed in the feed line to the boiler. By this means results can be secured which give the output to 1 per cent.

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*Portion of the report of the Committee on Power Generation of the American Street and Interborough Railway Engineering Association, Atlantic City, October 10-14.

Elevator Tank Fails

At 8:30 o'clock Saturday morning, October 22, the hydraulic elevator tank in the engine room of the Clarendon hotel, Brooklyn, N. Y., failed, with the result that the entire plant was put out of service. Six men were more or less seriously injured, none fatally, however. The tank was 5x9 feet in size and built of ⅜-inch plate with single-riveted lap joints. It had been in service for about 15 years. The normal pressure in the tank was 130 pounds. The rupture started at the longitudinal seam.

When the tank failed the pressure in the system forced the tank from its foundation and threw it across the room. In so doing two 4-inch engine leads carrying 100 pounds steam pressure; six 2-inch leads for various pumps; an 8-inch pump-suction line and a 6-inch pump-discharge line were broken. The main switchboard, one end of which extended across the path of the tank, was wrenched loose and turned partly around. A 12 and 10 by 8-inch duplex pump used for auxiliary elevator service, was moved off its foundation about 2 feet. A 10x12-inch Ames engine driving a 35-kilowatt generator and standing in the path of the tank was disabled.

The total damage to the plant amounts to about $5000. Temporary central-station electric service was installed within three hours after the accident had occurred.

Asher Nelson, chief engineer of the Clarendon plant, will conduct an investigation to discover, if possible, the exact cause of the failure. Accidental over pressure is the theory which he most favors at this writing, although the fact that the tank had been in use for about 15 years points to the probability that the joint may have become weakened to such an extent as to cause the failure.

NEW PUBLICATIONS

Standard Handbook for Electrical Engineers. (Third Edition.) Published by McGraw-Hill Book Company, New York, 1910. Soft leather; 1500 pages, 4x7 inches; liberally illustrated. Price, $4.

This excellent reference book has been extensively revised and the contents have been rearranged slightly. The quantity of data and information has been considerably increased, but the illustrations remain very unsatisfactory in most instances, and the treatment in many places is too concise to be explicit—for example, the statements in paragraphs 282 to 286, section 3, relating to the measurement of power in electrical circuits. There are many statements that should be elaborated or conditioned, as the definition of an atmosphere, on page 20; that of capacity, on pages 26 and 29; that of resistance, on page 27; the statement in paragraph 158 of section 7 and most of those resulting from the one cited. This statement is really worse than dogmatic or obscure; it is untrue, if the author means what is usually meant by the inductance of a direct-current armature—namely, the inductance of those coils short-circuited by the brushes during commutation. The total ampere-turns per pole in the armature winding are not a factor in the inductance of the short-circuited coils, unless there be assumed a fixed ratio between the number of coils simultaneously short-circuited and the total number on the armature, which would be impractical. However, the author is a thoroughly capable designer of electrical machines and doubtless meant something else than what he said.

The book is deficient in several respects, notably in the discussion of the Edison storage battery and in that no attention whatever has been paid to arc-lighting dynamos.

There are also some obvious inaccuracies, notwithstanding the full revision given the book. For example, on page 31: "Magnetomotive force is the force per unit cross-section of the circuit." The geometry of the magnetic circuit has nothing whatever to do with the m.m.f. of the exciting coil, which is solely a question of convolutions and electric current. Again, on page 300: "The flux in direct-current depends on the ampere-turns but in alternating-current magnets it depends on the voltage-turns." We do not pretend to know what voltage-turns are, but if any magnetic flux can be produced by volts and turns, without any amperes, we should enjoy seeing it done.

In the final formula of paragraph 94, section 5, the symbol R should be a P. Paragraph 58, section 5, does not belong in a discussion of field magnets. Paragraphs 60 to 73 of this section appear to have been collated at random.

The foregoing blemishes are typical of some sections of the book. Other sections are remarkably free of defects and weak spots. These few have been pointed out not in any carping spirit but with a view to inciting the editors to wipe off their spectacles more thoroughly, and remove the flyspecks from what would otherwise be a shining example of reference-book construction.

Rules and Formulae, with Suggestions Pertaining to Good Practice, Indorsed and Adopted by the International Master Boiler Makers' Association. Published by the association and supplied by the secretary, 95 Liberty street, New York City. Sixty-five pages, 3½x5¾ inches; cloth. Price, $1.

This little work was indorsed at the Louisville, Ky., convention as expressing the consensus of opinion of the association on what constitutes good practice in the construction and management of steam boilers.

It supplies in a concise form rules, information and tables of value to those who are engaged in designing, inspecting and operating steam boilers. Especial attention is given to a modification of D. K. Clark's formula for the staying of flat surfaces with riveted screw stays. Otherwise there is little or no original matter in the book, as it consists of selections from many sources, which are believed to be the best. Unfortunately, the book is not indexed.

The Construction of Graphical Charts. By John B. Peddle. Published by McGraw-Hill Book Company, New York, 1910. Cloth; 115 pages, 6x9 inches; 58 illustrations. Price, $1.50.

Professor Peddle has made a fine "start" in his little volume of chart exposition, but it is not much more than a start. The plotting of charts to solve a number of equations is explained—or rather illustrated—very concisely, and the book should be very useful to engineers who have to deal with the forms of equations covered, provided they are already well versed in the principles of the various charts and require merely a typographical substitute for memory. If the author had elaborated his work to the extent of making his explanations more elementary and increasing the number of examples of each form of equation, the book would have been instructive and valuable to those who have need to plot such charts but have not had the mathematical training presupposed by the author.

Hendricks' Commercial Register of the United States. Published by S. E. Hendricks Company, New York, 1910. Price, including express charges, $10.

This is the nineteenth annual revised edition of Hendricks' well known "Register." It is considerably larger than that of previous years and contains the names of the manufacturers of over 5000 articles, none of which have appeared in any previous edition. The total number of classifications in the present edition is 35,481, each representing some machine, tool, specialty or material required in the architectural, engineering, mechanical, electrical, railroad, mining and kindred fields. The total number of names and addresses is upward of 350,000.

By virtue of the manner in which classification is carried out, the register may be used for mailing purposes as well as for purchasing.

"Exhaust Steam Heating Encyclopedia" is the title of a profusely illustrated 117-page book describing and illustrating commercial systems of exhaust steam heating, which is being sent out by the Harrison Safety Boiler Works, of New York City. Other subjects discussed are exhaust steam-heating in connection with condensing plants, the application of low-pressure turbines, the use of exhaust steam in absorption ice machines, etc. Over 30 pages are occupied by tables and engineering data of value to those who design, install or operate exhaust steam-heating systems.

The new Cochrane steam stack and cut-out valve heater and receiver is described, and numerous illustrations show how it is connected to various exhaust steam-heating systems.

Bulletin S-77 on steam consumption and friction losses of Rice & Sargent Corliss engines gives data regarding compound-condensing and noncondensing and simple noncondensing engines. Steam consumptions as low as 18.33 pounds per indicated horsepower-hour are shown on a noncondensing compound engine at 150 pounds pressure with saturated steam, showing an efficiency of 79.2 per cent. in the use of steam. On condensing compound engines, at 150 pounds pressure, the steam consumptions run as low as 12.1 pounds, and on a noncondensing simple engine at 149 pounds pressure the steam consumption is shown to have been 20.98 pounds. The friction-load indicated horsepower runs as low as 3.8 per cent.

on a compound and 3.6 per cent. on a simple engine, while the mechanical efficiency at full load runs as high as $98\frac{1}{2}$ per cent. The tests to determine these results were made by engineers such as Prof. D. S. Jacobus, George H. Barrus and others, in all cases representing the purchasers of the engine.

PERSONAL

D. M. Forker, formerly with the Republic Iron and Steel Company, Youngstown, O., has been elected vice-president of the D. T. Williams Valve Company, Cincinnati, O.

The Elliott Company and the Liberty Manufacturing Company, of Pittsburg, Penn., announce that Herbert E. Stone has been appointed manager of their New England office, 102 High street, Boston, Mass.

Samuel M. Hildreth has been elected secretary and general sales manager of the Ideal Automatic Manufacturing Company, manufacturers of controlling valves, with officers at 125 Watts street, New York City.

Albert E. Guy, who for the last six years has been at the head of the engineering department of the De Laval Steam Turbine Company, of Trenton, N. J., has resigned that position in order to head a company which will manufacture and place upon the market the Guy centrifugal pump. Mr. Guy has had a wide experience in the engineering field, not only with a number of the more prominent corporations but also in the Bureau of Steam Engineering of the Navy Department. For the last four years Mr. Guy has given personal attention to the centrifugal-pump department of the De Laval company, during which time he has made a number of important discoveries in this branch of hydraulic engineering, all of which will be embodied in the designs of new pumps which he intends to put upon the market. The title of Mr. Guy's new company is the Standard Pump Company, and its headquarters are at 30 Church street, New York City.

BOOKS RECEIVED

Notes on Drawing. By Horace P. Fry. University of Pennsylvania, Philadelphia, Penn. Cloth; 57 pages, $6\frac{1}{4}$x$9\frac{1}{4}$ inches; 34 illustrations; indexed.

Power Gas and the Gas Producer. By J. C. Miller. Published by Popular Mechanics Company, Chicago, Ill. Cloth; 184 pages, $5\frac{1}{2}$x$8\frac{1}{4}$ inches; 18 illustrations. Price, $1.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. Charles W. Foster, New Haven, Conn. 972,841.

EXPLOSION ENGINE. William Anthony Jones, West New Brighton, N. Y. 972,869.

FLUID-PRESSURE TURBINE. Charles Algernon Parsons, Newcastle-upon-Tyne, England. 972,908.

INTERNAL COMBUSTION ENGINE. Martin L. Williams, South Bend, Ind. 972,966.

HYDRAULIC MOTOR. Paul Andersen, Pinole, Cal. 972,981.

HOISTING ENGINE. John Welker Freeman, Joplin, Mo. 973,028.

INTERNAL COMBUSTION ENGINE. Walter J. O. Johnson, Chicago, Ill. 973,048.

ROTARY ENGINE. Christian F. Paul, Jr., Peekskill, N. Y., assignor of one-tenth to Kenneth Mott, Peekskill, N. Y. 973,064.

GAS ENGINE. John M. Kroyer, Stockton, Cal., assignor to Samson Iron Works, Stockton, Cal., a Corporation. 973,118.

INTERNAL COMBUSTION ENGINE. Charles H. Sergeant, Bridgeport, Conn. 973,140.

CURRENT MOTOR. Albert Vestal and Delbert F. Bennett, Redding, Cal. 973,241.

INTERNAL COMBUSTION ENGINE. Chas. Elisha Fogg, Kennett Square, Penn. 973,268.

FLUID-PRESSURE HEAT ENGINE. Jean Molas, London, England. 973,289.

OSCILLATING MOTOR. Harry A. Adams, Reading, Penn., assignor, by mesne assignments, to Edwin E. Pryor, Reading, Penn. 973,400.

WATER WHEEL. Zachariah C. Ferris, Caldor, Cal. 973,419.

BOILERS, FURNACES AND GAS PRODUCERS

GRATE BAR. Anderson Wilderspin, Grand Saline, Tex., assignor of one-half to David C. Earnest, Dallas, Tex. 972,965.

OIL-BURNING STEAM BOILER. William A. Rogers, Los Angeles, Cal., assignor to Charles A. Hammel, Los Angeles, Cal. 973,138.

METHOD OF PRODUCING GAS. Edmund G. Jewett, Bellingham, Wash. 972,864.

LIQUID FUEL BURNER. Hooker Shapel, Iola, Kas. 972,931.

STEAM BOILER. Charles A. Hammel, Los Angeles, Cal. 973,111.

FURNACE. Hanna Haas, New York, N. Y. 973,273.

POWER PLANT AUXILIARIES AND APPLIANCES

SPARK PLUG. George Baehr, McKeesport, Penn. 972,798.

SPEED-REGULATING MECHANISM. John C. Carpenter, Houston Heights, Texas. 972,815.

DEVICE FOR INCREASING THE DRAFT OF CHIMNEYS AND FOR OBTAINING DRAFT WITH BAD CHIMNEYS. Guillaume Geessels, Brussels, Belgium. 972,847.

FEED-WATER HEATER AND PURIFIER. Thomas J. Cookson, Cincinnati, Ohio. 972,824.

HOSE COUPLING. John I. Creveling, New York, N. Y., assignor to Safety Car Heating and Lighting Company, a Corporation of New Jersey. 972,829.

PRESSURE-REGULATED VALVE. Alva Thrift Edmonson, Chicago, Ill. 973,016.

VALVE GEAR FOR ENGINES. Charles F. Prescott, Philadelphia, Penn. 973,137.

CARBURETER VALVE. Nicholas P. Mader, Sun Prairie, Wis. 973,056.

MIXER. Nicholas P. Mader, Sun Prairie, Wis. 973,057.

DEVICE FOR CLEANING BOILER CROWN SHEETS. William D. Randall, Monett, Mo. 973,059.

GAGE COCK. Jamieson V. Scaife, McComb, Miss. 973,074.

VALVE. Oliver Schlemmer, Cincinnati, Ohio. 973,075.

FLUID-PRESSURE GOVERNOR. George C. Hicks, Jr., and Ralph C. Enyart, Connersville, Ind., assignors to the P. H. & F. M. Roots Company, Connersville, Ind. 973,197.

FLEXIBLE METALLIC TUBING. Henry Tideman, Menominee, Mich., assignor of one-half to James Wolff, Chicago, Ill. 973,238.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

A Homemade Planimeter

By Howard H. Bliss

A planimeter which can be constructed with a few pieces of wood, a small steel wheel and a broken thermometer tube. A demonstration of the principle upon which it is based.

Many cases arise in the experience of people connected with engineering in which they find it desirable to know the area of irregular figures, such as indicator diagrams from a steam or gas engine, or maps of irregular pieces of ground, contour lines, etc. These areas can often be measured roughly by dividing them into triangles and parallelograms, measuring the dimensions of these and computing their size. However, in order to get an accurate measurement of irregular figures it is necessary to make use of an instrument called the planimeter.

Planimeters can be bought at prices ranging from ten or fifteen dollars upward. Anyone who is handy with tools can construct one which will cost nothing but the labor of making it. The writer owns one made in spare minutes which does the work required with an error of less than 1 per cent. This instrument is of an older type than the planimeters sold now, but it works practically as well and was considerably easier to construct. It is made chiefly of wood. Fig. 1 shows the general construction. At Q is a point designed to stick into the paper or drawing board and hold that end of the arm $Q\,Y$ so that he arm can sweep around in a circle. At Y is a block of wood held to the arm $Q\,Y$ by a screw pointing upward which turns freely in the arm and allows the block nd the rest of the planimeter to swing bout the screw as an axis. The rod R a glass thermometer tube whose graduions are cut into the glass. Upon this ides the steel wheel W, which is shown detail in Fig. 2 and which has a fairly sharp edge so that it does not slip over the paper when R is drawn through it by the motion of the instrument. The wooden stick $Y\,T$, called the "tracing arm," fits rather tightly into a groove in the top of the block Y. This groove is at right angles to the glass rod R, which fits into a hole bored through the block about half way between the top and bottom. At T is the "tracer," simply a nail with its point smoothed, which reaches down to the level of the paper, and which, when the planimeter is in use, is made to follow the outline of the diagram to be measured.

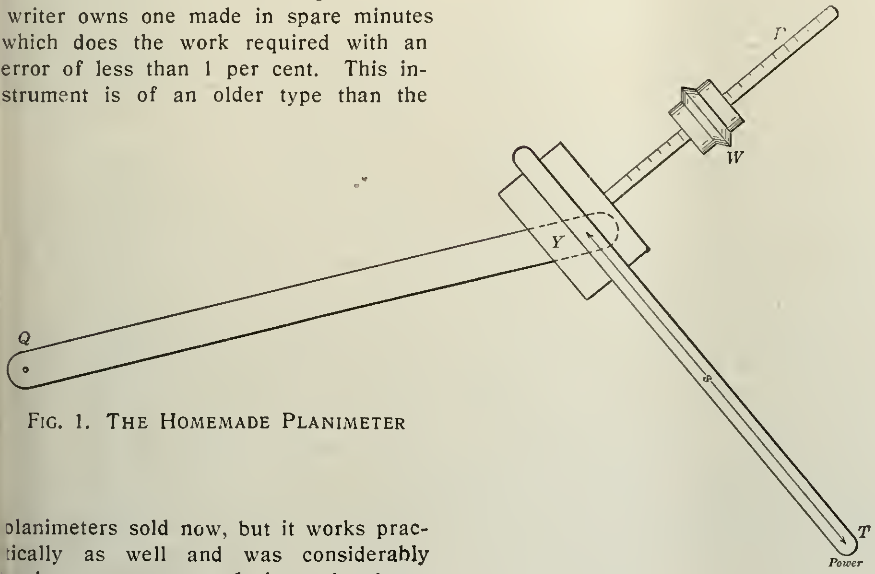


Fig. 1. The Homemade Planimeter

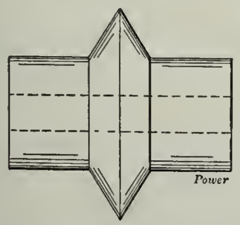


Fig. 2. The Tracing Wheel

The whole operation of measurement is extremely simple. The point Q is stuck into the paper at some convenient place, the wheel W is set by hand at the zero mark on the thermometer tube, the point T is run around the figure in a clockwise direction, finishing at the starting point, and then the area of the drawing is read off as the distance the wheel has moved on rod R, each scratch on the glass being read as a square inch.

The proofs and explanation of the action of the planimeter as given in textbooks either depend upon the calculus or are made very long and intricate in the effort to escape higher mathematics. The proof given below, however, is quite simple, and to understand it does not require any mathematical knowledge beyond arithmetic. It is practically the proof offered by the calculus with the technical parts simplified.

In Fig. 3 consider the tracing arm as detached from the rest of the instrument and occupying the position $T\,Y$. Let the arm be moved to the position $T'\,Y'$, the tracer T following all the time the lower line of the figure A, while the other end follows the arc $Y\,Y'$. In its travel the whole arm sweeps over the area $A + B$ (that is, it moves over every spot in both of the spaces A and B), swinging about to accommodate the curves of the figure, as shown by the dash-and-dot line. Then, let the arm move back to its first place, keeping the tracer now on the upper side of the figure A and the other end still on the circular arc. Thus it sweeps over the area B, but in the opposite direction; let us call it negatively. Then, during the whole trip, the resulting positive area swept over is simply A, for B is canceled by going back over it in the opposite way.

Having proved that the positive area swept over by the arm in its travel is the same as the area of the figure followed by the tracer, we have next to consider how we may measure what area is swept over by a stick moving sidewise, as this has done. This is best got at by considering a very small portion of its motion, as shown in Fig. 4, from a position $Y_1\,T_1$ to another $Y_3\,T_4$. Assuming

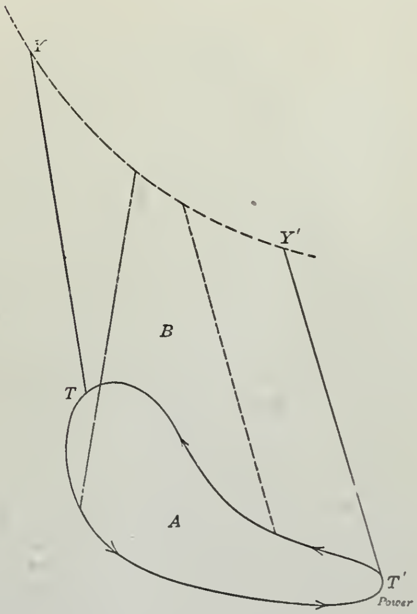


Fig. 3. Showing the Travel of the Tracing Arm

that the arm travels straight over to $Y_2 T_2$, slips lengthwise to $Y_3 T_3$, and swings to its last position, the small area swept over is the sum of the little rectangle (whose size is $s\,x$) and the "pie-shape" piece adjoining. The whole area covered by any motion of the arm will be the sum of an infinite number of these microscopic rectangles and "pie-shape" pieces.

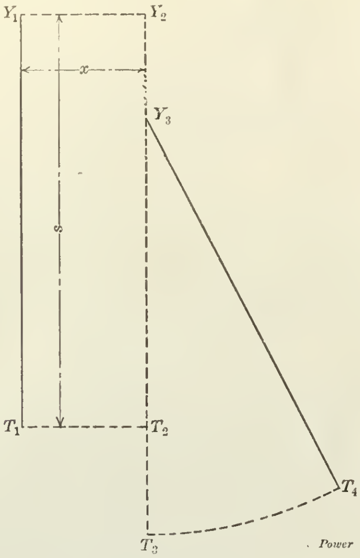


FIG. 4. DIAGRAM OF METHOD OF PLANIMETER OPERATION

The fact that the actual motion from $Y_1 T_1$ to $Y_3 T_4$ may not have occurred by the three simple shifts imagined is of no importance, since the positions may be taken infinitely close to each other and hence any error involved would reduce to zero. Now, the sum of all of the little rectangles will be the area obtained by multiplying s, the length of the tracing arm, by the sum of all the distances the arm has moved perpendicular to itself. The glass rod mentioned in the description will have slipped just that distance through the little wheel, slipping to the right when the tracing arm moves in that, the positive, direction, and to the left when the arm moves backward in the negative direction. It should be noted that the swinging of the tracing arm over the small "pie-shape" pieces will not cause any slip of the glass rod through the wheel. However, the sum of these pieces will be zero, for just as many are made with the arm swinging backward as forward, and hence the negative ones will balance the positive ones by the time the arm gets back to its original position, for, of course, the arm must swing just as much backward as forward in order to point in the same direction when it gets through as when it started.

It has been proved, first, that the total positive area swept over by the arm in its journey around and back to the starting point is A, and, second, that the area swept over is the product of the length of the arm and the total distance which it has moved perpendicular to itself, or the distance the rod has slipped through its wheel. Hence, A, the area of the figure around which the tracer has run, will be equal to the reading on the glass rod multiplied by the length of the tracing arm. By properly adjusting the length of the tracing arm the divisions of the glass rod may be made equivalent to the area passed over; if the marks are ⅛ inch apart, a tracing arm 8 inches long will give the result that a reading of three divisions on the glass will mean an area of three square inches passed around by the tracer.

In all that has preceded it has been assumed that the other end Y of the tracing arm has gone forward and back along its circular path. That is the fact in almost all cases where the planimeter is used, such as in scaling off indicator diagrams and similar small areas. But, there come times when it is necessary to measure such large areas that the tracer cannot reach all points on the boundary at one set up unless the fixed point Q is placed within the figure, and that will result in the travel of Y around a circle to its starting point, without going backward to it. If there is a right angle between $Q\,Y$ and $Y\,T$, Fig. 1, all the way round, which means that T will trace a circle around Q, there will be no motion of the wheel along the rod. The circle so traced by T is called the "zero circle."

To measure an area larger than the zero circle, as shown in Fig. 5, the planimeter is set with point Q near the middle of the figure. Suppose the tracer to start at G on the boundary and move around, as shown by the arrows, to the point J, which is very close to G. The reading of the instrument at this point can be proved to be the area included between the outer border and the zero circle (shown dotted), thus: If the tracer were brought toward the center along the line $J\,K$, a slight change would be made in the reading of the planimeter. Then if the tracer followed the zero circle around according to the arrows to point M, no change would be made in the reading. Moving from M to G would be practically the same as going back along $K\,J$, and the change in the reading would cancel the change brought about by the motion from J to K. The outline of the figure $G\,Z\,J\,K\,R\,M\,G$ would then be traced and the point Y would have returned to its original place along the path it followed in the first part of its journey. Hence, the reading on the recording apparatus would be this area, and it would be the same reading that was obtained at J. Of course, the distance between J and G need not be more than a millionth of a hair's breadth, which means that we really run the tracer clear around to the starting point, in practice.

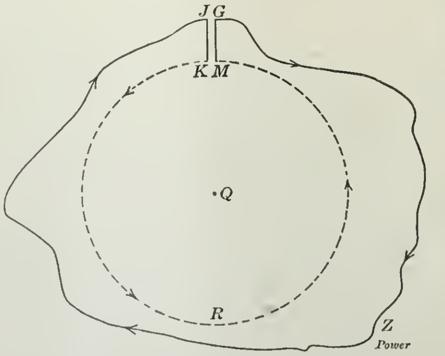


FIG. 5. DIAGRAM OF AREA LARGER THAN THE ZERO CIRCLE

Therefore, to find the area of any large figure, add the area of the zero circle to the reading of the recording mechanism. It sometimes happens that part or all of the boundary of the large figure lies within the zero circle, but it can be shown that every time the reading is to be added to the area of the zero circle, even though the reading should be negative, which will be the case when the area is smaller than the zero circle.

To find the area of the zero circle of any planimeter all that need be done is to trace out a circle larger than the zero circle and subtract from the computed area of the large figure the area read on the recorder. The easiest way to trace such a circle is to punch two holes through a narrow strip of paper near the ends, stick the pin at Q through one hole into the drawing board and stick the tracer through the other. By steadily pulling outward on the tracer the paper is kept tight and a perfect circle may be run around without the trouble of following a line with the tracing point.

Planimeters of more modern types dispense with the rod R, Fig. 1, and its sharp-edged wheel, substituting a differently shaped wheel placed under or beside the tracing arm with its spindle parallel to the arm. The distance moved by the arm parallel to itself is equal to the amount which the wheel rolls, and this is read off by means of marks cut into the rim of the wheel and a vernier or pointer attached to the arm. A disk geared to the wheel keeps count of the number of revolutions. The principle is exactly the same as that upon which the homely wooden instrument, here described, is based, for the sidewise travel of the arm is registered and the endwise motion causes no change in the reading.

A modification of this later type is the Coffin planimeter, which is much used in measuring indicator diagrams. The arm $Q\,Y$ is left off and the end Y of the tracing arm is guided in a straight line by a pin which follows a groove in the drawing board. The principle is still the same, for the fact that Y travels in a straight line instead of an arc makes no difference in the proof.

Power
and The Engineer

Vol. 32 NEW YORK, NOVEMBER 8, 1910 No. 45

In writing upon a certain subject a correspondent says that "after a thorough discussion of the subject all parties concerned will have the same opinion as before." Is this a true statement of the frame of mind of the majority of engineers?

Is it true that instead of searching after the truth in the matter they seek rather to find reasons for believing that which may or may not be true, is true, because they wish it to be true?

It is true that thinking is hard work and it is also true that a new thought or a new idea almost invariably strikes the mind unfavorably. Steps from the beaten path of conventionality are taken reluctantly. But are engineers less progressive than other men?

Centuries ago someone said, "Can any good come out of Nazareth?" It was prejudice that asked the question that carried with it the implication that only an answer in the negative could be given.

Does prejudice dictate the position which engineers take on questions of policy or practice in their vocation?

If one is unwilling o put himself in another's place, to see as he sees, to get a view from the same position, to give full credit to ideas differing from his own, prejudice is the main-pring of his logic.

Are engineers whose aily contact with conditions and problems are always of such a practical nature, less able to take a practical view of any problem in their line than men in other walks in life?

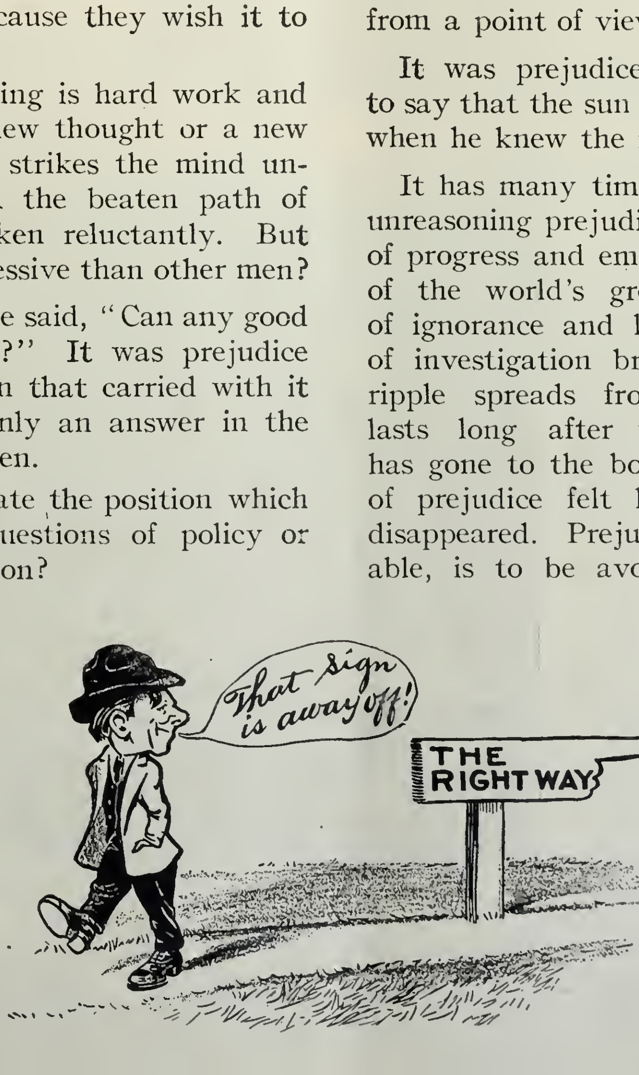
In the discussion of any topic where full credit is not given to every scrap of evidence no conclusion can be reached which is entitled to confidence of any one, and least of all to the confidence of him who refuses to examine from a point of view other than his own.

It was prejudice that compelled Galileo to say that the sun revolved around the earth when he knew the reverse to be the truth.

It has many times been shown that blind unreasoning prejudice has clogged the wheels of progress and embittered the lives of some of the world's greatest men. It is born of ignorance and lives only until the light of investigation brings knowledge. As the ripple spreads from shore to shore and lasts long after the stone that made it has gone to the bottom, so is the influence of prejudice felt long after its cause has disappeared. Prejudice, favorable or unfavorable, is to be avoided like the pestilence, for it always leads to the wrong road and to the retracing of numberless wasted steps.

Engineers, above all men, should be free from this character-destroying influence of prejudice.

"Prove all things."


That sign is away off!
THE RIGHT WAY

Central Station at Newcastle, Indiana

By Osborn Monnett

In this small central power station of only 700 kilowatts capacity, the exhaust steam is used to supply a hot-water heating system to neighboring customers. The main features of the plant are its simplicity and its adaptability to the existing requirements.

It is not always necessary to go to the large metropolitan power plants in order to find interesting things in steam engineering; frequently the smaller plants show the result of just as much careful forethought as any of the large and more prominent installations. Whether in a large or small plant the best solution of any engineering problem is that which will realize the largest returns upon the investment under the existing conditions.

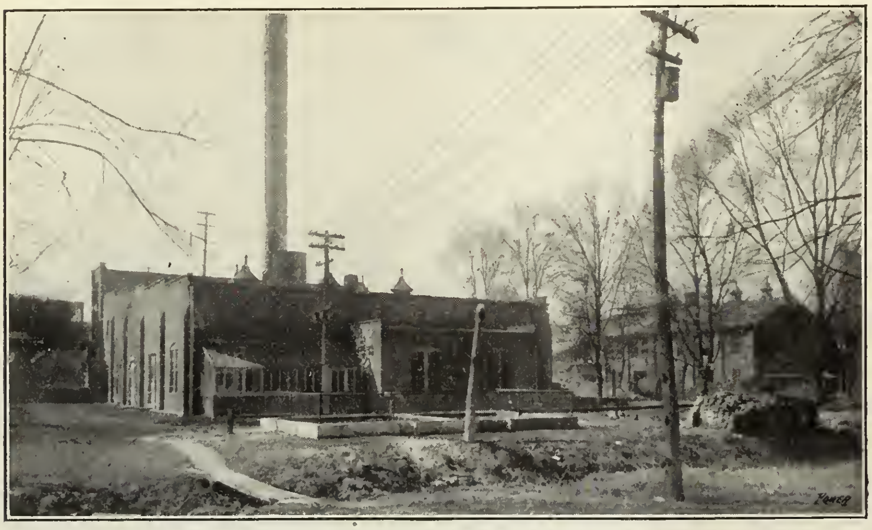


Fig. 1. Exterior View of Plant

The plant of the Newcastle Light, Heat and Power Company is a small central station containing compound-noncondensing engines; for during the greater part of the year the exhaust steam can be more profitably utilized for heating than by exhausting into a condenser. The location of the plant is such as to simplify the coal-handling problem, it being placed adjacent to the elevated tracks of the Pennsylvania railroad, a spur of which runs directly over the coal bunkers at an approximate hight of 16 feet above the boiler-room floor. Fig. 1 is an exterior view of the power house, showing a carload of coal on the elevated track. From the bottoms of the dump cars the coal falls into a storage pocket of 400 tons capacity, from which point it descends, partly by gravity, into hand cars at the boiler-room floor. These cars each have a capacity of 1800 pounds of coal and every load is weighed on a platform scale before being delivered to the boilers. Fig. 2 is a view of the boiler room, showing the hand cars standing against the coal-pocket bulkhead.

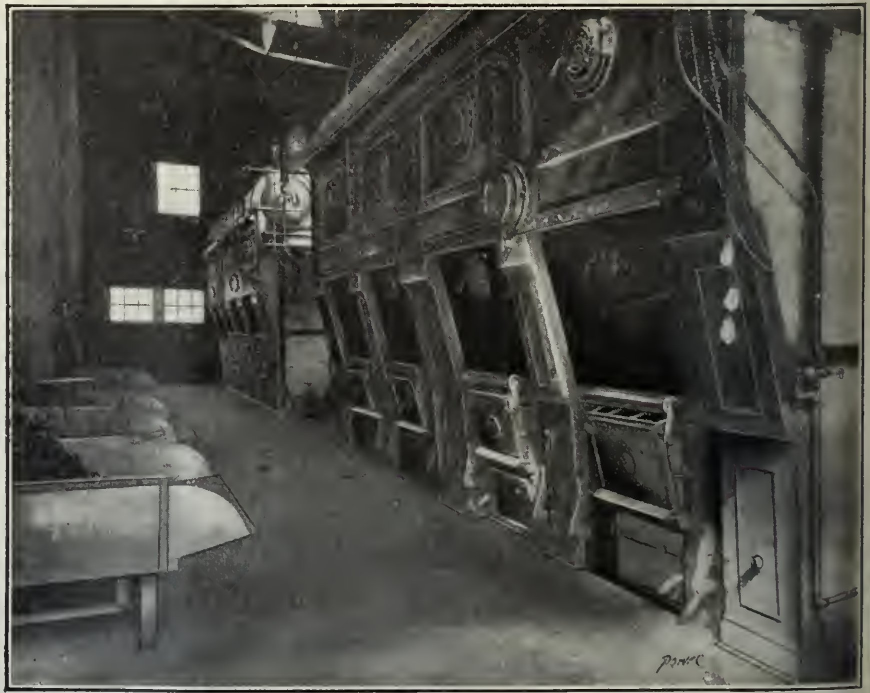


Fig. 2. Boiler Room

The plant, as shown in the sectional elevation, Fig. 3, consists of four separate compartments, the coal pocket, the boiler room, the pump room and the engine room, arranged in the order named. Four 250-horsepower hand-fired Stirling boilers are installed, discharging the products of combustion into an underground flue and thence to a steel stack 6 feet in diameter and 150 feet high, lined to the roof of the boiler room with firebrick. Two of the boilers are equipped for burning natural gas, if desired, in addition to using coal for fuel.

Six-inch long radius bends lead from the boilers to a 10-inch header situated in the pump room, from which branches extend to the engines. The header contains three drip pockets for catching condensation, and in addition there is a separator on each engine line, the condensed steam being taken care of by Morehead tilting traps and returned to the heater.

In the engine room there are three generating units: two 200-kilowatt cross-compound Russell engines, each direct connected to a two-phase, 60-cycle, 2300-volt Western Electric generator, to which is belted an exciter; and a third Russell engine and generator of 300 kilowatts capacity. Each engine has an independent gravity oiling system which is shown in the view of the engine room, Fig. 4. The electrical load consists of both lighting and power circuits, the distributing circuits being at 2300 volts, which is stepped down to 220 volts for factory lighting and power and to 110 volts for the lighting of private residences. A satisfactory power load has been developed which, with the night lighting load, enables the plant to run continuously 24 hours a day.

One of the interesting features of the electrical equipment is the location of the switchboard. This consists of blue Vermont marble panels carrying the usual switches and recording instruments, but the panels are set flush with the engine-room wall so that the board takes up no room in the engine room proper, although it is accessible from the back, which opens into the electrical-repair department and meter-testing room, where all of the switchboard connections are easily available for inspection or repairs.

A point in which this plant differs from the ordinary small central station is in the hot-water heating system by which practically all of the exhaust steam is utilized for profit. Therefore, the pump room in which this part of the apparatus is located is of considerable interest. A view of the pump room is shown in Fig. 5. The question of water supply is one of importance in this class of plant as the system must be kept fully charged t all times to enable the full benefit o be realized from the heating mains. hree driven wells, 4 inches in diamter and 43 feet deep, furnish all the ecessary water. The water in the wells sually stands at a level of about 7 feet elow the ground and by means of an ir lift it is raised and discharged into a oncrete reservoir of 18,000 gallons capaity. This reservoir, which is shown in e foreground in Fig. 1, is built in two ompartments and stands a few feet above e level of a creek which runs through e property; hence, either compartment ay be drained by gravity and cleaned hile the other compartment is in use. his makes it possible to use chemicals for treating the water before it enters the system, if such a practice is found desirable.

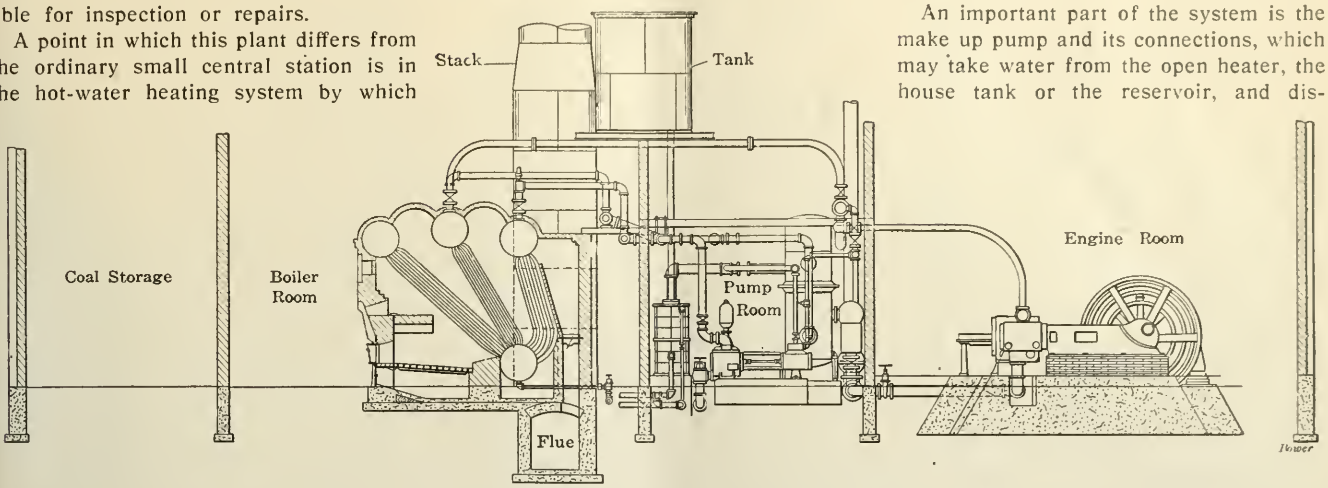


Fig. 3. Sectional Elevation through Plant

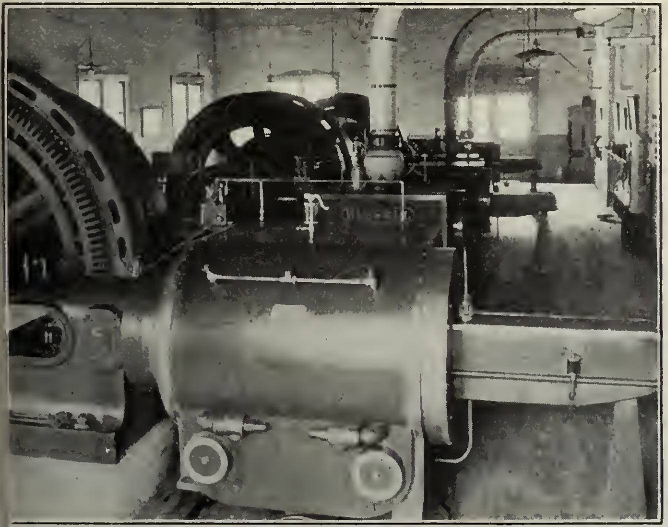


Fig. 4. Engine Room

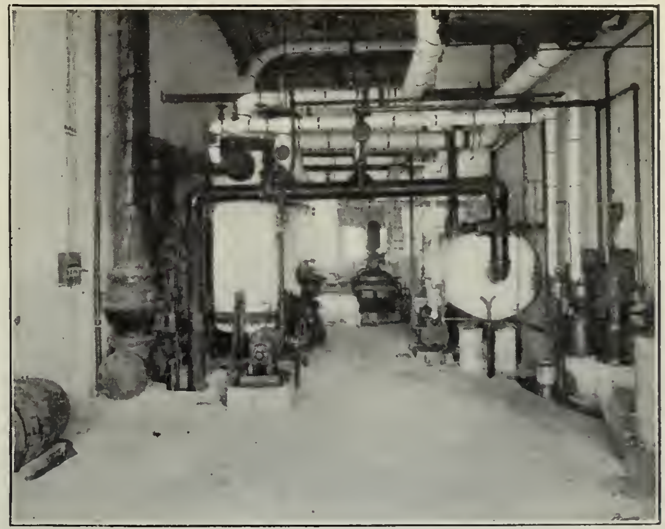


Fig. 5. Pump Room

From this reservoir the water is pumped to a 4000-gallon steel tank situated on the roof of the power house and giving a head of about 35 feet, from which the water is fed automatically to a 500-horsepower Hoppes feed-water heater located in the pump room. The boilers are fed by a 10 and 6 by 12-inch Dean pump, and two 7½ and 5 by 6-inch Blake pumps. Two separate and distinct feed lines are arranged, enabling one to be entirely taken down for repairs while the other is being used.

The exhaust steam from the engines and also from all pumps is collected in one large exhaust main and passes through either one of two Baragwanath heaters through which is circulated the water for the heating system. The water leaves the power house through an 8-inch main, provided with expansion joints, wrapped with mineral wool and inclosed in wooden conduits about three feet below the surface. The pressure is kept up on the system by two 12 and 10 by 18-inch Stillwell-Bierce pumps, the outgoing pressure being about 60 pounds and the incoming pressure 30 pounds. Under ordinary conditions the water loses from 20 to 30 degrees in making the circuit.

An important part of the system is the make up pump and its connections, which may take water from the open heater, the house tank or the reservoir, and discharge it into the suction of the circulating pumps. This pump, as well as all other pumps in the power house, is controlled by a Carr pump governor and works automatically.

In addition to the heating equipment described, provision is made so that one of the boilers may be connected in series in the hot-water line by which means can be supplied any necessary additional heat in excess of that supplied by the exhaust steam. It has been found that in ordinary weather, practically all of the exhaust steam is condensed in the heaters, thereby giving up all of its energy for useful work. The exhaust line is well protected by oil extractors so that the condensed steam may be used for feed water.

The plant was installed by the Arbuckle-Ryan Company, of Toledo, and by its simple design it is well adapted to the particular service, whereas a more complicated and costly equipment would not have solved the problem as well.

The Steam Turbine in Germany

By F. E. Junge and
E. Heinrich

Increase in use of steam and water power in Prussia, Switzerland and the United States. Increase in economy of the steam engine and factors influencing development. Three principal types of axial turbines and the tendency to merge into one standard design embracing the desirable features of each.

THE EVOLUTION OF POWER GENERATION

The per capita production of power in a certain country affords a fair measure for the condition of power generation, technical progress and for the industrial ability of the people inhabiting it. The diagram in Fig. 1, which, like the others in this article, is taken from a paper communicated to the Society of German Engineers by Professor Kammerer, Charlottenburg, shows that the curve of steam-power generation in Prussia is almost a straight line, rising from 0.055 horsepower per capita in 1890 to about 0.14 horsepower in 1907. (In the following, metric horsepower is used exclusively. It should be remembered that one metric horsepower equals 0.986 English horsepower.) The steam-power curves for the United States and Switzerland, which are given for comparison, show similar tendencies, while the development of water power is much more marked in Switzerland than in the States, for the simple reason that the cost of fuel in the latter country is comparatively low, while the former is absolutely dependent on her water-power development for industrial production.

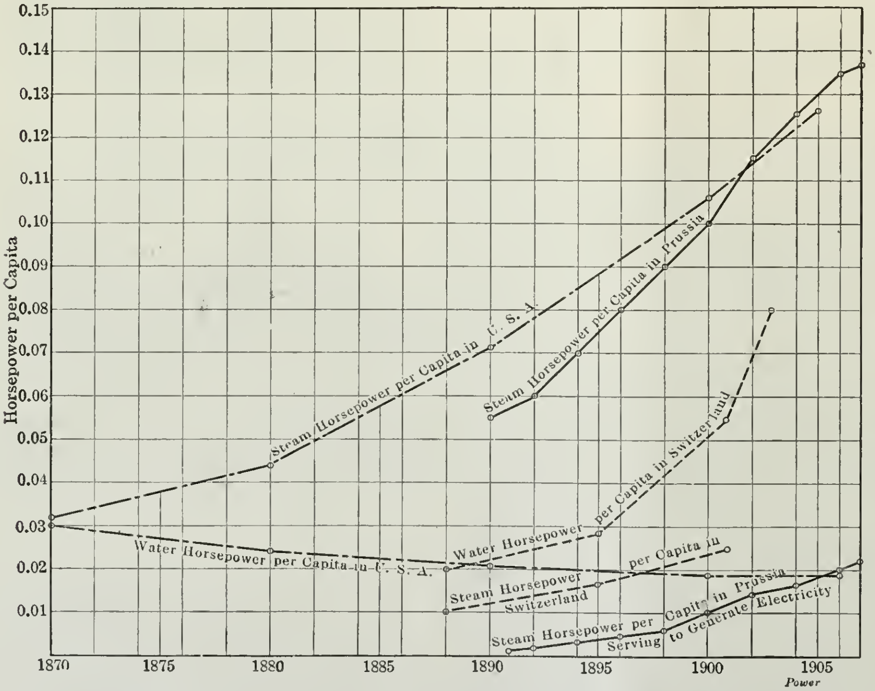


FIG. 1. INCREASE OF STEAM- AND WATER-POWER GENERATION IN PRUSSIA, SWITZERLAND AND THE UNITED STATES

The lowest curve in the diagram shows what portion of steam power in Prussia is transformed into electric energy. This curve, too, is in the ascent, revealing the growing tendency in that country to effect the distribution of power by means of electric energy instead of by individual generating plants. As a matter of fact, we are coming more and more to the point where power will be generated at the mouth of pit or other convenient places, using inferior grades of coal unfit for transportation, whence it is distributed in a high-tension net covering and supplying the whole country. The figures relating to power production in Prussia are taken from the latest official statistics, but they refer to the condition of plant when the latter is for the first time put into practical operation. As the load factor increases with the growing demand for power, the plant output rises in proportion. Hence, the capacity of power production is in reality considerably higher than the curve would imply.

Technical and commercial progress is illustrated further by the improvements in the *efficiency* of energy transformation. When making comparisons, however, we must embrace in our conclusions a consideration of the differences in the geographical structure of the different countries, which will make the attainment of highest economic excellence less desirable, for instance, in industrial circles in the United States, as long as coal, oil and natural gas are available in sufficiency, than in the industries in Germany, where natural limitations and a dense population force the engineer to strive for the utmost utilization of meager resources. All points considered, we may say, briefly, that the success of the German industry on the world's markets rests on the principle: get quality, rather than on the demand: get tonnage.

The diagram in Fig. 2 shows the actual consumption of heat in steam engines from the year 1860 up to 1908. In the first-named year the consumption per horsepower-hour was over 11,000 calories or 43,560 B.t.u. (single-cylinder engine working at 2.5 atmospheres or 36.7 pounds per square inch boiler pressure). By raising the steam pressure to 5.5 atmospheres the heat consumption of a single-cylinder engine was reduced to 7600 calories or 30,096 B.t.u. per horsepower per hour in 1874. The marked reduction of heat consumption to 6400 calories or 25,344 B.t.u. was effected in the same year by the introduction of the compound engine. In the following years the heat consumption per horsepower-hour decreased gradually to 5000 calories or 19,800 B.t.u., while the boiler pressure at the same time increased to 8 atmospheres or 116.8 pounds per square inch.

A further reduction was effected by the advent of the triple-expansion engine in 1890, with a corresponding rise in steam pressure to 11 atmospheres or 161.7 pounds per square inch. During all of that time the temperature of the steam had been hardly changed because saturated steam was used almost exclusively up to 1890. It was only on account of the difficulties experienced when trying to raise the steam pressure beyond 13 atmospheres or 191 pounds that the superheating of the steam was first attempted. The result was a further reduction of heat consumption to 3300 calories or 13,068 B.t.u. per horsepower per hour, attained with a triple-expansion engine of 2000 horsepower in Görlitz. This, it seems, represents now the upper limit of steam temperatures for practical operations. Attempts to go still higher have failed on account of the difficulty of constructing durable superheaters.

At this point, when the development of the reciprocating engine seemed to come to a halt, steam-power generation received a new impetus by the development of the steam turbine. While the earlier types of this prime mover showed a higher heat consumption than the best steam engines, on account of the low temperatures and pressures originally employed, it has been possible of late to bring the thermal efficiency of steam turbines to the same level of excellence attained by their competitor. Regarding heat consumption, both types are now practically on a par. But the steam turbine has brought about economies in a new direction: lowering the cost of buildings, floor space, attendance and lubrication.

The diagram in Fig. 2 contains, besides the curves showing actual steam consumption, boiler pressure and steam temperature, a curve giving the theoretical heat consumption of an ideal engine, working with the same initial temperatures as the rest. The ordinates between these two curves represent the loss accruing through imperfections in the engine, said loss having been reduced from 9000 calories or 35,640 B.t.u. in 1860 to 2000 calories or 7920 B.t.u. in 1908. The straight line running parallel to the abscissa axis of the diagram represents the heat consumption of an ideal engine working with an initial temperature of 1000 degrees Centigrade. The ordinate fragments between this line and the curve of theoretical heat consumption represent the loss due to the employment of low temperatures. Thus, the diagram in Fig. 2 gives a graphic history of the development of steam-power generation as far as heat consumption is concerned. It shows that inventive talent will seek and find new outlets if natural obstacles hamper progress in the direction originally pursued. It shows further that the art of steam-power generation is still far from perfect and that its practioners will have to concentrate their best efforts in order to compete successfully with other factors such as gas, oil and water power, none of which has reached its highest level of excellence.

The diagram in Fig. 4 shows in a similar way the development of the reciprocating steam engine for ship propulsion from 1850 to 1890, according to tests made on the steamers of the two largest German shipping companies: The Hamburg-American Line and The North German Lloyd.

The influence of the size of engine on the cost of steam power is shown in the diagram in Fig. 3. Experience has taught the designer that large engines can stand a careful working out of details much better than small engines, without considerably increasing first cost. Thus the upper limit of all-round economy is greatly raised by simply enlarging the capacity of heat engines. The diagram in Fig. 5 shows the effect of prime-mover capacity on the factor of heat consumption. The latter decreases and economy increases with growth in the size of the plant. Some of the later thermotechnical improvements, for instance superheating in steam engines and increasing compression pressures in gas engines, have benefited small and large engines alike. But large-scale power generation has the great advantages of lower initial cost and lower cost of attendance as well as cheaper supply of fuel.

The operating cost of heat engines was reduced among other things by the adoption of automatic lubrication. With steam turbines, forced lubrication was a necessity in order to supply oil to the bearings under pressure. But, in reciprocating engines, ring and central lubrication have been adopted with an equal advantage. The cost of attendance was further greatly reduced by the introduction of automatic stoking. In all improvements which regard the reduction of power cost by the elimination of the personal element or, generally speaking, in the replacement of human labor bv mechanical means, American industry has been our teacher, while in all cases which regard the raising of economy by the rational utilization of matter, of fuel and other raw materials the German industry has been in the lead. In the one case scarcity and high cost of skilled labor, in the other scarcity and governmental control of natural resources were responsible for the results attained, an example which seems to prove that technical progress does not travel along the path of least resistance, but attains highest excellence when it is compelled to overcome obstacles.

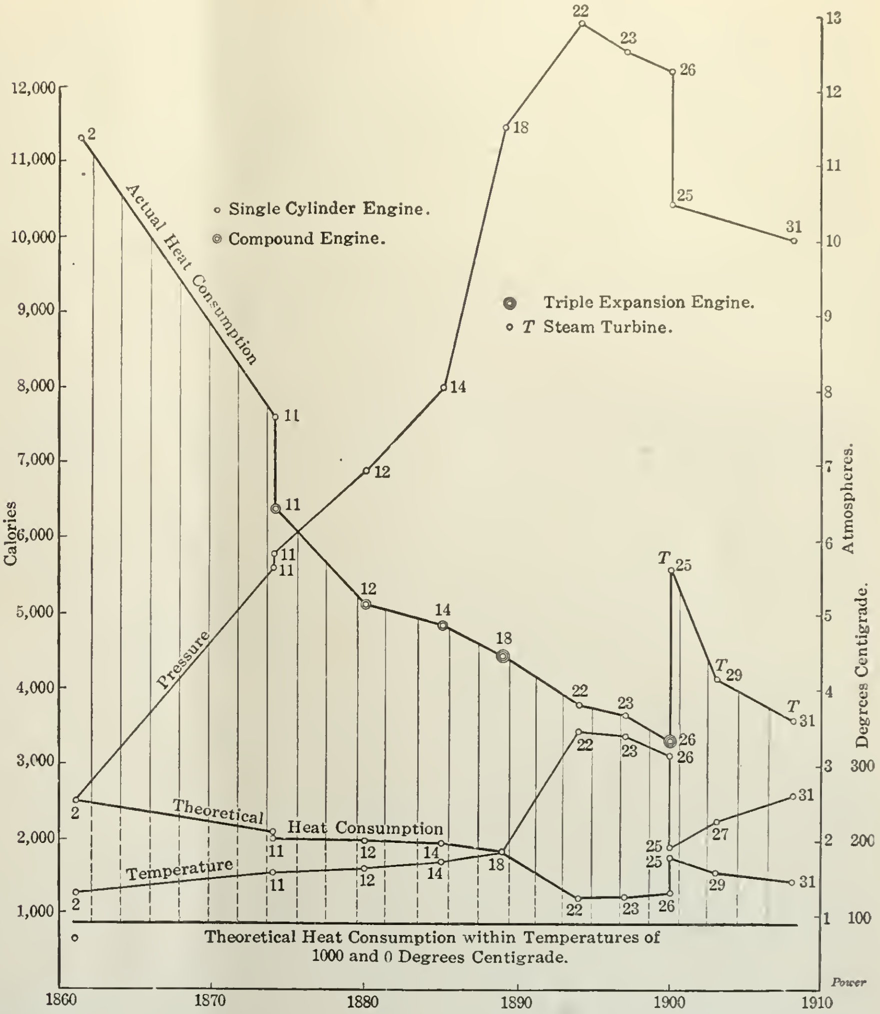


FIG. 2. HEAT CONSUMPTION OF STEAM ENGINES AND TURBINES FROM 1860 TO 1908

No. 2. Single-cylinder marine engine from 1861, 204 h.p., 17 r.p.m.
No. 4. Single-cylinder marine engine from 1874, 282 h.p., 62 r.p.m.
No. 11. Compound marine engine from 1874, 267 h.p., 71 r.p.m.
No. 12. Compound stationary engine from 1880, 132 h.p., 71 r.p.m.
No. 14. Compound balance pumps from 1883, 251 h.p., 13 r.p.m.
No. 18. Triple-expansion engine from 1889, 200 h.p.
No. 22. Triple-expansion engine from 1895, 1042 h.p., 60 r.p.m.
No. 23. Compound Schmidt engine from 1899, 100 h.p., 81 r.p.m.
No. 25. Steam turbine from 1900, 1000 kw., 1500 r.p.m.
No. 26. Triple-expansion engine from 1900, 2000 h.p., 83 r.p.m.
No. 29. Steam turbine from 1903, 1000 kw., 1500 r.p.m.
No. 31. Steam turbine from 1908, 3500 kw., 1360 r.p.m.

The diagram in Fig. 6 shows the economic result in a boiler plant of the adoption of coal conveyers and automatic stoking. Formerly there were required in this plant 54 firemen and 2 overseers, necessitating an outlay in wages of 0.164 mark or 3.9 cents per ton of steam. After the introduction of mechanical means only 20 firemen, 2 overseers and

2 machinists were required. The number of high-class workmen was doubled, while the number of unskilled laborers was reduced in the proportion of 2.5 to 1. The amount of wages paid decreased to 0.061 mark or 1.45 cents per ton of steam. To this must be added the cost of upkeep, interest and amortization for chain grates and conveyer bands amounting to 0.034 mark, so that the total cost was reduced to 0.095 mark or 2.3 cents, which is almost two-thirds of the original amount. This saving was effected by employing, instead of unskilled labor, automatic machinery and high-class labor. Of course, owing to the somewhat cheaper and more reliable labor which we have in Germany, the replacing by mechanical means of the human element in industrial plants is not always so profitable as it is in the States. But the general tendency of power generation is the same in our country as it is in America: concentration of production in large plants in order to secure utmost commercial economy.

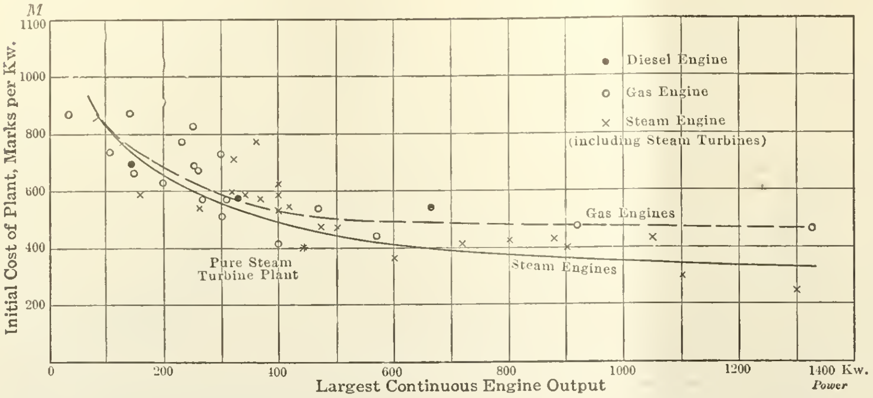


FIG. 3. RELATION OF INITIAL COST OF STEAM ENGINES AND TURBINES TO SIZE OF PLANT

DEVELOPMENT OF THE STEAM TURBINE

In Germany the steam turbine had to compete with a highly developed steam engine as well as with the gas engine, whereby its coming has been somewhat delayed, and even today it has to meet the growing competition, especially for ship propulsion, of a dangerous rival: the Diesel oil engine. Not until 1898 was the first large steam turbine, a 1000-kilowatt Parsons turbine, erected in the electric central station of Elberfeld, where it has given complete satisfaction. The following rapid evolution is notable for the desire on the part of manufacturers to attain highest possible fuel economy rather than simplicity of construction and operation. The result of this endeavor is best illustrated by the fact that a consumption of 5.4 kilograms or 11.9 pounds of steam per kilowatt-hour is attained with a 4000-kilowatt steam turbine installed by the Allgemeine Elektrizitäts Gesellschaft in the central station at Rummelsburg, Germany.

It is an acknowledged fact that the development of steam-turbine construction, after having acquired a high degree of thermal excellence, tends now toward a standardization of the different systems, which at present are dominated and characterized by a few simple principles of steam flow and structure. The following discussion of these principles will be confined merely to axial turbines, because the radial turbine has not so far shown an equal capability of development, at least for purposes of large-scale power production.

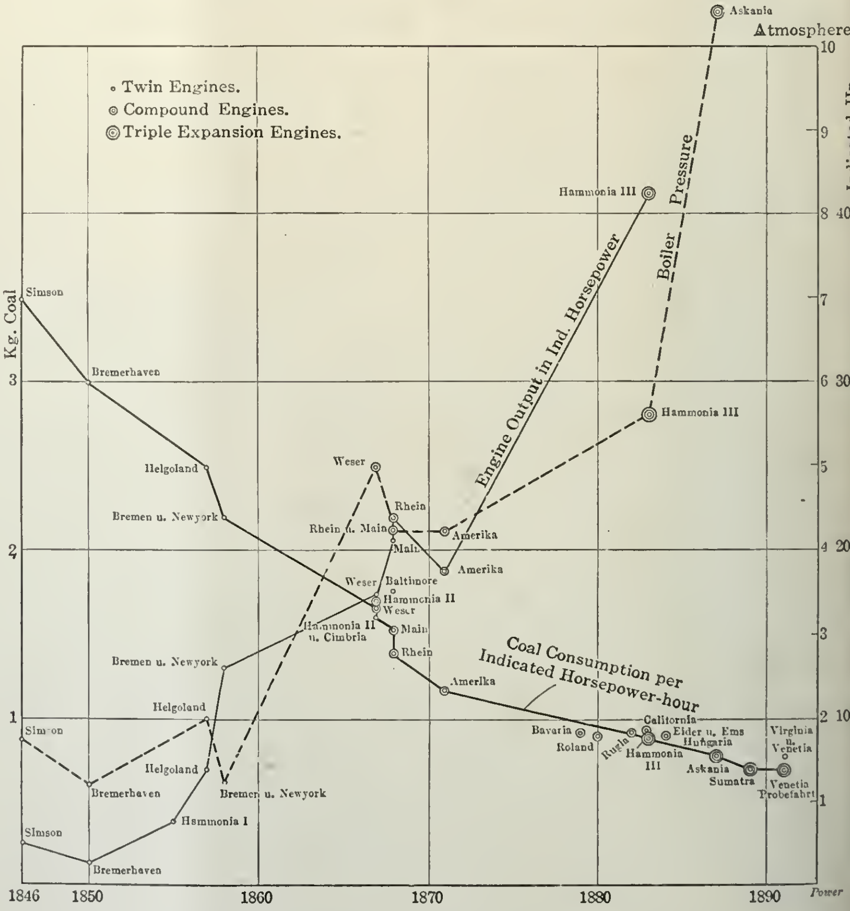


FIG. 4. COAL CONSUMPTION OF NORTH GERMAN LLOYD AND HAMBURG-AMERICAN LINE STEAMERS FROM 1850 TO 1890

First. The many-stage reaction principle developed by C. A. Parsons. As regards the utilization of steam energy this system is characterized by the fact that the conversion of the potential energy of the steam into kinetic energy takes place not only in the guide-blade channels, but also in those of the rotating part. Regarding construction, the Parsons system is characterized by the use of a drum with dummy pistons and labyrinth packing.

Second. The many-stage action principle with one row of blades on each rotating wheel, developed by Rateau and modified by Zoelly. As regards the utilization of steam energy this system is characterized by the fact that the conversion of the potential steam energy into kinetic energy takes place in the guiding apparatus only, while the whole kinetic energy of a stage is utilized in a single row of rotating blades. Regarding construction, the Rateau-Zoelly system is characterized by the employment of several disk wheels and by the separation of the different stages through guide-wheel disks, which reach unto the hubs of the rotating wheels.

Third. The many-stage action principle with velocity staging developed by Curtis. As regards the utilization of steam energy, this system is characterized by the fact that the whole potential energy of one stage is, as a rule, transposed into kinetic energy in expanding nozzles, and further, that the kinetic energy of each stage is utilized in several rows of revolving blades. Regarding construction, the Curtis system possesses rotating disk wheels, like No. 2, but fewer in number, which are equipped with several rows of blades; in addition, it possesses deviating blades which are fixed in the casing. (Some designers do not provide for the complete expansion of the steam in the nozzle, but transpose part of the potential energy in the first row of revolving blades in order to attain higher efficiency. But this feature is not generally practised and the amount of steam converted is quite small, so that the above classification is approximately correct.)

The principle of one-stage action has not been included in the above. This system, as developed by De Laval, shows in its practical execution several remarkable solutions; for instance, the flexible shaft and the disk wheel with high peripheral velocity. But the very nature of the De Laval turbine, at least in its original form, militates against the attainment of higher capacities. The high peripheral velocity and the resulting high number of revolutions necessitate the employment of gears in order to get down to practicable speeds. Gears for high output have not, however, found favor in stationary practice. In the Riedler-Stumpf turbine the attempt was made to attain reasonable speeds by adopting very large wheel diameters and best material. But practice has evinced that buckets milled into the rim of the wheel would not stand the eroding action of the steam jet, and that the steam consumption increased very materially owing to the destruction of the sharp admission edges of the buckets.

The three above-mentioned fundamental principles of steam-turbine construction have developed each for itself in Germany. The representative firms of each are: First, of the Parsons system: Brown, Boveri & Co., Baden, Switzerland, and Mannheim, Germany, licensee of C. A. Parsons; second, of the many-stage action system: the Zoelly syndicate, embracing six of the leading firms in Germany; third, of the Curtis system: Allgemeine Elektrizitäts Gesellschaft, (A. E. G.), Berlin. Gradually there has been a fusion of the different systems. For example, the A. E. G. has abandoned the Curtis construction of the low-pressure part of its 1500 revolutions per minute type and has adopted the principle of many-stage action as developed by Rateau and Zoelly. Up to this year the pure Curtis principle was followed in its 3000 revolutions per minute type, showing two pressure stages with two velocity stages each. The best known type of this class is the turbine of 1000-kilowatt capacity. But for outputs beyond 1000 kilowatts the A. E. G. now builds the low-pressure part after the Rateau-Zoelly principle. Evidently the cost of construction of the said type of 3000 revolutions per minute, which contains only two rotating wheels with two rows of buckets each and only one guide-wheel disk, is lower than the cost of a turbine of equal output which contains, besides a rotating wheel with two rows of buckets, from five to seven wheels with one row of buckets and the same number of guide-wheel disks. But the severe demand for a low steam-consumption guarantee, which would enable the firm successfully to compete with other German makes, forced the designer to adopt a somewhat more expensive construction. The Bergmann Electric Works build their turbine similarly, using a combination of the Curtis and Rateau principles.

Even the foremost of the German firms representing the Parsons interests, Brown Boveri & Co., have, after long hesitation, resolved to follow the example of other builders of reaction turbines. They have replaced the high-pressure stages of the Parsons turbine by a Curtis wheel, reducing the length and weight of the unit enormously. Only for large units and special-service conditions is the pure Parsons principle maintained. Last, but not least, the Curtis principle begins to intrude into the Zoelly syndicate, which, so far, has resisted with tenacity all attempts to change its original mode of construction. The Augsburg Nürnberg Works, members of the syndicate, have just begun to build all of their 3000 revolutions per minute types on a combination principle, using the Curtis and Zoelly features. The Görlitz Machine Works, licensees of Zoelly, have also

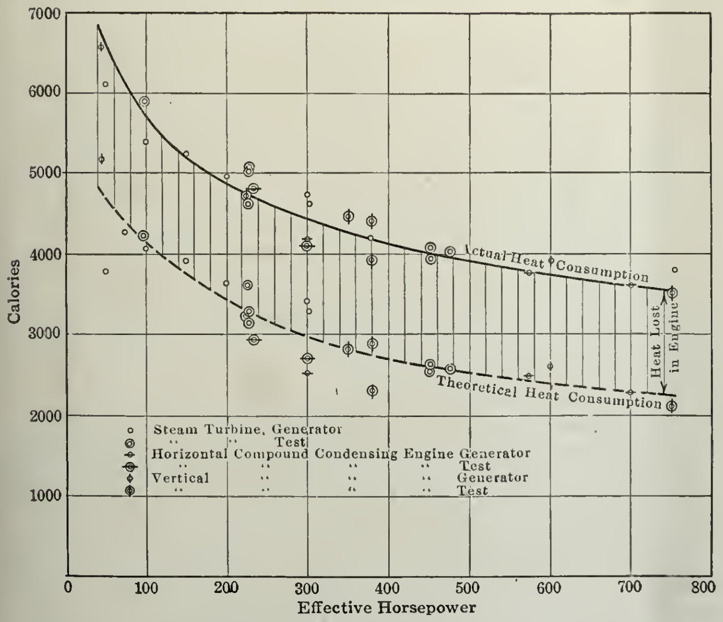


FIG. 5. INFLUENCE OF SIZE OF ENGINE ON HEAT CONSUMPTION

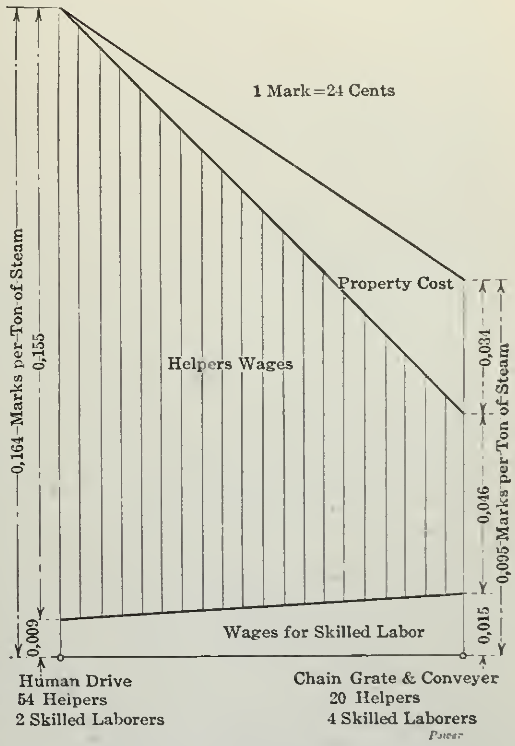


FIG. 6. REDUCTION OF STEAM COST IN BOILER HOUSE BY ELIMINATION OF HUMAN LABOR

built normal types of turbines in accordance with the above principle, but only for sizes below 700 kilowatts. The other members and licensees of the Zoelly syndicate have not adopted the new practice.

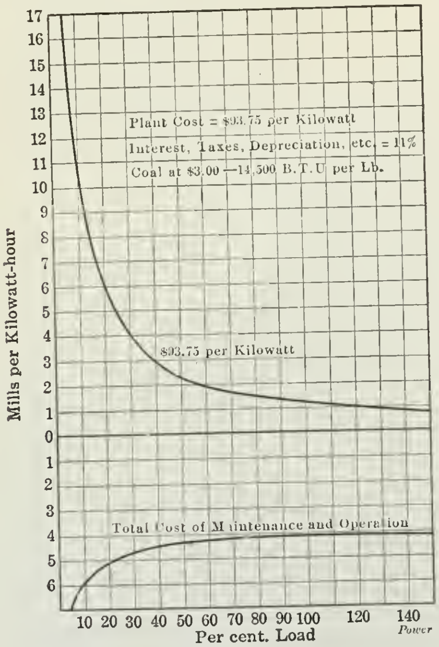
Summing up, it may be said that turbine construction tends toward the creation of a standard system of unity, which will include all of the best features of the different types. In the present stage of development the following variations exist: First, the combined Curtis and Rateau system; second, the combined Curtis and Parsons system; third, the pure Rateau-Zoelly system; fourth, the pure Parsons system; fifth, the pure Curtis system. The above order of importance has been arranged according to the influence which, in our notion, the various systems exercise upon the art of turbine building for stationary purposes in Germany today. Which of the systems will finally come to the front the competition of the future will evince.

Carrying Peak Loads Economically*

By H. G. Stott

> *The solution of this problem is to reduce the investment per kilowatt to the minimum. This may best be effected at present by the use of steam turbines, large grate area and forced draft.*

In any electric railway the first requisite of the system is that the power supply shall be absolutely reliable, and the second is that it shall be produced economically. The first principle is so thoroughly understood that nothing further need be said in regard to it. The second, however, involving *all* the cost of power, is not so clearly understood as might be wished, especially as regards peak loads.



Operating Costs and Fixed Charges for Various Load Factors

The curves shown in the accompanying diagram show the total maintenance and operating costs per kilowatt-hour for various percentages of load factor *below* the horizontal axis, while the curve *above* this axis shows the fixed charges per kilowatt-hour for various load factors. The sum of the ordinates above and below the axis give the total cost of power under the conditions which have been assumed.

From a consideration of these curves it will be seen that for peak loads having an annual load factor of from 5 to 15 per cent., the operating and maintenance charges are relatively unimportant and the fixed or capital charges are of the greatest importance.

The various sources of power to be considered for peak-load purposes are as follows:

1—Storage batteries.
2—Purchased power.
3—Hydroelectric power.
4—Gas engines.
5—Steam turbines.

1. Storage batteries would seem at first sight to be an ideal means of carrying peak loads, as obviously it would not call for any additional plant capacity, but if we apply our double curves to it, we will find that the fixed charges, on a battery capable of discharging at its maximum rate for two hours at a time, are so high as to leave it out of the question.

2. Purchased power almost invariably carries a heavy fixed charge per kilowatt of maximum demand, varying from $15 to $25 per kilowatt per annum, with a further charge per kilowatt-hour actually used. The first charge should be treated as a fixed charge and so plotted above the axis. This will in all probability show an extremely high cost per kilowatt-hour of peak load.

3. Hydroelectric power, if transmitted more than a few miles, will inevitably show very high fixed charges, owing to the large investment in the power plant, hydraulic development, transmission lines, etc.

4. Gas engines, while attractive from the point of view of small cost for fuel during the period the plant is idle, are inadmissible from the point of view of high fixed charges due to the large investment per kilowatt.

5. Steam turbines are at present the most satisfactory type of prime mover in the market for the peak-load problem, owing to their low first cost. If a reciprocating-engine plant in good condition is available, then the addition of low-pressure turbines will result in the development of a kilowatt at an investment of about $25, provided there is enough room available in the engine room and if the boilers can be forced sufficiently to carry the extra load.

In the boiler room the investment can be kept down by adding grate surface instead of more boilers, and by the use of forced draft the old rating of 10 square feet of heating surface per boiler horsepower can safely be reduced to 4 or 5 without materially adding to the cost of boiler or furnace maintenance.

While the overall boiler efficiency will begin to fall off gradually beyond 175 per cent. or 200 per cent. rating, the small loss thus entailed is insignificant compared to the saving in fixed charges as shown in the diagram.

The solution of the problem of carrying peak loads economically is therefore to be found in reducing the investment per kilowatt to a minimum and this can be best accomplished at present by the use of steam turbines and by the use of large grate area, such as a ratio of 30 or 40 square feet of heating surface to each foot of grate area, instead of the present ratio of 55 or 60 to 1. Forced draft may be employed with advantage with any grate area and will in almost any case result in increased economy as well as capacity.

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*Portion of the report of the Committee on Power Generation of the American Street and Interurban Railway Engineering Association, Atlantic City, October 10-14.

A travelin' feller kum in t' Bill Grimes' ingin room tother day an' told him th' old yarn 'bout th' feller thet sed he'd diskivered perpetshul motion an' thet it wuz his mother-in-law's tongue. Bill thot it wuz sich a good joke that he sprung it on his mother-in-law.

Th' docter thinks they kin patch Bill up so he kin get out uv th' hosspittle in 'bout three weeks.

New York City is still the leader by a long margin in the number of licensed engineers, there being nearly 13,000 It is doubtful if there are as many stationary engineers in any other State, excepting Massachusetts, Ohio and Pennsylvania.

Boiler Tube Soot Cleaners

By Warren O. Rogers

> *For years tube scrapers have been used to remove soot from boiler tubes. Many kinds, good, bad and indifferent, have been designed, and of these twenty-seven types are briefly described in this article.*

The formation of clinkers on a grate not only makes the fireman's work harder, but prevents proper combustion from taking place, as the amount of air passing up through the grate is limited. Imperfect combustion causes smoke, which deposits soot on the boiler tubes, and this reduces the efficiency of the heating surface of the boiler.

A fireman can immediately notice the effect of removing the clinkers from the grates, because the fuel burns more freely, the steam pressure is more easily maintained and less attention is required in handling the fires.

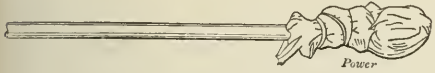


FIG. 1

An accumulation of soot on the heating surface of a boiler should be avoided just as much as clinkers on the grates. The fireman may not think much about its influence in preventing heat transmission through the tubes to the water, but if the sooty tubes of a boiler are cleaned during running hours the result will be at once manifest.

Just to what extent soot prevents heat transmission through the metal is not definitely known. This is a matter that seems to have been neglected, but even so it is no reason why the individual fireman should not keep his boiler or boilers as free from soot as the running conditions will permit.

The result of boiler tests recently conducted showed that with soot on the tubes of a boiler the evaporation from and at 212 degrees per pound of coal was 6.2 pounds of water, 13.4 pounds of coal being burned per square foot of grate surface per hour. The temperature of the gases escaping up the stack was 627 degrees Fahrenheit.

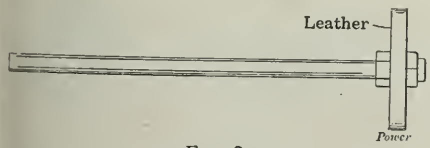


FIG. 2

With the tubes cleaned, an evaporation from and at 212 degrees per pound of coal of 7.04 pounds of water was obtained, which is a gain of 13.5 per cent. over the soot-coated tubes. The coal consumed per square foot of grate surface per hour was reduced from 13.4 to 9.09 pounds and the chimney gases reduced to a temperature of 546 degrees, or lowered 81 degrees.

Boiler tubes should be cleaned at least once in every twelve hours' run, if soft coal is used. Various methods are used for cleaning tubes. The most simple devices are the tube brush and tube scraper. Steam soot-blowing and sucking devices are also extensively used and lately several types of permanently fixed soot blowers have been devised. Many of these types of soot-handling devices will be described and illustrated in this and following articles.

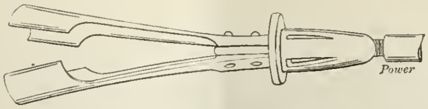


FIG. 3

Probably the first boiler attendant who recognized that the transmission of heat through boiler tubes was retarded on account of soot, and decided to use a tube cleaner, tied a bunch of rags to a length of piping or a stick, something as shown in Fig. 1, and, by pushing the bunch of rags through the tubes, forced some of the soot out into the combustion chamber.

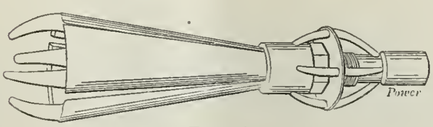


FIG. 4

Better results from cleaner tubes led to a slight improvement in design, such as Fig. 2, which might have consisted of a leather washer secured on a rod by means of two nuts. The leather washer, being stiff, would give better results than the rag arrangement. But as soot readily adheres to the inner surface of the tubes, these crude devices proved inadequate, and, as inventors realized the necessity of producing an efficient tube scraper, many designs have been brought out.

Fire-tube-boiler soot cleaners may be grouped into two general classes, soot tube scrapers and soot steam blowers and suckers.

An early type of scraper is shown in Fig. 3. It is self-explanatory. The scraper knives cut the soot, and the disk at the inner end of the scraper blades pushes the loosened soot out of the rear end of the tube. As the blades required springing together by the hand of the operator before entering a tube, such a scraper could not be expected to receive a welcome in the fire room.

The scraper, shown in Fig. 4, is made by the Gem Manufacturing Company. It consists of four cutting blades, each of which is fitted with a curved finger so that the cutting knives will readily enter the tube. The blades are made of tempered steel. When the scraper is inserted in a tube the blades get under the soot scale and remove it from the tube surface. The disk at the base serves two purposes, that of drawing out the loose soot on the return passage of the scraper, and of forming a support for the curved ribs which keep the scraper head in line when being pushed through a tube.

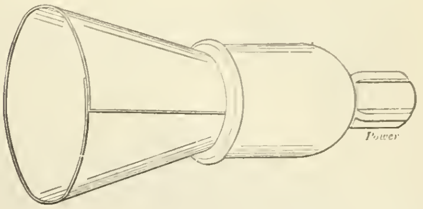


FIG. 5

Fig. 5 illustrates a scraper made by the Gilmore Tube Scraper Works. The cutting edge consists of a steel cone. A sleeve moving on a threaded shank contracts or expands the cutting edge of the cone, and a socket to carry the handle acts as a lock nut to hold the sleeve in place. The elasticity of the steel cone permits of a close fit to the inner tube surface, and when the cleaner is pushed through the tube the thin cutting edge of the cone passes between the soot and the tube, cutting the soot free and carrying it forward where it is deposited in the rear connection of the boiler.

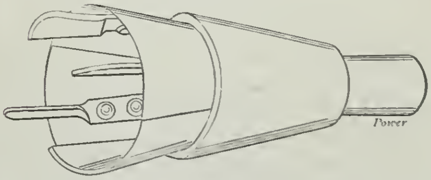


FIG. 6

What is known as the Detroit tube cleaner is shown in Fig. 6. It cuts off the soot and carries all loosened refuse back into the combustion chamber. The size of the head is adjusted by turning the handle to the right or left. The device can be connected to a steam supply and used as a blower if desired. The elasticity of the spring blades permit a close fit of the blades to the internal surface of the tube, thus preventing the steam blowing out at the front end of the tube. The cleaner consists of a brass cone, three steel blades and springs and a malleable-iron cone. The projecting fingers permit of easy entrance to the tube. It is made by J. T. Wing & Co.

A different design of scraper is shown in Fig. 7. It consists of four scraping blades hinged at the inner end. Adjusting screws are provided so that a proper fit in the tubes may be obtained. The disk at the base assists in cleaning the tubes of loose soot. It is made by Zilliox, Greene & Co.

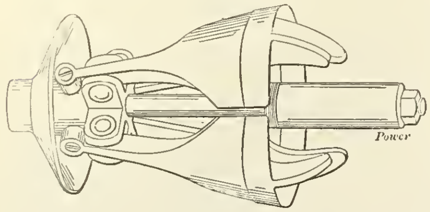


Fig. 7

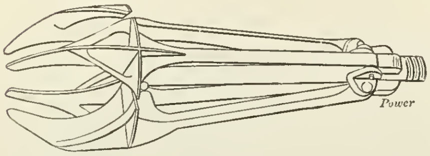


Fig. 8

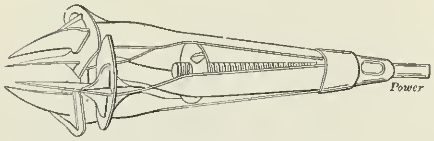


Fig. 9

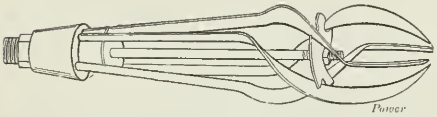


Fig. 10

The scraper shown in Fig. 8 is made up of four scraping blades, secured to arms that are hinged at the inner end, as shown. Suitable springs keep the scraper arms in an out position, but allow the malleable-iron head to easily enter the tube. The scraper blades are made with a shearing plough which cut the accumulation of soot and other deposits from the tube. It was made by the American Steam Packing Company twenty years ago.

The steel tube cleaner, made by the H. W. Johns-Manville Company, is shown in Fig. 9. It is automatic in action, simple and light. The scraper blades are so constructed that they always have a bearing on the inner surface of the tube. The scraping head is made in sections, so that the cutting blades, while free to give for any irregularity in the tube, will fit the surface throughout. The spring arms keep the scrapers expanded, and the tapered outside ends allow the scraper to easily enter the tube. An adjusting nut is shown on the center rod.

A tube scraper, used for years, is shown in Fig. 10. It consists of a cylindrical head, containing four spring scraper blades, together with a central rod on which an outer end-supporting pin is secured. The ends of the scraper blade are so formed that entrance to the tubes is easily accomplished. The scraping edges are bent at about 45 degrees with the spring portion of the blades, and so curved that one overlapped the other, so that the entire surface of the tube is scraped at one operation. This scraper is made by the Jackson Flue Scraper Company.

In Fig. 11 is shown a tube cleaner that was designed some sixteen years ago. The springs were made of rolled tempered steel and were fitted with cast-steel heads that could be reground until worn out. The knives had a shear cut, which enabled them to pass over obstructions where a square-cut knife might stick. The small springs forced the front ends of the flat spring together to enable the scraper to enter the tube. After the rear end of the cleaner had entered the tube, both ends of the scraper operated to clean the tube, the leather center pushing or drawing out the soot.

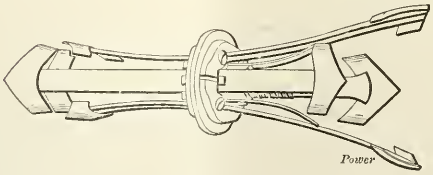


Fig. 11

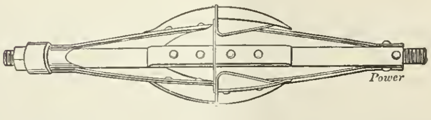


Fig. 12

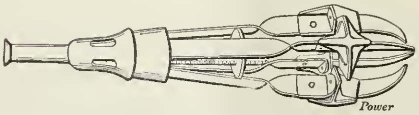


Fig. 13

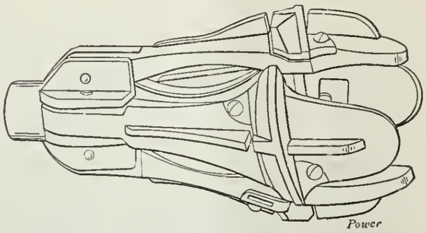


Fig. 14

With the design of tube scraper, shown in Fig. 12, the scraping knives are placed in the center, as shown. Each is riveted to a spring that extends from one end of the scraper to the other. On each side of the scraping blades is a curved metal ridge which assists the head to enter the tube. This scraper is made by the Jarecki Manufacturing Company.

Fig. 13 shows an early design of cleaner. The scraping blades are riveted to springs on one side and to a solid piece on the head end, the four pieces forming a cone-shaped head. This permits of the cleaner easily entering the tubes.

Still another design is shown in Fig. 14. It consists of four scraping knives which are hinged at the base. Springs keep the scrapers out against the tube surface.

The cleaner illustrated in Fig. 15 consists of four steel scraper blades, each being secured to a spring which is secured at both ends of the head. Owing to the projecting lug on each scraper knife they are prevented from expanding beyond the tube diameter. When the head is pushed into a tube the outer scraping blades are forced in, and the lugs, acting on the opposite end of the scrapers, permit an easy entrance of the head to the tube. The scraping blades are so arranged that the entire surface of the tube is acted upon each time the cleaner is pushed through.

Fig. 16 shows the tube scraper made by A. A. Hull. It has a spindle on which are fitted the scraper arms and cone-shaped disks. The scraping plates consist of four half disks hung loosely on a square central body of malleable iron. Forged-steel springs are fitted under the scraper arms, as shown. The scraping disks are case-hardened to prevent rapid wear.

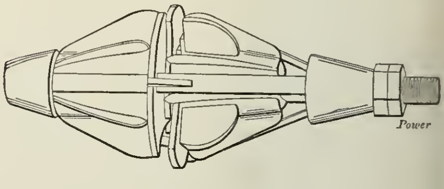


Fig. 15

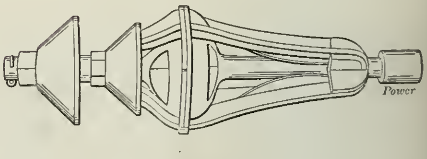


Fig. 16

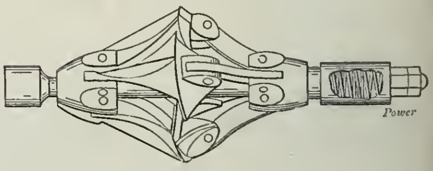


Fig. 17

One type of scraper, made by James L. Robertson & Sons, and several other manufacturers, is illustrated in Fig. 17. It has four scraping blades arranged as shown. The blades are forced out against the surface of the tubes by the spring shown in the end casing. Owing to the gradual curvature of the end links, the scraper readily enters a tube end.

The design of cleaner shown in Fig. 18 consists of four cutting knives which adapt themselves to varying thicknesses of a boiler tube. When the scrapers are forced past an obstruction the spring at the end is compressed by the collar and connections forcing it into the protecting sleeve. It is made by the J. W. Paxson Company and others.

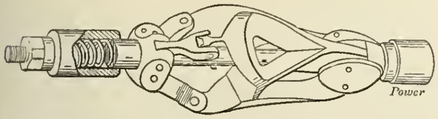


FIG. 18

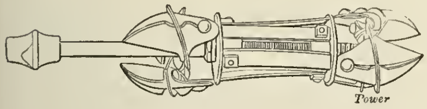


FIG. 19

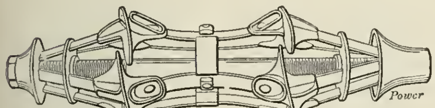


FIG. 20

An adjustable cleaner, made by the Michigan Lubricator Company, is shown in Fig. 19. It has a double scraping head and hardened-steel scraping knives. The adjustment is accomplished by screwing in the center piece which, owing to the collar, will force the adjusting plates toward the center, thus expanding the scraping blades.

What is known as a duplex cleaner is illustrated in Fig. 20. The scraper is fitted with two sets of scraping blades, as shown, so that it may do double duty. The adjustment is made on the center rod which is threaded at both ends. The scraping blades are placed at the center of the springs and are not easily broken off. It is made by the Sherwood Manufacturing Company.

Another design of tube cleaner is shown in Fig. 21. It can be adjusted while in the tube to make a close fit, or made smaller if necessary to pass over uneven surfaces. The adjustment is accomplished by turning the handle rod to which the cleaner is attached. The double heads contain the scraping edges, the latter being riveted to steel springs. A center rod is provided with a right- and left-hand thread on which the two expanded plates are screwed. As both plates move toward or away from the center of the scraper at the same time when turning the center rod, the adjustment of the scraper blades is very sensitive. The center of each spring is bolted to the center piece. Turning the expanded piece toward the center of the cleaner forces the springs out, also the steel scraper tips, thus increasing the diameter of the cleaner. A reverse motion reduces the diameter of the cleaner. It is made by A. W. Chesterton & Co. and others.

The Pratt elastic tube scraper is shown in Fig. 22. It is made by the Millers Falls Company. It has two end caps in which are set the scraping knives. These knives are flexible, but are forced out by means of springs which bear on the underside and at the center, as shown. Owing to the shape of the blades the head has two scraping ends.

An early type of cleaner is shown in Fig. 23. It is self-adjusting and made without springs or nuts. The scraper blades are adjusted by the movement of the rod while the head is in the tube. A central rod has a collar permanently fastened at each end. A loose slotted collar is placed on the rod between the bell collars to which the arms, with scrapers attached, are fastened. When the scraper is pushed into a tube the collar next to the handle becomes engaged with the convergent ends at the adjacent scraping arms, and the opposite ends of the arms are thrown out against the side of the tube. When the scraper is withdrawn from the tube the action of the arms is reversed by the rod sliding through the opposite bell collar.

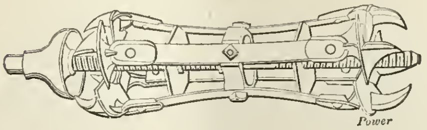


FIG. 21

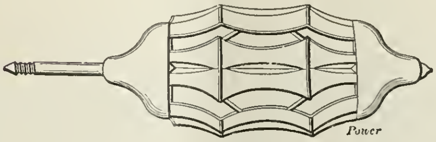


FIG. 22

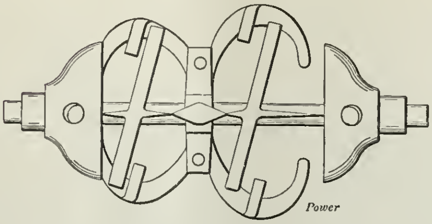


FIG. 23

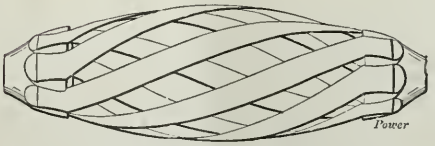


FIG. 24

A modern cleaner of similar design is shown in Fig. 24. The knives are dovetailed on the two heads, as shown, and are so constructed that they are always in contact with the metal surface of the tubes. The knives are self-sharpening. No screws or rivets are used in its construction. It is made by the Scully Steel and Iron Company and others.

An adjustable hand flue cleaner, made by Joseph T. Ryerson & Son, is shown in Fig. 25. Two nuts on a single bolt are all that are required to hold the parts together. The eight scraping blades are made fast at one end of the cleaner, the other ends being free to move radially inward. This gives the cleaner a degree of adjustability enabling it to meet any ordinary variations in the tube diameter and at the same time thoroughly clean the inner surfaces. The blades are made of spring steel and are so placed as to be easily removable in case it is necessary to make repairs. A similar cleaner is made by the Quincy Flue Cleaner Company.

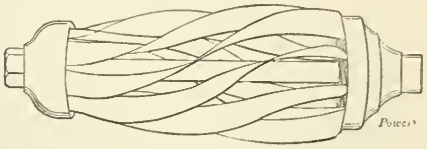


FIG. 25

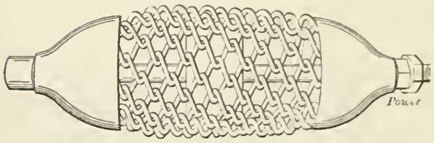


FIG. 26

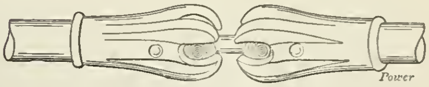


FIG. 27

The "Criss-Cross" cleaner, made by the Reversible Tube Cleaner Company, is illustrated in Fig. 26. It is made of square steel wire which is so woven that it presents to the inner surface of the tube a large number of independent, oblique, serrated cleaning edges, the resilience of which keeps them expanded with considerable force, and yet permits compression in case of blister or other permanent obstruction in the pipe. The cleaner may be adjusted for various sizes of tubes by bringing together or by drawing apart the two ends between which the woven wire is held. The cleaner blades cut when moving in either direction.

A handy ball-joint tube-scraper rod for use in restricted places is shown in Fig. 27. It is made of white ash in sections of various lengths. The sockets and joints are made of malleable iron, the connecting balls of steel. It will fold in any direction and does not kink. It is made by James McCrea & Co.

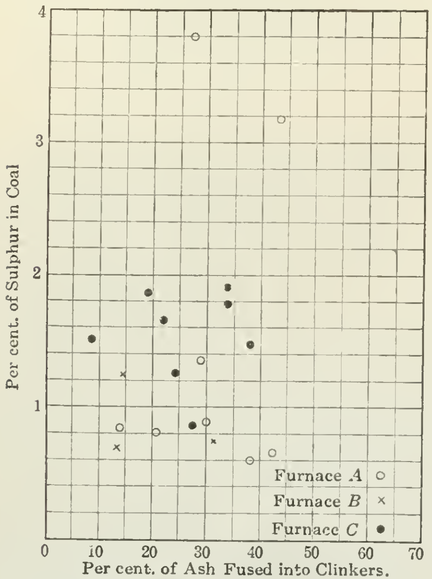
Fusing Temperatures of Ash*

By E. G. Bailey and
and W. B. Calkins

> *The quantity of clinker formed by a given coal depends upon the composition and the fusing temperature. The effect of the first factor is so little understood that dependence must be placed upon the latter and conditions regulated to suit.*

Everyone who is at all acquainted with the burning of coal or coke realizes that the clinker problem is one of very great importance. Probably more shutdowns in boiler plants and steam failures on locomotives are due to the formation of clinkers than to any other one cause. The purchase of coal on specification is often unjust, as no proper method of determining its clinkering properties has been used, and excessive penalties are often exacted for sulphur, due to the erroneous idea that it causes clinker, while other coal commands a premium for high B.t.u. and low ash, but clinkers badly.

The accumulation of clinkers on the grate of any furnace has several bad effects. The most serious one is the reduction of the air supply, which in turn decreases the rate of combustion and the heat-generating capacity of the furnace. In order to obtain maximum efficiency in the burning of coal it is necessary to have the required supply of air distributed uniformly throughout the fuel bed, but the presence of clinkers greatly interferes with this proper distribution of air and causes a reduction in the economy, as well as the capacity, of the plant. Clinkers also add to the labor of firing and greatly increase the expense for repairs to brickwork, grate bars, etc.



Note:- All Furnaces Hand Fired. *Power*
Rate of Combustion,
A. 23 Lbs. per Sq. Ft. of Grate per Hour.
B. 11 Lbs. per Sq. Ft. of Grate per Hour.
C. 24 Lbs. per Sq. Ft. of Grate per Hour.

FIG. 1. RELATION BETWEEN SULPHUR CONTENT AND PERCENTAGE OF CLINKER IN ASH

A clinker is a fused mass of ash. The formation of clinker depends upon two factors, the temperature at which the ash will melt and the temperature to which the ash is subjected. It is evident that the temperature to which the ash is subjected is largely dependent upon the method of firing and the rate of combustion and it is often possible to lessen the trouble due to clinkers by changing the method of firing. Even where the furnace conditions are similar, there is a great difference in the quantity and nature of clinker formed from different coals. This difference is directly dependent upon the temperature at which the ash will fuse or melt.

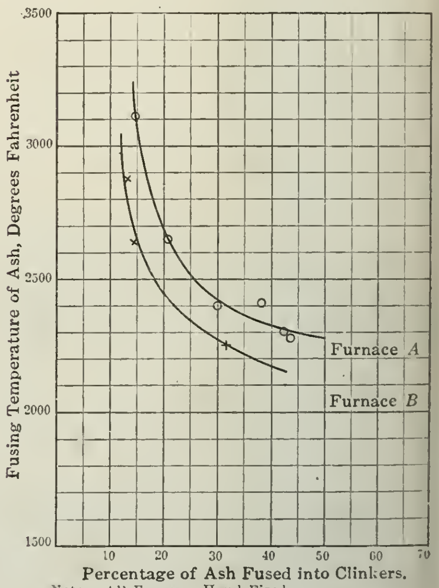
Heretofore, no realible method has been known whereby the clinker-forming properties of a coal could be predetermined, nor has it been possible to make comparisons of this characteristic in different coals. There is a prevalent opinion that the percentage of sulphur is an indication of the trouble which may be expected from clinkers, but this has been proved to be an extremely unreliable and unjust method of comparison. Fig. 1 illustrates very clearly that there is no relation between the percentage of sulphur in a coal and the percentage of its ash which is fused together as clinker. Each of the 19 tests shown was of 24 hours' or more duration and was made on hand-fired boiler furnaces with similar conditions, except the rate of combustion as indicated. All clinkers one inch in size or larger were picked out from the ashes and refuse resulting from the burning of different kinds of coal and the percentage based upon the total ash in the coal burned.

It was impossible to determine the fusing temperature of the ash from all of the above tests, as most of the samples of coal had been destroyed before we were prepared to determine this temperature, but all such data available at the present time are plotted in Fig. 2. The relation here seen to exist between the percentage of the ash that fused into clinkers and the fusing temperature of the ash assures the value and practicability of this determination.

It will be seen that a difference of a few hundred degrees at the higher temperatures has little effect upon the amount of clinker formed, but the percentage of clinker increases very rapidly with a lower fusing temperature of the ash.

An important factor, not brought out by these curves, is that the clinker made from an ash of low fusing temperature will obstruct more grate area than will the same weight of clinker formed from an ash of higher fusing temperature. If this objectionable characteristic could be plotted on the curves of Fig. 2, in addition to the percentage of clinker, the curves would bend toward the horizontal much more abruptly, indicating a still greater increase in the difficulties connected with the burning of a coal with an ash of low fusing temperature.

The percentage of ash must also be taken into consideration, as coal containing 9 per cent. of ash, 20 per cent. of which formed clinker, might be just as objectionable as one containing only 6 per cent. of ash, 30 per cent. of which formed clinker.



Note:- All Furnaces Hand Fired. *Power*
Rate of Combustion,
A. 22 Lbs. per Sq. Ft. Grate per Hour.
B. 11 Lbs. per Sq. Ft. Grate per Hour.

FIG. 2. RELATION BETWEEN TEMPERATURE AND PERCENTAGE OF CLINKER IN ASH

Another interesting point brought out by Fig. 2 is the difference between the results obtained in the furnaces *A* and *B*, which is almost entirely due to the difference in the rates of combustion. The coal burned per square foot of grate during the different tests on furnace *A* varied from 19 to 26, with an average of 22 pounds per hour; while in furnace *B* it averaged only 11 pounds per hour and varied from 10.6 to 11.8 pounds. The average temperature in furnace *A* was higher than in *B* on account of the higher rate of combustion, and there would naturally be a greater percentage

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*Abstract from *Bulletin* No. 3, issued by the Fuel Testing Company, Boston, Mass.

of clinker formed from ash of the same fusing temperature.

Many people who have attempted to ascertain the clinkering properties of different fuels have realized the uncertainty of sulphur as an indicator and, due to their inability to determine the fusing temperature of the ash, have depended on other characteristics. The color of the unfused ash and the percentage of iron in the ash have been more generally used for this purpose, but a little study of the accompanying table indicates that no definite relation exists, and that these characteristics are as misleading as is the percentage of sulphur.

Even the complete analysis of the ash, giving the content of silica, alumina and the oxides of iron, lime, magnesia, potash and soda, has often been made with a view to obtaining some knowledge of the fusing temperature. But these constituents, singly or collectively, do not hold a known relation to fusibility, and the importance of the relation of each constituent to all others makes the problem so complicated that only a vague and general idea can possibly be obtained from such an analysis. Doctor Seger, the eminent German scientist, has shown that the fusing temperature of such material may be lowered by adding some flux, such as lime, up to a certain point, while a still greater percentage of it will increase the fusing temperature. In like manner, additional silica may raise or lower the fusing temperature, depending upon its relation to the other constituents, as well as their relation to each other.

Even if one were able to estimate the clinkering property of a coal from the complete analysis of the ash, this analysis alone is much more laborious and expensive than is the direct method of determining the temperature.

TABLE OF COAL CHARACTERISTICS

No.	Character of Coal.	Fusing Temperature of Ash, Degrees Fahrenheit.	Percentage of			Color of Unfused Pulverized Ash.	
			Ash in Coal.	Sulphur in Coal.	Iron Oxide in Ash.	Relative Density of Color.*	Color Shade.
1	Anthracite	3000	16.11	0.75	1.1	2	pink
2	Semibituminous	2910	7.97	0.96	7.0	10	brown
3	Semibituminous	2880	7.33	0.70	10.0	1	gray
4	Semibituminous	2640	8.90	1.25	12.0	1	pink
5	Gas	2640	9.84	0.72	6.6	3	yellow
6	Semibituminous	2550	12.85	3.10	23.7	16	brown
7	Gas	2520	7.22	1.22	13.0	8	yellow
8	Anthracite	2520	10.08	0.61	6.1	6	yellow
9	Gas	2460	6.91	2.35	24.7	15	brown
10	Gas	2350	7.41	1.98	17.5	12	red
11	Semibituminous	2320	13.06	0.92	7.1	5	yellow
12	Semibituminous	2320	10.76	2.22	20.6	13	red
13	Gas	2300	6.78	0.98	10.6	7	yellow
14	Gas	2260	7.25	3.12	34.0	14	red
15	Semibituminous	2250	7.11	0.73	12.9	11	yellow
16	Coke	2190	10.22	1.40	13.9	9	yellow

*Graded according to density; No. 1, lightest; 16, darkest.

Exhaust Steam Regenerators

By Charles H. Smoot

> *Regenerators are divided into two general types. In one, the water passes through the incoming steam. The energy required to circulate the water in such regenerators is so great that it considerably lowers the efficiency. In the other type, the steam passes through the water; circulation is rapid, and the mixture of steam and water most thorough.*

The exhaust-steam regenerator is a heat accumulator whose function is to equalize variable fluxes of steam, receiving a variable flux and delivering a constant one. When the flux of steam received by the regenerator exceeds that delivered, the function of the regenerator is to absorb the excess, holding it in reserve until a future period occurs when the steam received is less than that required, the deficiency being made up by withdrawing from the regenerator steam which has previously been stored.

When the flux of steam received from the engines by the regenerator is just equal to that taken by the turbines, the pressure within the regenerator is maintained constant, no steam is condensed and the regenerator floats idle. As soon, however, as steam comes to the regenerator in greater quantities than required by the turbine, the pressure in the regenerator is slightly raised, with an increased temperature of steam, which, becoming hotter than the water, allows the water to absorb the excess.

Should the quantity of the incoming steam be less than that required by the turbine, the pressure in the regenerator falls, lowering the temperature of the steam, and allowing the water to give off steam, removing the heat which was previously absorbed during a time of excess.

The regenerator is a storage reservoir, which is floating continually on the system, and it neither absorbs nor gives off steam when the engine exhaust equals the turbine requirements. But as soon as the engine exhaust varies from the turbine requirements, the regenerator comes into play and it absorbs the excess steam or delivers the deficient steam.

That the regenerator may have its entire heat-storage capacity available to absorb a sudden large excess of steam, such as occurs in steel-mill practice, the fundamental necessity is that a sufficient amount of water be always present in the zone of condensation. If the regenerator is so arranged that the entire mass of water is not always present in the zone of condensation, a sudden excess does not come in contact with the entire mass of water and consequently the full absorption capacity of the regenerator is not available to absorb excess steam.

In practice, the conditions of absorption are far more severe than those of delivery, for when the regenerator is called upon to absorb steam it generally happens that the steam to be absorbed is entering at a rate far in excess of the average value of flow to the turbines, so that the maximum flux of steam which the regenerator must absorb is considerably in excess of the maximum rate of flow which the regenerator is obliged to give off.

The steam delivery from the regenerator seldom exceeds the average rate of engine exhaust, which is equal to the maximum steam consumption of the turbine. This marked difference between the capacity for absorption and that for delivery is well illustrated by the accompanying curves, based on actual tests of a large rail mill. The tests show that the length of time during which a regenerator is absorbing steam is only a small portion of the total time of operation, and that when the regenerator is absorbing steam it must absorb fluxes whose rate of flow is far in excess of the rate of flow for steam delivery. This point cannot be too strongly emphasized, for it is here that the steam regenerator as a steam economizer becomes either a failure or a success.

The practical type of regenerator as at present in use employs water as a heat-storage medium, the steam being stored in the water as heat. When the steam flow is in excess, the pressure within the regenerator increases and therefore the steam temperature also increases. If the regenerator is well designed, the entire mass of water being in intimate mixture with the steam, the temperature of the water is obliged to increase with that of the steam; consequently the water absorbs heat from the steam, condensing portions of it and reducing the pressure.

The closeness with which the temperature of the water follows variations in the temperature of the steam depends upon how thoroughly mixed the two elements are in the zone of condensation, for if insufficient water is present to condense the surplus steam flowing at any instant, the variations in pressure are large and the temperature difference between steam and water is excessive. Since a temperature difference of 2.8 degrees Fahrenheit at atmospheric pressure corresponds to one pound per square inch pressure difference, the importance of small temperature variations is apparent.

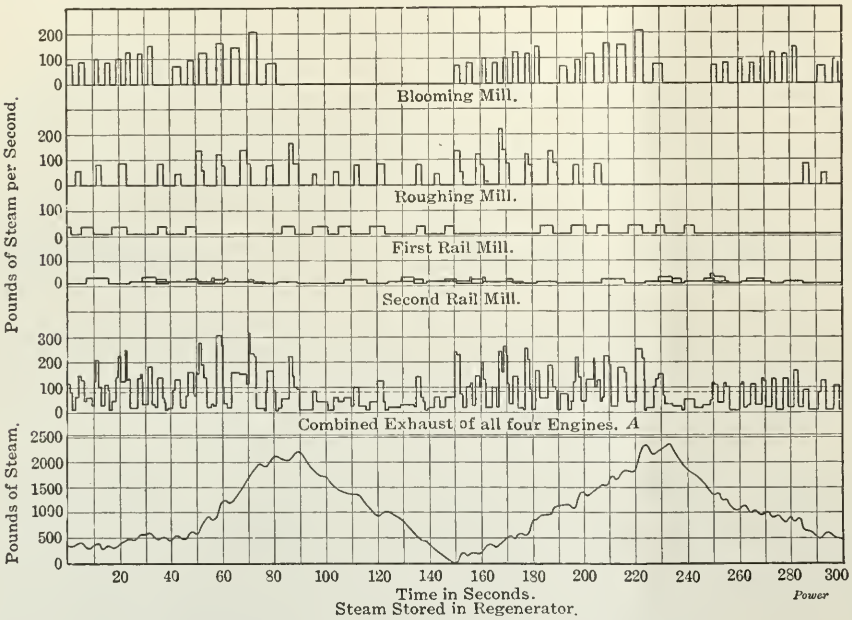

Fig. 1. Diagrams Showing Exhaust of Rail-mill Engines

It is possible, theoretically, to store heat in metal or any other substance which is capable of absorbing heat when its temperature rises. Water, however, is the most effective of all substances for absorbing heat, since it has the highest heat capacity per unit of weight.

Steam regenerators are usually installed in connection with low-pressure steam turbines, although their use is not limited to turbines alone and they may be applied wherever it is desired to change from a variable flux of steam to a continuous flux, or *vice versa*. Regenerators equalize the flow of steam in the same manner as a water reservoir equalizes a flow of water. When used in conjunction with engines whose exhausts vary from time to time, the regenerator, by absorbing the excess steam for future use, saves the future use of an equal amount of high-pressure steam, which, if a regenerator were not present, would be required to make up the future deficiency in the exhaust-steam supply.

Application of Regenerators in Rolling-mill Practice

Fig. 1 is an illustration of a typical steam flux which regenerators are called upon to equalize in a rail mill. The engines whose exhausts comprise the steam received by the regenerators consist of one blooming-mill, one roughing-mill and two finishing-mills, all of which engines work more or less in conjunction, since they act upon the same steel in its passage through the mill, beginning as a bloom and ending a finished rail.

These curves were obtained for the purpose of investigating the advisability of a regenerator plant. The abscissas or horizontal distances represent time in seconds. The ordinates or vertical distances represent quantities of steam.

Curve *A* represents the flux of steam in pounds per second delivered by all four engines working under normal conditions. The steam flow is extremely variable, having small rises and falls occurring within four- or five-second intervals, as well as much larger periods of excess and deficiency, whose intervals are of about one minute duration.

The dash line passing through this curve at the hight for 80 pounds per second shows the average steam flow from the four engines. Wherever the line of total steam flux passes above the dash line, the flux is in excess of the turbine requirements and some of it must, therefore, be absorbed by the regenerator. The feature which is most strikingly apparent is that the rate of steam flow in excess of the average value reaches maximums which are far in excess of the rates of flow needed to make up deficiencies, thus illustrating the fact that the regenerator is more severely taxed during the periods of steam absorption than during the periods of steam delivery.

The lowest curve, more or less sinusoidal in shape, shows the total weight of steam in pounds which must be stored in the regenerator at the corresponding instant. Where this curve is rising, the steam from the engine exceeds the flow to the turbine, and, conversely, where it is drooping, the regenerator, by giving off steam, is making up the deficiency of steam supplied by the engines.

The average time between crests and troughs is something like one minute; consequently, the approximate period during which the regenerator is storing an excess or supplying a deficiency is of one minute duration.

The greatest total amount of steam stored in the regenerator occurs at 233 seconds from the start, at which time the regenerator contains 2350 pounds of steam more than at 150 seconds; at the latter point the reading is zero.

A well designed low-pressure turbine installation employing this amount of steam, viz., 80 pounds per second, or 290,000 pounds per hour at 28 pounds per kilowatt, is capable of generating 10,000 kilowatts.

To apply a regenerator to this typical rail mill requires that the regenerator be capable of passing to the turbines an average steam flow of 80 pounds per second, and that it be capable of absorbing between the greatest and lowest pressures permissible in the regenerator (usually atmosphere and four pounds above) 2350 pounds of steam.

The amount of water theoretically necessary to absorb this quantity of steam equals:

$$\frac{2350 \times \textit{average latent heat}}{\textit{Temp. at 4 lb. gage} - \textit{temp. at 0 lb. gage}} =$$

$$\frac{2350 \times 962}{12.4} = 182{,}000 \textit{ pounds}$$

The economy effected by the regenerator is easily estimated. The omission of the regenerator would oblige all surplus steam over the 80-pound rate to discharge to the atmosphere. The total weight of steam which would be so discharged during the time interval covered by the curve *A* in Fig. 1 amounts to 8800 pounds in the 300-second period. This is equivalent to a continuous loss of 106,000 pounds per hour.

In addition to this waste of exhaust steam, in the periods of deficiency the required amount would have to be supplied from the high-pressure mains. The total amount supplied during the time covered by the curve is equivalent to about 80,000 pounds an hour.

The steam taken by the turbine, amounting to 80 pounds per second, is equivalent to 290,000 pounds per hour. Consequently, the loss of steam due to the absence of a regenerator in such an installation causes a waste of

$$\frac{186{,}000}{290{,}000} = 0.64$$

or 64 per cent. of the average steam taken by the turbines.

ABSORPTION

As affecting the regenerator design, usefulness and efficiency, the consideration of prime importance is the question of steam absorption and not that of steam delivery. Any mass of water will give off steam when its pressure is reduced below the critical point, or the point at which the temperature is equal to the temperature of saturated steam at the same pressure. On the other hand, the absorption of steam is of vital importance. since all of the important energy osses in the regenerator occur during this part of its action, which losses, however, can be fully controlled by the design of the apparatus.

The efficiency of the regenerator as a piece of power apparatus is, as usual, the atio of power output to power input.

If the regenerator delivers steam at he same pressure as that at which it eceives steam, barring the very slight osses due to radiation, its efficiency is 00 per cent. On the other hand, if the egenerator always receives steam at a igher pressure than that at which it can eliver steam, its efficiency is markedly ess than 100 per cent., and the loss is epresented by the pressure drop between he absorbing condition and the condion of delivery. This depends entirely pon the regenerator design and especial upon the provision for maintaining ithin the condensing zone a large volme of water in proportion to the flux f steam.

If the flux of water is 276 pounds per ound of steam, the temperature difference between the steam which is being ondensed and the water doing the conensing will be not less than 3½ degrees ahrenheit, or 1¼ pounds per square ch pressure difference. The pressure fference demonstrates itself at the inant when the regenerator changes from n absorbing action to a delivery action, nce the pressure must fall 1¼ pounds. hese figures apply to a highly efficient generator.

A machine which provides a flux of ater through the condensing zone less an 276 pounds per pound of steam is oportionately less efficient. If the flux water is reduced to, say, 20 pounds r pound of steam, the temperature difrence between the steam and the water comes approximately 48 degrees, corsponding to a pressure drop from 19 pounds absolute to 7 pounds absolute, or a loss in pressure of 12 pounds.

In that type of regenerator in which steam and water are mixed in a separate condensing vessel located below the main regenerator reservoir, and in which water passes from the upper vessel by the action of gravity into the lower vessel, where it is sprayed in small streams into the incoming steam flux, practical considerations of construction limit the flux of water to not over 40 pounds per pound of steam; and since 2.8 degrees Fahrenheit difference in temperature corresponds to one pound per square inch difference in pressure, this type of regenerator cannot give off steam until the pressure is reduced by at least

$$\frac{962}{40} = 24$$

degrees Fahrenheit, or

$$\frac{24}{2.8} = 8.6$$

pounds per square inch in the loss of pressure between the conditions of absorption and those for delivery. Since the total pressure range available to the turbine between the admission pressure and the pressure in the condenser does not exceed some 13 pounds per square inch, the above loss in pressure due to the insufficient absorption capacity of the regenerator amounts to 57 per cent. in available pressure at the turbine.

In order that a regenerator may be capable of absorbing excess steam as it occurs in usual rolling-mill practice, it is necessary that an intimate mixture of steam and water be continually preserved and that the temperature of the water follow closely the temperature of the steam as it rises or falls with corresponding variations in pressure. This requires that a large mass of water be always in contact with the steam flux. The only appreciable losses of energy which occur in a regenerator arise from a failure in this respect.

The temperature of the steam during the process of absorption must always be, unavoidably, slightly greater than the temperature during the evolution of steam. The design of the regenerator, however, can greatly reduce the resulting loss of energy, and it is this consideration which led Professor Rateau, the originator of the regenerator principle, to abandon long ago the idea of regenerators in which water is caused to flow through steam, substituting for this arrangement that in which steam is caused to flow through water.

The first arrangement is incapable of absorbing large quantities of steam in short periods, because it is impracticable to force the circulation of the enormous volume of water required to condense the steam when the temperature difference between the steam and the water is limited to less than 5 degrees Fahrenheit, which is the limit in the design adopted for all the Rateau regenerators and which corresponds to about 1.8 pounds per square inch pressure drop between absorption and delivery of steam.

To work within a limit as close as this requires a circulation of 195 pounds of water for each pound of steam, since the latent heat of steam is 962 B.t.u. per pound in the neighborhood of atmospheric pressure, and to absorb the heat contained by one pound of steam by water whose temperature increase is limited to 5 degrees Fahrenheit requires, therefore, about 195 pounds of water.

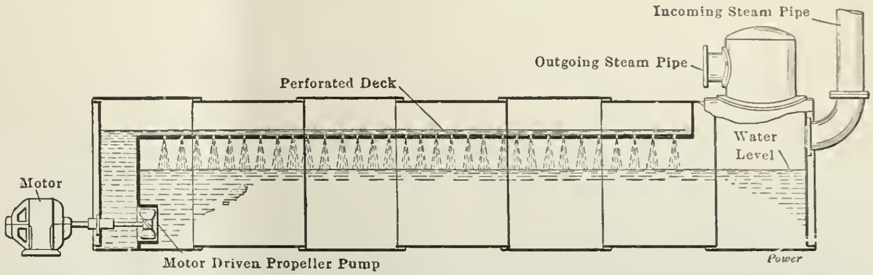

FIG. 2. REGENERATOR IN WHICH WATER PASSES THROUGH THE STEAM

The enormous flux of water required by practical regenerators is, therefore, obvious. Where steam is to be handled in quantities as great as 300,000 pounds per hour, the circulation of water through the condensing zone of the regenerator must be 195 times this, or 585,000,000 pounds per hour. To pump such an amount of water against a head of even 10 feet is an obvious impossibility, but in the Rateau regenerator fluxes of water as great as this are readily realized, since the water circulation has a very large cross-sectional area and the length of the path is reduced to only a few feet. The circulation occurs around a guide plate and is maintained by the passage of steam in very small bubbles through the upward rising current of water.

Any attempt to operate a regenerator with a forced circulation of water, in which the water is sprayed into the steam, must inevitably prove a failure in so far as economy of operation is concerned, for the reasons just outlined.

The designer of a regenerator acting on this principle is confronted with the following dilemma: In order that the regenerator may be capable of absorbing steam efficiently, i.e., with a small temperature difference between absorbing water and steam to be absorbed, the water circulation through the condensing vessel must be in the neighborhood of 195 pounds of water per pound of steam. To pump this quantity of water requires a large amount of power, which is lost, impairing the efficiency. On the other hand, if to economize the power required to maintain the water circulation, the quantity of water is reduced, the regenerator has its absorption capacity reduced and becomes incapable of absorbing the full excess of steam as it occurs, and necessitates equipping the regenerator with a relief valve on the turbine side, which will blow to atmosphere a large portion of the excess incoming steam.

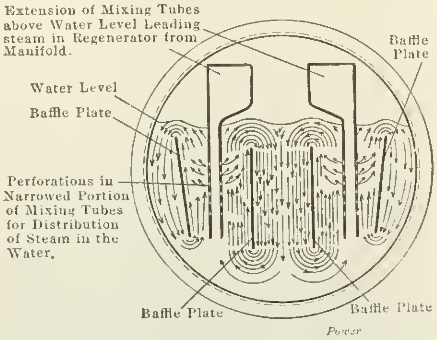
Extension of Mixing Tubes above Water Level Leading steam in Regenerator from Manifold.
Baffle Plate
Water Level
Baffle Plate
Perforations in Narrowed Portion of Mixing Tubes for Distribution of Steam in the Water.
Baffle Plate
Baffle Plate
Power

Fig. 3. Regenerator in which Steam Passes through the Water

A regenerator which is well designed will absorb steam fluxes which may be five times as great for short intervals as the flux of steam continuously taken from the regenerator by the turbines, but this can be accomplished only by a regenerator which is so designed that the entire mass of water is brought into the zone of condensation many times in each minute. A regenerator in which the water is forced to circulate through a special condensing chamber separate and distinct from the main regenerator reservoir, for the reasons outlined above, is incapable of absorbing an excess even as great as the average flux of steam required for the turbine without discharging steam to atmosphere.

The practicability of designing a regenerator along these lines, which will efficiently absorb large fluxes of steam, can be readily grasped by comparing it with a barometric condenser, whose cooling water is not more than 5 degrees Fahrenheit colder than the steam to be condensed. It takes a very effective condenser to heat the condensing water up to within five degrees of the temperature of the incoming steam; consequently, a regenerator which acts along similar lines is obliged to be more effective as a condenser than the best of barometric condensers at present in use, since a temperature difference of five degrees between water and steam will correspond to a very appreciable loss of energy in a regenerator.

In constructing regenerators of new types, the feature of prime importance is this question of steam absorption; the question of steam delivery is a matter which takes care of itself.

Relief Valves

In addition to the regenerator proper, the steam valves which are called on to pass variable fluxes of steam encountered in steel-mill work are of great importance. The usual relief valves as at present designed for service with condensing engines are wholly inadequate for this work. The passage of steam occurs with the violence of an explosion, and the valve must be capable of opening and closing its full travel many times in each minute without pounding itself to pieces.

Some of the valves required are over 36 inches in diameter, with moving parts whose weights may reach as high as 1000 pounds, and must oscillate up and down with a travel of some 6 inches or more, 25 or 30 times a minute. The mechanical difficulties of constructing such valves are apparent. The energy of the blow with which such a valve would strike its seat would be on a par with that of a steam hammer if special provision were not made to absorb the energy of the blow. This violence of action arising from the rapidly changing flux of steam given off by blooming mills and similar engines absolutely prohibits the use of any moving parts in a regenerator which are subject to this action, since such parts would very rapidly wear themselves to pieces unless the rapidity of their action were restricted; and if this is done, then the free passage of steam into the regenerator becomes restricted, and large back pressures are placed on the engines.

Types of Regenerators

To classify the different types of regenerator construction is somewhat difficult as numerous forms have been built and proposed. The principal difference, however, between the proposed methods of construction consists in either causing the exhaust steam to flow through the water contained in the regenerator shell, or causing the water which is to absorb the steam to flow in finely divided streams through the steam.

Fig. 2 illustrates diagrammatically an effective type of regenerator, in which the water passes through the steam. The motor-driven pump acts against a head of some 5 feet, lifting the water from the main reservoir to the upper chamber, from which it passes in a very large number of fine streams through the steam to be condensed. The action of this regenerator to absorb steam is the same as the usual barometric condenser, only a far larger space must be provided in which to mix the steam and water. The great difference between this regenerator and a condenser, however, lies in the fact that the temperature of the water which is to condense the steam must be nearly the same as that of the steam itself; consequently far more water is required than for a condenser.

If a very large volume of water is not kept in circulation, the absorption capacity of this type of regenerator becomes limited, and the loss of energy becomes large.

On the other hand, the type illustrated in Fig. 3, in which the steam is caused to pass through the water, allows the entire mass of water in the regenerator to be intimately and rapidly mixed with the incoming steam. The circulation of water is maintained by the passage of steam in very small bubbles upward in the rising current of water, causing a velocity of flow which brings all of the water into the zone of condensation at least once every three seconds. This feature gives to this type of regenerator a vastly greater rate of steam absorption than can be obtained in the type illustrated in Fig. 2.

With reference to the giving off of steam which has been absorbed, these two types of regenerator are very much alike. There is one exception, however, which gives considerable preference to the type shown in Fig. 3. This lies in the fact that the passage of the steam through the water maintains a circulation, as indicated by the arrows. This circulation brings to the surface all of the water at regular intervals, where its pressure is reduced to the lowest value existing within the regenerator shell. Since all of the water within the regenerator shell is kept at the same temperature, a reduction in pressure of steam above the water allows the surface water to give off sufficient steam to cool it from the maximum temperature to the new temperature determined by the new pressure. On the other hand, the water lying at the bottom of the regenerator cannot begin to give off any steam until the pressure of the steam over the water has been reduced by an amount somewhat greater than the pressure corresponding to the hight of water above that at the bottom.

In other words, if the water in a regenerator is uniformly at the temperature of 212 degrees, corresponding to atmospheric pressure, and if the depth of the water is 6 feet, then a reduction in steam pressure of 2.62 pounds per square inch will just bring the pressure of the water at the depth of 6 feet to atmospheric pressure, at which point it begins to give off steam.

The water which was at the surface of the regenerator, however, begins to give off steam immediately a reduction in steam pressure occurs. If no circulation exists, the water lying at the bottom of the shell will take very little part in the action of giving off steam, unless the reduction of pressure above the water is excessive. This difficulty can be entirely overcome by giving the water a continuous circulation, bringing to the surface at regular intervals all of the water contained within the shell, so that during the process of giving off steam all of the water within the regenerator is brought to the surface, where the static head is removed, and the water is allowed to give off its full share of steam.

The type of regenerator illustrated in Fig. 2 cannot be provided with any very practical means for keeping the large mass of water circulating in the main reservoir, and therefore the steam given off by the water in the lower portions of the vessel will be very much less than for the upper portions, thus reducing the effective capacity of the regenerator as a whole.

An interesting comparison of these two types of regenerator can be made by applying them to the steam flux illustrated for the rail mill in Fig. 1. The maximum flow of steam to the regenerators occurs at the instant 60 seconds, and amounts to 300 pounds per second. In order that no exhaust steam will be wasted, the absorption capacity of the regenerators for this installation should be equal to this maximum rate of flow. Taking the pressure variation permissible in the regenerators to be four pounds per square inch above atmosphere, the maximum and minimum temperatures of steam corresponding to these two pressures will be 224.4 degrees at four pounds above and 42 degrees at atmospheric pressure. The latent heat of steam within this pressure range is 962 B.t.u. per pound; consequently, to condense one pound of steam,

$$\frac{962}{224.4 - 212} = 77.5$$

pounds of water will be required.

The total weight of water required withn the shells, on the basis that each pound of water is capable of giving off its full share of steam is

$$77.5 \times 2350 = 182{,}000 \text{ pounds.}$$

In a regenerator of the type of contruction illustrated in Fig. 3, the full mount of water is brought where it may ct upon the incoming steam once every hree seconds, giving, therefore, a flux of water for the purpose of condensing team which amounts to 60,700 pounds er second.

The maximum flow of steam which the egenerator must absorb is

$$300 - 80 = 220$$

ounds per second, which requires

$$220 \times 77.5 = 17{,}000$$

ounds of water per second. Therefore, his type of regenerator is capable of aborbing a steam flux far in excess of hat which actually occurs. In order that he regenerator illustrated in Fig. 2 may e capable of absorbing this same steam ux, the flow of water through it must e 17,000 pounds per second, and if the effective head through which this water must circulate be taken at 10 feet, the power expended in the circulating pump will be

$$\frac{17{,}000 \times 60 \times 10}{33{,}000} = 308$$

water horsepower, which, with an overall efficiency for pump and motor of 40 per cent., will require an input of 770 horsepower.

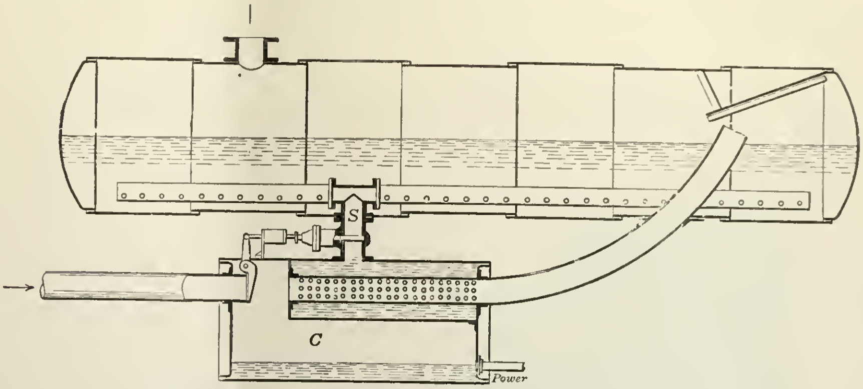

Fig. 4. Regenerator Based on the Injector Principle

Another type of regenerator, similar to that illustrated in Fig. 2, since in it a flow of water is introduced in such a way as to pass through the incoming steam, has recently been constructed in this country, and tests are soon to be made. This apparatus is illustrated diagrammatically in Fig. 4. In principle it differs from that illustrated in Fig. 2 through using the impact of incoming steam against condensing water to lift the water from the lower chamber back into the upper chamber, and relies upon the action of gravity to introduce condensing water from the upper into the lower, or condensing vessel. In this apparatus the available head of water for creating a flow from the upper to the lower chamber is approximately 10 feet. The maximum theoretical velocity which water can attain flowing under 10-foot head, with no allowances whatever for frictional resistance or obstructions, is 25.5 feet per second. Consequently, a regenerator of this type, to have a steam-absorbing capacity great enough to absorb the maximum flow of steam (220 pounds per second) given by the engines whose performance is represented on curve *A*, Fig. 1, must be 17,000 pounds per second, which, at the above given velocity, would require that the pipe *S* in Fig. 4 have a sectional area of 10 square feet or a diameter of 3 feet 7 inches.

To properly subdivide a flow of water of this extent will require that the lower chamber *C* be of very large dimensions, and it is somewhat doubtful if the incoming steam can entrain and deliver to the upper chamber so great a flow of water, whose weight in pounds per unit of time is

$$\frac{17{,}000}{300} = 56.6$$

times the weight of incoming steam for the same interval.

To use the steam flow for the purpose of lifting this water in place of the usual motor-driven pump is an advantage, since it eliminates some auxiliary apparatus, but on the other hand, the efficiency of the motor-driven pump for lifting water is very much greater than the efficiency of the incoming steam to do the same thing, since the incoming steam lifts the water in a similar manner to the use of steam in an injector for boiler feed, the efficiency of which when well designed for the purpose cannot exceed 20 per cent. The efficiency of the steam in this type of regenerator for lifting water must be far less than this, since in designing the steam and water chamber, no attempt has been made to conform to the lines of the boiler injector.

The first plant equipped with a steam regenerator was designed and put in operation by Prof. A. Rateau at the Bruay mines, France, in 1890, the source of exhaust steam being a hoisting engine.

In our country the first installation was that of the Wisconsin Steel Company, South Chicago, where the regenerator receives the exhaust steam of a reversing blooming engine and delivers steam to a low-pressure turbine of some 600 kilowatts capacity.

A second plant was installed at the Vandergrift works of the American Sheet and Tin Plate Company the following year; the regenerator handles the exhaust of a reversing mill engine and supplies steam to a Rateau-Smoot low-pressure turbine of 600 kilowatts capacity.

The Rateau Steam Regenerator Company has installed eight plants with capacity ranging from 500 to 2000 kilowatts.

In Europe, plants aggregating a capacity of 400,000 kilowatts are operating in conjunction with Rateau steam regenerators.

Old Hal Mossback struck his firm fer a raise tother day an' he got it. Th' Old Man kicked him so dumd hard thet he raised 'bout six feet frum th' floor of the engine room.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Induction Motor Hand Starters

By R. H. Fenkhausen

Starters for Large Motors

For motors of 75 horsepower and above, some type of starter giving several starting positions is desirable in order that the line disturbance may be reduced. Fig. 14 shows a starter of this class having five starting points. The lever is moved progressively from the "off" position through the several starting positions (exactly as in a direct-current starting box) to the running position, where it is locked against backward movement. To stop the motor it is necessary only to move the lever one notch further in the same direction as when starting, which carries it to the "off" position. The internal connections of this starter are shown in Fig. 15. The contacts for each phase or circuit division are mounted on a separate drum, and to prevent short-circuiting successive taps of the autotransformers, the contacts of one tap are opened before those of the next are closed. The drums are so geared together that only one phase of the motor winding is open-circuited at any given moment, so that at least one phase of the motor is receiving current at any time, which prevents a drop in speed between starting points in case the operating handle is moved too slowly from point to point.

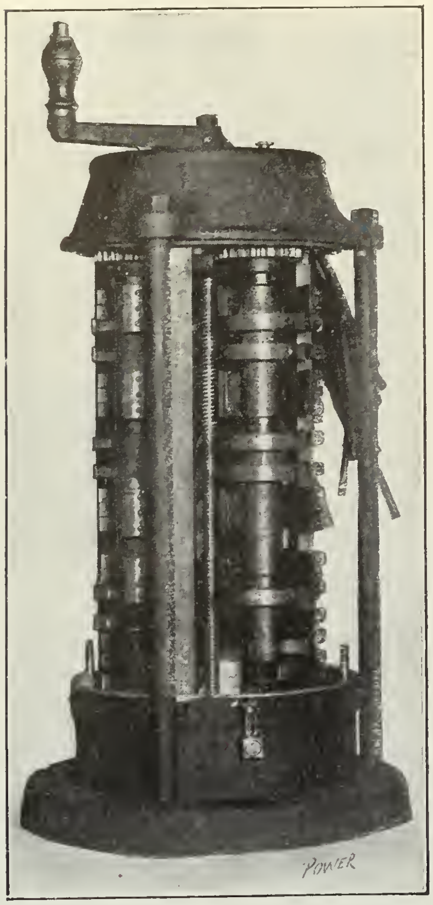

Fig. 14. Drum and Finger Type of Starter

For 2200-volt motors the circuit is never opened between successive points, and to limit the current which would necessarily flow if two autotransformer taps were short-circuited, a "preventive" resistance is automatically inserted in the circuit between each pair of taps just before they are connected.

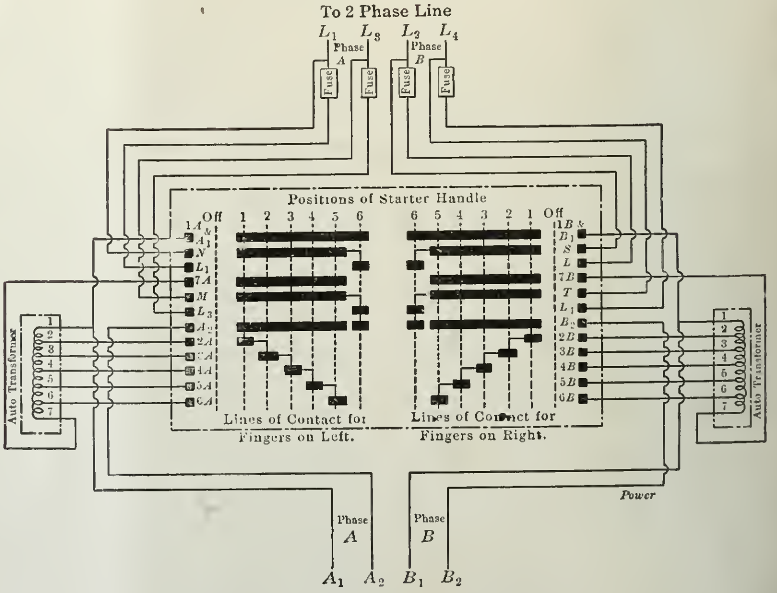

Fig. 15. Circuit Diagram for Fig. 14

Starters of the type shown in Fig. 14 are built with the switch contacts submerged in oil contained in a sheet-iron tank. To gain access to the contacts the oil tank must be lowered, which is accomplished by shifting a clutch and turning the operating handle backward. It is essential, however, that a pit be provided below the starter into which the tank may be lowered. The reason for this will be apparent from the following incident. An engineer once installed a starter of this type and poured in the oil at the filling plug provided at the top, without giving the removability of the oil tank a thought. The starter was used in connection with a motor driving an important pump, and when one morning some of the fingers failed to make contact with the drum, it was impossible to gain access to them because there was not room to lower the tank; consequently, the pump could not be started. Luckily plenty of help was available, so with the writer's assistance a pit was hastily dug, the base of the starter being supported by a couple of timbers across the excavation, and the tank was lowered. New fingers were soon in place and the pump was running twenty-five minutes after the telephone call for assistance had been received. You may be sure that pits were immediately provided under all the starters of this type in the plant. The best method to employ is to sink a short length of oil-well casing beneath the starter just large enough to clear the tank, and then fill the bottom of the casing and the space surrounding it with concrete, in the top of which the foundation bolts for the starter base may be placed.

Motor Containing its Starter

Constant-speed induction motors with wound rotors and the starting resistance mounted on the rotor spider are self-contained and require no external starting device other than the main switch. The starting lever on the motor frame should, if possible, be mechanically interlocked with the main switch in some way that will prevent a careless operator from closing the main switch when the starting handle on the motor is not in the starting position.* A motor of this type is illustrated in Fig. 16.

Protective Devices

The hand controllers now built for induction motors do not embody any mechanism for overload or no-voltage release. The overload protection is obtained as a rule by the use of fuses or small circuit-breakers separate from the starter proper. No-voltage release is, however, a feature of some of the oil-break circuit-breakers now made, and it is valuable to large motors, because if the power supply should be interrupted it will prevent the motor from starting under full voltage when the power is again turned on.

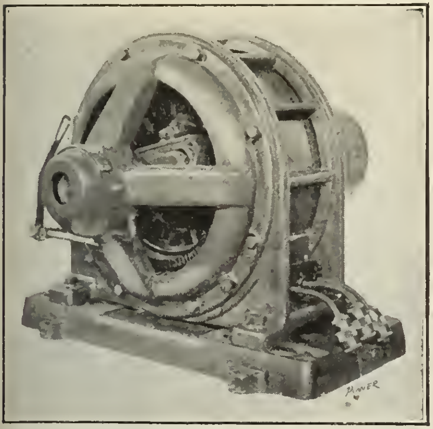

Fig. 16. Motor with Starter in Rotor Spider

When selecting fuse blocks for the protection of an induction motor, the National Underwriters' Code rating of fuses should be considered. The size of fuse-block changes for every few capacities and the range of fuses suitable for each size of block is given below. A different size of block is used also for 250 and 600 volts.

1 = 0 to 30 amperes, ferrule contacts.
2 = 35 to 60 amperes, ferrule contacts.
3 = 65 to 100 amperes, knifeblade contacts.
4 = 100 to 200 amperes, knifeblade contacts.
5 — 200 to 400 amperes, knifeblade contacts.

If, for instance, a 100-ampere fuse is required, a 100- to 200-ampere block should be installed with a 105-ampere fuse. If the starting current of the motor is greater than estimated it is a simple matter to insert a larger fuse, but if a 100-ampere block were installed a larger fuse could not be used and the fuse blocks would have to be changed.

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*One way of doing this was described in my article on page 807 of the issue of May 3, 1910.

Primer of Electricity

By Cecil P. Poole

Dynamo Field Excitation

In the lessons of November 16 and December 7, 1909, and January 4 of this year the principles of the electromagnet were explained. The student should "brush up" on these principles now because the field magnet of every dynamo (excepting a few special types of very small output) is a big electromagnet, and the principles explained in the lessons mentioned apply to these big magnets exactly as they do to small ones.

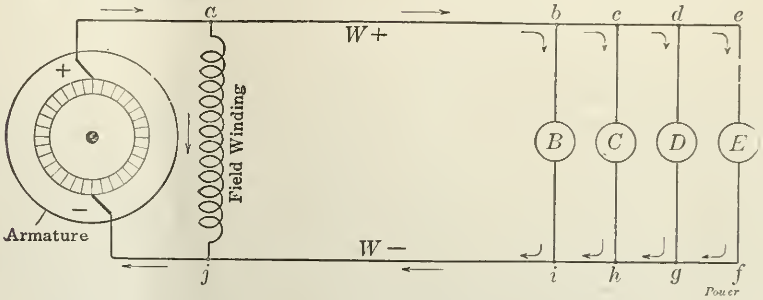

Fig. 70. Elementary Diagram of Constant Potential System

The field magnet of a dynamo is given that name because it creates and maintains the magnetic field through which the armature conductors must pass in order to generate electromotive force. Many writers use the term "field" indiscriminately to mean "field magnet," "field-magnet winding," "field-magnet coil," etc.; the word has only the one meaning, however, of a space occupied by magnetic lines of force: the magnetic field. The magnetic field of a dynamo is always located in the short spaces between the magnet polefaces and the armature, appropriately known as the "air-gaps," and around the edges of each poleface.

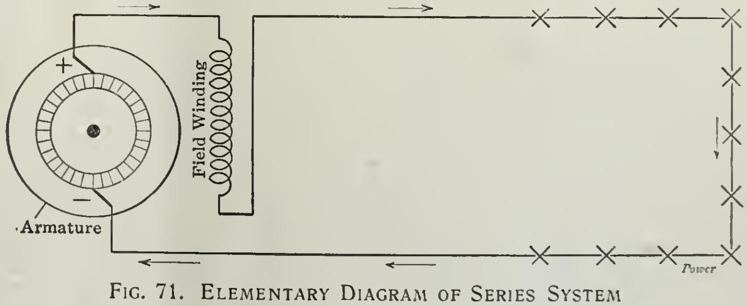

Fig. 71. Elementary Diagram of Series System

In most cases the field magnet of a dynamo is magnetized by current supplied by its own armature; such machines are called "self-exciting." As may be imagined from the sense in which it is used, "exciting" is another name for "magnetizing"; it is a much more convenient term to use, however, as will be evident as we go along and the various ways of using it develop. Moreover, there is a certain distinction between "excitation" and "magnetization"; an electromagnet is "excited" by passing a current through its winding, and in this case "excited" also means "magnetized"; but a piece of steel may be magnetized by rubbing it against a powerful magnet, and, although it is "magnetized," it is not "excited." From this it is evident that to excite really means to magnetize by means of a current passing through a winding surrounding the core of the "excited" magnet.

Of course, the current with which an electromagnet is excited may be taken from any suitable source; when the field magnet of a dynamo is excited by current supplied from some other source than its own armature, it is known as "separately excited."

Field-magnet windings of self-excited dynamos are divided into two general classes: one called "shunt" windings and the other called "series" windings. If the student remembers the early lessons on parallel and series circuits, he will understand from these two names that the shunt field winding is connected in parallel with some other circuit and the series field winding in series with some other circuit; in both cases this "other circuit" is the external or work circuit supplied by the dynamo. Fig. 70 is an elementary diagram of the connections of a shunt-wound dynamo and Fig. 71 that of a series-wound dynamo. As the diagrams indicate, the load of a shunt-wound dynamo consists of a number of receiving devices (represented in the diagram by circles) connected in parallel with each other, and the field winding is connected in parallel with these devices, while the load supplied by the series-wound dynamo consists of devices (represented in this diagram by crosses) connected in series with each other and with the field winding of the dynamo.

THE SHUNT-WOUND DYNAMO

The shunt-wound machine is used more generally than the series-wound machine, for reasons which will appear as the student goes along. Referring to Fig. 70, the arrows indicate the direction in which the current passes through the various paths in the system. Starting at the positive brush of the dynamo (marked +) the full armature current passes to the point *a*, where a small portion of it splits off to go through the field winding; the remainder goes on to the point *b*, where another portion leaves the main wire and passes through the device *B*, indicated by the circle; this happens successively at the points *c* and *d*, and at *e* what is left of the original current goes through the device *E* to the other wire of the circuit at *f*, and joins the current from *D* at *g*; at *h* the combined currents from *f* and *g* join the current from *C*, and at *i* the current in the main wire $W-$ is further augmented by the current from *B*. Between the points *i* and *j* the current in this wire is the sum of the currents passing through the four devices, and at *j* it is increased to the full armature current by the small current coming through the field winding.

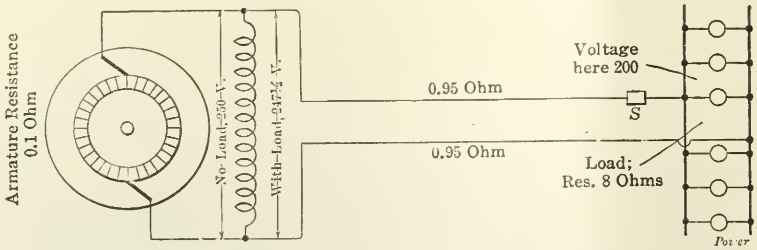

FIG. 72. ILLUSTRATING EFFECT OF LINE DROP

If the main wires $W+$ and $W-$ to which the load is connected be regarded for the moment as having no resistance, it will be evident that the electrical pressure, or difference of potential, from *a* to *j* is exactly the same as that from *b* to *i*, that from *c* to *h*, and so on; in other words, the potential between the two main wires $W+$ and $W-$ is the same at all points of those wires, and is equal to the electromotive force of the armature. For this reason, a parallel circuit supplied by a shunt-wound dynamo is called a "constant-potential" circuit.

When the armature of a shunt-wound dynamo is driven at a constant rate of speed, if the resistance of the field winding remains unchanged, the voltage or difference of potential at the brushes of the machine remains almost constant no matter what the load may be; if the armature winding had no resistance, the voltage would remain absolutely constant. The reason for this is that the voltage of a dynamo is determined by the speed of the armature and the strength of the magnetic field; the strength of the field depends entirely on the current passing through the field winding, and this depends on the voltage at the terminals of the winding and the resistance of the winding. It is obvious, then, that so long as the resistance of the field winding is not changed the current in the winding depends on the volts at its terminals, and if the armature speed is not changed, there is nothing to change the voltage. In practice, the voltage at the brushes of a shunt-wound dynamo drops slightly as the load (and consequently the armature current) increases, because the resistance of the armature winding and brushes uses up more and more of the generated voltage as the armature current increases, leaving less and less voltage for the outside circuit. Because of the property just described, of maintaining almost a constant voltage under all loads, the shunt-wound dynamo is frequently called a "constant-potential" dynamo, although it is not strictly one.

The current that passes through the field winding of a shunt-wound dynamo is a small proportion of the total armature current at full load; in small machines it may be as much as 15 or 20 per cent. of the full-load armature current, but in machines of ordinary and large sizes it is usually from 10 to 2 per cent. of the total current, according to the size of the dynamo. Hence, the field winding constitutes a shunt or bypass to the main or load circuit, and that accounts for the name "shunt winding."

Because of the fact that circuit wires have resistance, and therefore cause a "drop" in voltage between the dynamo and the receiving devices, it is necessary to adjust the voltage of shunt-wound dynamos instead of allowing them to maintain a practically constant voltage at the brushes. Reference to Fig. 72 will assist in making this clear. Suppose that the dynamo, when running at normal speed and with the load cut off by opening the switch *S*, generates 250 volts and that the resistances of the various parts of the system are as stated on the diagram. Now, suppose the switch *S* to be closed; the dynamo will be sending current through a total of 10 ohms (ignoring the field winding), and the current therefore will be

$$\frac{250}{10} = 25 \text{ amperes}$$

The voltage used up in driving the current through the armature would be

$$25 \times 0.1 = 2\tfrac{1}{2} \text{ volts,}$$

and that used up in driving the current through the two line wires would be

$$1.9 \times 25 = 47\tfrac{1}{2} \text{ volts,}$$

making a total loss of 50 volts in the armature and the main-circuit wires. This leaves only 200 volts at the load, which should receive 250 volts. If the load should be increased until its resistance is only 3 ohms instead of 8,* the total resistance of the whole system would be 3 + 1.9 + 0.1 = 5 ohms, and the current then would be

$$\frac{250}{5} = 50 \text{ amperes}$$

The drop in the armature and brushes would be

$$50 \times 0.1 = 5 \text{ volts}$$

and that in the main-circuit wires would be

$$50 \times 1.9 = 95 \text{ volts,}$$

making a total of 100 volts. This would leave only 150 volts at the load.

The assumed conditions just cited were made much worse than they would be in actual work in order to emphasize the disadvantage of voltage drop in the connecting wires of a circuit, but under real operating conditions with the full load connected the drop might easily be 10

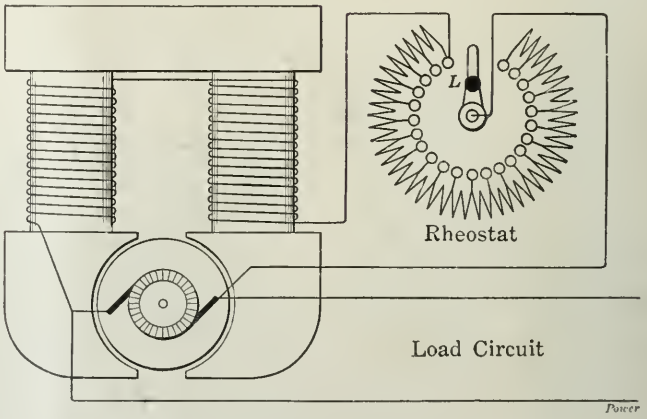

FIG 73. SHUNT-WOUND DYNAMO CONNECTIONS

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*Increasing the load in a parallel circuit is done by adding more devices in parallel, which reduces the joint resistance of the total load.

per cent. of the voltage that the dynamo would give with no load at all; that is, the voltage at the load might be 225 volts instead of 250.

In order to correct this disadvantage, the field winding of the dynamo is made "stronger" than would be required to produce the rated voltage of the machine, and an adjustable resistance, known as a "rheostat," is inserted in the field-winding circuit, as represented diagrammatically in Fig. 73. The zig-zag line arranged in a circle represents a high-resistance conductor connected at intervals to contact buttons; a lever *L* is pivoted at the center of the circle and arranged so that its free end will come into contact with the buttons when the lever is swung around the pivot.

As it is shown in the diagram, the lever *L* does not touch any of the buttons, and the circuit of the field winding is therefore open; if the lever be moved to touch the first button on the right, the entire resistance will be connected in series with the field winding; moving the lever to the next button will leave out the first section of the resistance conductor and therefore the resistance of the whole field circuit will be reduced. Each "step" of travel of the rheostat lever puts an additional section of resistance out of circuit, until the last button is reached, when the rheostat conductor is all cut out and the current in the field winding is maximum. The resistance of the regulating rheostat is usually two or three times as great as the resistance of the field winding, and the range of field-strength adjustment is therefore ample for all ordinary conditions.

Stability and Maximum Voltage

If the magnetism of a dynamo field magnet increased directly in proportion to the strength of current passed through the field winding a shunt-wound dynamo would be very unstable—that is, its electromotive force would not tend to stay at any given value; if it happened to increase, the current in the field winding would increase proportionately, this would increase the field strength proportionately, and that would cause another increase in the electromotive force, which would increase the exciting current again, and so on, until the voltage rose so high as to destroy the insulation of the machine. Fortunately, when the magnetic density in iron or steel reaches a fairly high value (about 65,000 lines per square inch in the best grades of steel or wrought iron) increasing the exciting current does not produce a proportionate increase in magnetic flux. Therefore, when a dynamo is running at normal speed, if its field circuit be closed it will begin to generate a feeble electromotive force and this will increase rapidly until the magnetic density in the iron and steel reaches such a point that a large increase in voltage at the terminals of the field winding will produce only a small increase in magnetic flux; the electromotive force cannot increase any further automatically because the requisite increase in magnetic flux cannot be caused by the increase in exciting current that it itself would produce.

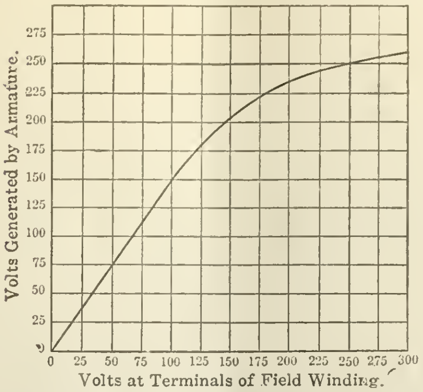


Fig. 74. Excitation Curve of Shunt-wound Dynamo

For example, suppose that in order to produce enough magnetic flux to generate 250 volts in the armature, an exciting current of 5 amperes were required in the field winding and the resistance of the winding were 50 ohms; then the voltage at the terminals of the winding would have to be

$$50 \times 5 = 250 \text{ volts,}$$

or exactly the same as the generated voltage. Now suppose also that in order to generate 251 volts the exciting current had to be increased to 5.1 amperes; as the resistance of the field winding is 50 ohms, a voltage of

$$50 \times 5.1 = 255$$

would be required, and as the armature would only give 251 volts with this additional excitation, it would be impossible for it to supply the 255 required, and the voltage would remain at 250.

This effect may be better understood, perhaps, by reference to Fig. 74, which is an excitation curve for an actual dynamo rated at 230 volts. Starting at any number on the scale of generated volts and tracing over to the curve and thence down to the scale at the bottom will show the number of volts required at the terminals of the field winding to make the armature generate the number of volts at which the tracing began. For example, in the machine tested, to generate 125 volts it was necessary to apply 81.3 volts at the field-winding terminals; a voltage of 162 at the field-winding terminals caused the armature to produce 212.5 volts, and so on, *the generated volts being higher than the exciting volts* at all points of the curve until 250 was reached; then the two voltages were the same, and as any greater generated voltage required more exciting volts than the armature could generate, the machine could not produce more than 250 volts when operated self-exciting. The test from which this curve was plotted was made by exciting the field magnet with current taken from another circuit instead of from its own armature; by doing this, the test could be carried beyond the ability of the machine. To generate 260 volts in the armature required 300 volts at the terminals of the field winding.

From the foregoing it will be evident that *the maximum possible voltage of any shunt-wound self-exciting dynamo is that which is required at the terminals of the field winding to produce sufficient magnetic flux to enable the armature to generate the same voltage.*

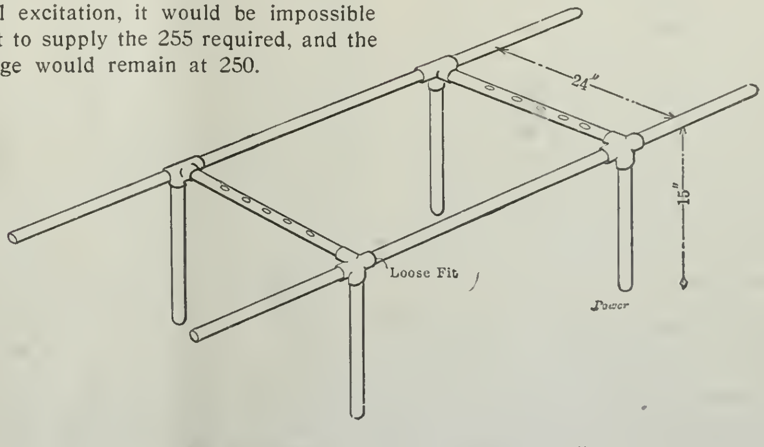
LETTERS

An Armature Stretcher

The accompanying sketch illustrates a sketch used for carrying damaged armatures to and from the shop. Although it is used for this purpose more than any other, it is handy also for carrying other things, such as short shafting. The sketch shows the construction. The legs and cross pieces are made of 1¼-inch pipe and the handles of 1-inch pipe. The handles fit loosely in the crosses so that the pairs of legs may be set at any distance apart. The ⅜-inch holes in the cross pieces are for the insertion of pins to keep the armature from rolling.

J. J. O'Brien.

Buffalo, N. Y.



An Armature Carrying "Stretcher"

Gas Power Department

Elementary Lectures on the Gas Producer

By Cecil P. Poole

Producer Gas

What is commonly called producer gas is really a mixture of several gases, as the reader might suppose from looking at Fig. 12 and reading the third lecture of this series.* As the incandescent zone of the generator makes carbonic-acid gas and the decomposition or second zone makes carbonic oxide and hydrogen the gases leaving the generator must include these three, unless all the carbon dioxide (CO_2) is used up in the second zone in the production of carbonic oxide (CO), which is practically impossible. Then, again, the air that supports combustion in the fire zone contains nitrogen, which will not burn or combine with any of the other gases present in a generator, and this, too, passes out through the delivery pipe.

The exact composition of the delivered gases depends largely on the kind of coal used in the generator. The soft coals all contain considerable quantities of hydrocarbons which are readily distilled out in the top zone of the generator, and the gas made by such coals therefore contains these hydrocarbons. They are marsh gas, consisting of one part carbon and four parts hydrogen, and olefiant gases containing two parts of hydrogen to one of carbon, by volume. There are also some hydrocarbons that are not "fixed" when they are distilled out, but condense after they get away from the heat of the generator, forming tar, and some which are "broken up" by excessive heat in the upper part of the generator, the hydrogen passing off as a separate gas and the carbon being carried off in the form of fine dust, commonly called lampblack.

The hard coals (anthracite and semi-anthracite) contain very little of hydrocarbons and what there are are fixed gases, that is, they do not condense into tar when cooled. Consequently, the gas made from hard coals contains only a small proportion of hydrocarbons, no tarry matter and no lampblack. The percentage of hydrogen in anthracite gas is also slightly less than in gas made from bituminous coals and lignites, partly because the coal itself contains less and partly because less steam is required in the air supply to keep the temperature of the fuel bed within the desired limits.

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

The proportion of free hydrogen in the delivered gas depends largely on the amount of steam admitted to the fuel bed, per pound of coal "gasified," and the steam supply depends on the character of the fuel and the type of generator used. Roughly, under average conditions, the total amount of steam passed through the fuel bed, including that produced from the moisture in the coal, should not be less than 6 per cent. nor more than 10 per cent. of the weight of air passed through the fuel bed in the same length of time. In estimating the composition and heat contents on the delivered gas, it is customary to assume that one-fourth of the oxygen used in forming CO_2 and CO is supplied by steam and three-fourths by air, and the amount of steam admitted from the vaporizer is less, of course, when the coal contains much moisture than when it is fairly dry. For example, a hard coal containing 4 per cent. of moisture would require twice as much steam from the vaporizer as a soft coal containing 8 per cent. of moisture, per pound of carbon gasified, and a lignite containing 16 per cent. of moisture would need only half as much steam per pound of carbon gasified as the soft coal with 8 per cent. of moisture, *provided the percentage of fixed carbon were the same in all three.* This, of course, is an absurd proviso, because the percentage of fixed carbon in anthracite cannot possibly be the same as that in bituminous coal, and the percentage in the latter is always more than in lignite.

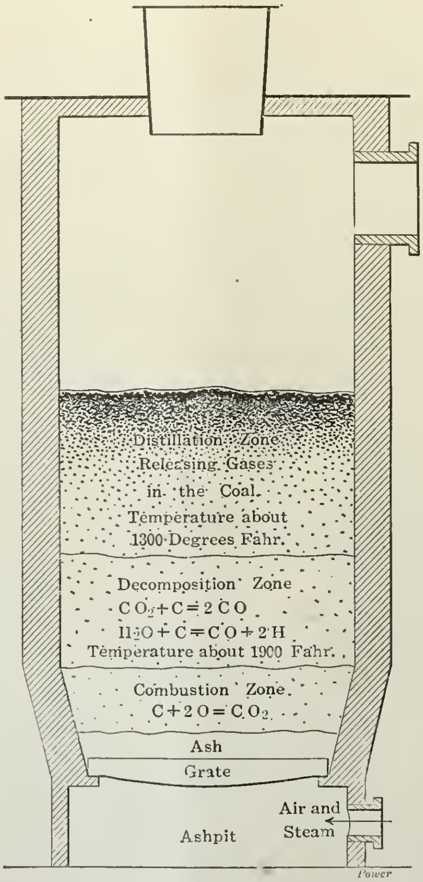


Fig. 12. Approximate Character of the Zones of a Fuel Bed When Steam Is Admitted with the Air

To get at a real comparison of the steam required by different grades of coal, therefore, both the amount of fixed carbon and the amount of moisture contained in the coals must be considered. Thus, if an anthracite coal contained 80 per cent. of fixed carbon and 4 per cent. of moisture, and if one-tenth of the carbon were burned to CO_2 and nine-tenths used to make CO, the process would work out as follows for 100 pounds of coal:

8 pounds carbon burned to CO_2 require	$18\frac{2}{3}$ lbs. oxygen
72 pounds carbon burned to CO require	96 lbs. oxygen
Total	$114\frac{2}{3}$ lbs. oxygen
One-fourth of the oxygen supplied by water	$28\frac{2}{3}$ lbs. oxygen
Total weight of water required to supply oxygen $= 28\frac{2}{3} \times 9 \div 8 =$	$32\frac{1}{4}$ lbs
Moisture in the coal	4 lbs.
Steam required to be supplied by vaporizer	$28\frac{1}{4}$ lbs.
Oxygen to be supplied in the air	86 lbs.
Air required	$345\frac{1}{4}$ lbs.

Proportion of steam in the air supply $= 28.25 \div 345.25 = 0.0818$, or about 8.2 per cent.

If a grade of bituminous coal containing 69 per cent. of fixed carbon and 4 per cent. of moisture be considered on the same basis, it will be found that 24½ pounds of steam are required from the vaporizer for each 100 pounds of coal, as compared with 28¼ for the anthracite having the same percentage of moisture; and considering a grade of lignite to be used in which there are 36 per cent. of fixed carbon and 15 per cent. of moisture, it will be found that *no steam at all* is required from the vaporizer, because the moisture in the lignite is sufficient to make all the steam needed.

These examples, it must be remembered, are based on the assumed proportions of one-fourth oxygen supplied in steam and three-fourths in air, but there is nothing hard and fast about this division of responsibility; the air may supply as much as 80 or 85 per cent. of the oxygen and as little as 65 or 70 per cent. The proportions assumed were taken because they are a fair average, and some definite proportions had to be assumed before any estimate could be made of the steam supply.

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*August 9, 1910.

TABLE 6. GAS FROM LIGNITES.

	Composition.	Colorado Lignite.	Texas Lignite.	Peruvian Lignite.
Fuel	Fixed carbon	41.65	29.76	31.00
	Volatile matter	32.26	29.25	38.40
	Moisture	20.24	33.71	18.75
	Sulphur	0.60	0.53	3.25
	Ash	5.65	6.75	8.60
Gas	Carbon monoxide	17.38	18.22	22.15
	Hydrogen	11.05	9.63	9.67
	Hydrocarbons	5.00	4.81	2.06
	Carbon dioxide	10.12	9.61	6.47
	Nitrogen	56.45	57.73	59.65

While it is impracticable to draw any positive comparisons between the results obtained with hard coals and those given by soft coals and lignites, because of the wide variation in operating conditions and, consequently, the wide differences between results obtained with exactly the same coal, it is interesting and instructive to consider some representative performances of producers using different grades of fuel. Table 6 gives the analyses of gases made from three kinds of lignite, together with the proximate analyses of the lignites themselves. The composition of the gas is given by volume, which is customary, but the composition of the fuel is by weight, as also customary.

TABLE 7. GASES FROM HARD AND SOFT COALS.

Constituents.	Bituminous.	Anthracite.
Carbon monoxide	22%	26%
Hydrogen	18%	15%
Hydrocarbons*	3%	2%
Carbon dioxide	4%	7%
Nitrogen	53%	50%

*Not counting tar.

It will be noticed that all three of the gas analyses show more than half of he delivered gas to be nitrogen. This is almost always the case, no matter what kind of coal is used, because the largest part of the oxygen required for combusion and making CO is supplied in the air, as already explained, and for each pound of oxygen thus obtained there must also pass through the generator about 3.3 pounds of nitrogen, air being composed of about 76 parts nitrogen to about 23 of oxygen, by weight (the proportions of oxygen and nitrogen in air have never been determined precisely).

There is not as much difference between the composition of the gas made from bituminous coals and that made from hard coals as there is between the lignite gases and others, except as to the tarry matter and lampblack already mentioned. These are not included in the gas analysis, however, because they are not gaseous when cooled. Table 7 gives specimen analyses of gases made from hard and soft coals. Of course, these are not to be considered as a definite comparison of the two kinds of gas, for the reasons previously given.

Heat Value of Producer Gas

The number of heat units contained in a pound or a cubic foot of producer gas varies considerably. Gas made from lignite and soft coals contains from 140 to 160 heat units per cubic foot (the temperature being assumed to be 32 degrees, Fahrenheit); that made from anthracite coal ranges from about 120 to 145 heat units per cubic foot. The higher heat value of the gas made from soft coals is due to the greater proportion of hydrocarbons and the slightly greater percentage of hydrogen usually in the gas.

The heat value of the composite gas* delivered by the generator depends, of course, on the proportions of the various combustible gases contained in the composite gas. As in the case of coal, the heat value is the number of heat units liberated by combustion, but it is common practice to figure the heat value of a cubic foot of gas instead of a pound, as is done in the case of coal. As with coal, the heat value of a gas or a mixture of gases can be determined more accurately by means of an apparatus known as a calorimeter, but it is frequently desirable to get at an approximate idea of the heat value without going to the trouble of making a physical test. For purposes of study, too, the estimated heat value, derived from an analysis of the gas, is usually the only one that can be made, because the gas being studied exists only on paper.

As gases expand when heated, and the thermal units contained depend ultimately on the weight of each constituent gas, the heat value is usually based on a temperature of 32 degrees Fahrenheit, not because gases are commonly used at any such temperature, but because that is the temperature at which ice melts and it is, therefore, easily determined with exactness. Unless some other temperature is specified, therefore, the reader will understand that all gas heat values are supposed to be those of a cubic foot of gas at 32 degrees.

Before attempting to figure heat values of composite gases, it is necessary to know the heat values of the elementary and compound gases of which they are composed. There are only four combustible gases to be considered in producer work; namely, hydrogen (H), marsh gas or methane (CH_4), ethylene (C_2H_4) and carbon monoxide (CO); in practice, marsh gas and ethylene are not usually considered separately because there is such a small quantity of the latter in the producer gas that it is not worth the trouble to figure on it separately from the marsh gas; the two are ordinarily lumped together under the head of "fixed hydrocarbons."

Table 8 gives the properties of the four combustible gases so far as is necessary in estimating the heat value and weight of a composite gas containing them, and the air required for combustion. From the data in this table the heat value, combustion air and products of combustion may be readily reckoned

TABLE 8. COMBUSTION DATA OF GASES.
At 32° Fahrenheit and Atmospheric Pressure.

Gas.	Symbol.	Pounds per Cubic Foot.	B.t.u. per Cubic Foot.	Oxygen Used.*	Air Used.*	Products of Combustion.			
						Water Vapor.	Carbon Dioxide.	Nitrogen.	Total.
Hydrogen	H	0.005589	347	0.5	2.4	1.0	0.0	1.9	2.9
Marsh Gas	CH_4	0.04464	1072	2.0	9.6	2.0	1.0	7.6	10.6
Ethylene	C_2H_4	0.07809	1713	3.0	14.4	2.0	2.0	11.4	15.4
Carbon Monoxide	CO	0.07807	342	0.5	2.4	0.0	1.0	1.9	2.9

*Cubic feet used for combustion of each cubic foot of gas.

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*The expression "composite gas," though somewhat clumsy, is used to keep in the mind of the reader the fact that a producer makes a mixture of gases, uncombined chemically with each other, and not a compound gas, which consists of chemically combined elementary gases.

for any mixture of gases of which the proportions are known. For example, suppose a composite gas made from bituminous coal contained 18 per cent. hydrogen, 2 per cent. marsh gas, 1 per cent. ethylene and 20 per cent. of carbon monoxide; the heat value per cubic foot would be reckoned as follows:

Hydrogen, 0.18 cubic foot at 347 B.t.u. = 62.46
Marsh gas, 0.02 cubic foot at 1072 B.t.u. = 21.44
Ethylene, 0.01 cubic foot at 1713 B.t.u. = 17.13
Carb. monox., 0.2 cubic foot at 342 B.t.u. = 68.40

Total........................... 169.43

As rich a gas as this is seldom obtained in actual practice. The reason for the abnormally high heat value is the assumption of 3 per cent. of hydrocarbons: if these are assumed not to be there, the heat value of the gas immediately falls to 131 B.t.u. per cubic foot, which is a value often obtained.

The air required for the combustion of such a gas would be ascertained as follows:

For the hydrogen.........0.18 × 2.4 = 0.432
" " marsh gas...........0.02 × 9.6 = 0.192
" " ethylene............0.01 × 14.4 = 0.144
" " carbon monoxide...0.2 × 2.4 = 0.480

Total........................... 1.248

or 1¼ cubic feet of air per cubic foot of gas burned.

Repairing a Broken Crank Shaft

By J. A. Luther

A short time ago the crank shaft on our 10-horsepower engine broke near the cheek, at *A*, Fig. 1, and upon ordering a new one we were informed that it would be at least five weeks before one could be shipped. Having a forfeit job on hand that had to be completed immediately, we simply could not afford to wait, and therefore decided to repair the broken shaft in our own factory, although it was not at all an easy repair to make.

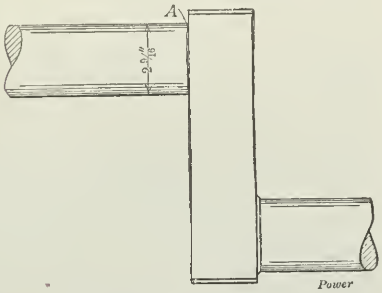


Fig. 1. The Broken Part of the Shaft

Upon examining the broken ends we found that the shaft had an old crack in it, the fresh break being only about ⅞ inch cross-section, and it must have been running for months in that condition.

How the Shaft Was Repaired

The piece of the shaft that was broken off the crank cheek was accurately trued up in a lathe and the broken end was turned to a taper 1½ inches from the end, the taper being ⅛ inch to the foot; see Fig. 2. Next, the crank cheek was drilled with a tapering hole to fit the tapered end of the broken-off shaft, as indicated at *C* in Fig. 3. During the boring process a round piece of brass, fitted tightly in the space, was inserted between the crank cheeks in line with the shaft, as at *D*, Fig. 3, to avoid springing the cheek being bored, and a weight was attached to the lathe chuck to counterbalance the crank. The entrance to the taper hole *C* in the crank cheek was bored straight for ¾ inch and was made 0.004 inch smaller than the diameter of the shaft, to get pressing fit. The tapered end of the piece of shaft was pressed home in the crank cheek and the two were pinned together with a ½-inch tapered pin through the center of the shaft (at right angles to the axis), the protruding ends of the pin being heavily upset to prevent working loose.

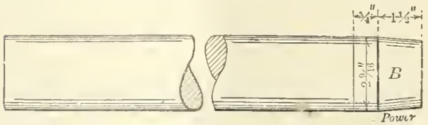


Fig. 2. Tapered End of Broken Shaft

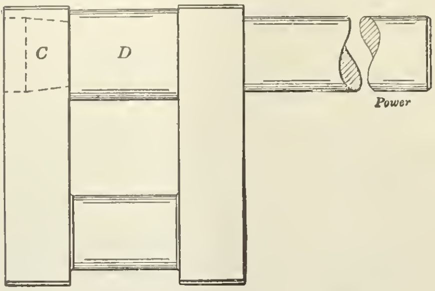


Fig. 3. Crank Braced for Boring Cheek

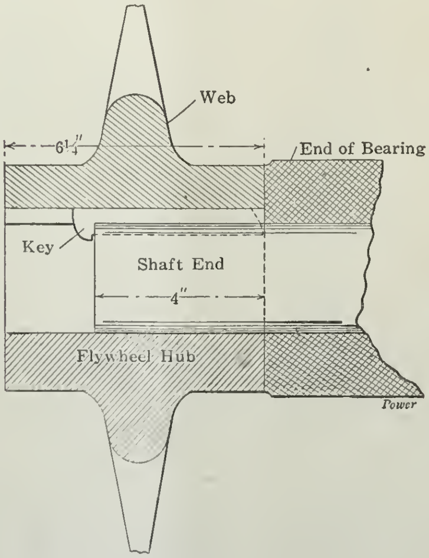


Fig. 4. Shortened End of Shaft in Flywheel Hub

The finished shaft was, of course, about 2¼ inches short on the repaired end because it had been inserted that far in the crank cheek. Consequently, it did not reach all the way through the flywheel hub when placed in position in the bearings, the result being about as indicated in Fig. 4; there was only 4 inches of shaft in the hub, which was 6¼ inches long. We expected this would give trouble, but we drove the key home solidly and started up, trusting largely to good luck.

Contrary to our apprehensions, the wheel on the repaired end of the shaft ran practically true but the one on the other end, where nothing had been changed except to drive out the old key and fit a new one, ran considerably out of true. This was eventually corrected by repeated peening.

The engine has now been running with full load for nearly three months, and apparently is as good as new. The expected trouble of the flywheel working loose on account of its short bearing on the shaft has not occurred, and we have practically stopped looking for any new complications to develop from the repaired crank shaft.

Corrosion of Water-cooled Exhaust Piping

By M. W. Utz

At the plant where I am employed there are two three-cylinder four-stroke-cycl vertical gas engines running on natura gas; each engine runs 12 hours per day as the plant runs 24 hours a day. W have a small stream of water piped t the exhaust manifold to cool the exhaust gases; as this seemed to attack th exhaust pipe, causing it to waste away i a short time, the pipe was tapped nea the exhaust manifold, making a sort o trap for the water from which it wa drained by a 1-inch pipe to the sewer.

This drain pipe and fittings were eate through by the combination of water an exhaust gases in two months, so I change to galvanized pipe and fittings, whic seem to last much longer. Before changing to galvanized pipe and fittings I trie a few galvanized nipples in the line, usin black pipe elsewhere, and when the iro pipe was eaten through the galvanize pipe was still in fairly good shape.

If any of the readers of Power hav had any similar trouble and overcon it, I wish they would explain how in the columns.

Steam Turbines Superseded Gas Engines

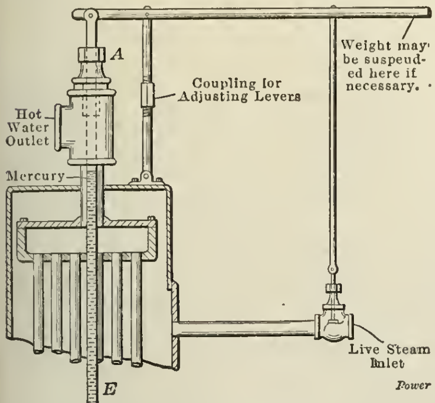
Steam turbines as prime movers the coal mines of Mitsui & Co., in Japa says *Gas and Oil Power*, are to be supe seded by gas power. Two large 320 brake horsepower Nürnberg engines be run on coke-oven gas have be ordered for the generation of electric at the Müke mines. The steam turbin will be retained as reserve units.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Temperature Regulating Valve

Some months ago a temperature-regulating valve was installed in connection with our heaters. The apparatus is patented, but the accompanying sketch shows some improvements upon the original arrangement. A ¾-inch wrought-iron pipe *E* extends from the top *A* to the bottom of the heater, passing down through the center tube. This pipe is filled with mercury which, in expanding and contracting, regulates the admission of steam through an arrangement of levers. The valve is for live-steam heating only and is entirely automatic.


A
Weight may be suspended here if necessary.
Coupling for Adjusting Levers
Hot Water Outlet
Mercury
Live Steam Inlet
E
Power

Valve Connected to Heater

James E. Noble.

Toronto, Canada.

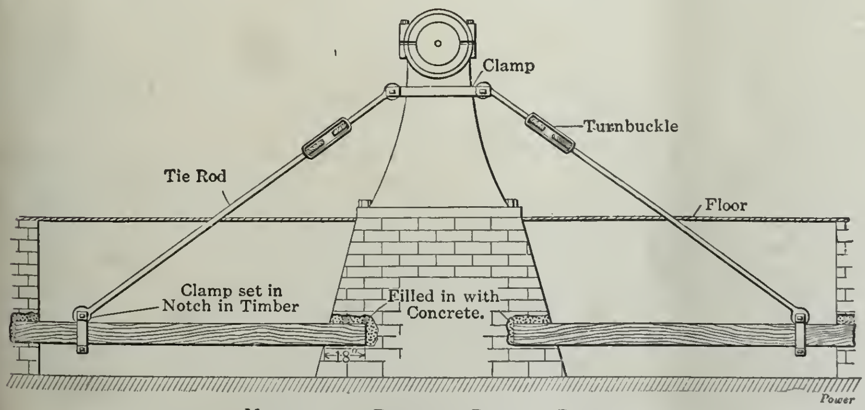
Repairing a Pillow Block

Several years ago I was engaged to make some repairs at a power plant containing three 300-horsepower tandem-compound engines. The outboard bearings of these engines were giving considerable trouble by vibrating badly, which was due partly to the form of foundation employed and partly to the fact that the plant was built on swampy ground. The accompanying sketch shows how the repairs were accomplished.

A heavy iron clamp was fitted and held by cap screws around the pedestal, close up under the bearing, and to this were attached heavy tie rods, each containing a turnbuckle near its upper end. The lower ends of the tie rods, which also terminated in "eyes," were clamped to timbers in the cellar. These timbers were placed at right angles to the shaft and were held in place by being set in notches in the foundation and walls. The timbers were then blocked up and the notches rammed full of concrete. After the concrete had set, the motion of the pillow blocks was taken up by the turnbuckles on the tie rods and the vibration ceased.

George Little.

Passaic, N. J.



Method of Bracing Pillow Block

Repairing Valve Stems

At *A*, Fig. 1, is shown the valve stem in a pump, worn where it passes through the stuffing box. As it was impossible to keep it packed while in this condition, and as the pump was used only during the winter, it being a receiver pump on a heating system, the stem was cleaned where it had worn, and filled in with solder, as shown at *B*. The solder was then filed down to the size of the stem, and the pump ran with this repair for about a year before a new stem was put in.

In Fig. 2 is shown a main-valve stem *G*, and the cutoff-valve stem *F*, concentric with the former. The cutoff-valve stem wore where it passed through the stuffing box on the main-valve stem *G*. The end *E* of the stem screws into the cutoff valve; whereas the ball, of the ball and socket joint, screws on at *C*. By cutting a thread at *C*, the same length as the thread at *E*, reversing the stem and screwing the end *C* into the valve, the stem was made to last just twice as long as usual.

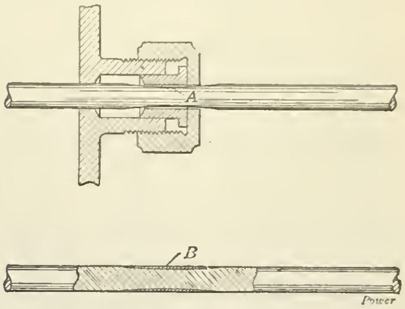


Fig. 1. Worn Portion Filled with Solder

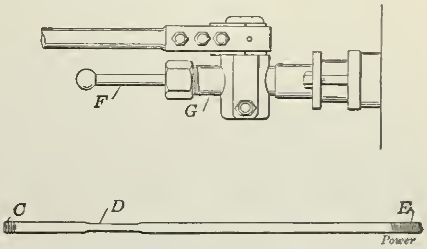


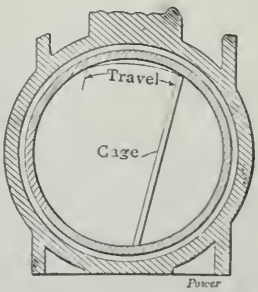
Fig. 2. Cutoff-valve Stem Reversed

R. L. Mossman.

Tampa, Fla.

Fitting Pistons to Cylinders

The accompanying table for fitting pistons to cylinders was adopted by a large marine-engine firm in England after a series of exhaustive tests, and is applicable to cylinders which have been bored, and, to a certain extent, to cylinders which have not worn too much out of round.



Gage Applied to Cylinder

In using this table a wire gage is first made of ¼- or 5/16-inch round iron and of a length equal to the diameter of the cylinder at its smallest section. The ends of the gage are tapered and are further filed to give the proper travel. By travel of the gage is meant the distance one end may be swung when the other end is stationary; see the accompanying sketch. If the piston is made of a diameter equal to the length of the gage, it will bear on almost one-half its circumference and will be comparatively tight against leakage, even if a packing ring is omitted. In making gages it is advisable to make them in duplicate, one for the machinist who is to turn the rings, and the other to be kept by the engineer.

TABLE OF GAGE TRAVELS.

Diameter, Inches.	Gage Travel, Inches.	Diameter, Inches.	Gage Travel, Inches.
1	¼	63	4
3	½	66	4
6	⅝	69	4¼
9	11/16	72	4½
12	¾	75	4¾
15	⅞	78	4¾
18	1	81	5
21	1¼	84	5¼
24	1½	87	5½
27	1¾	90	5¾
30	1¾	93	5¾
33	2	96	6
36	2¼	99	6¼
39	2½	102	6½
42	2¾	105	6¾
45	2¾	108	6¾
48	3	111	7
51	3¼	114	7¼
54	3½	117	7¼
57	3¾	120	7½
60	3¾	...	...

W. E. Rector.

Brooklyn, N. Y.

Priming a Centrifugal Pump

We had a centrifugal pump which raised water from the river to a settling tank located about 20 feet above the top of the pump. The latter, which was of the double-suction type, was direct connected to a horizontal engine which ran at 175 revolutions per minute. Each suction branch was 30 inches in diameter with a check and stop valve and was connected at right angles to the main suction line, which was 42 inches in diameter. Two other pumps of the plunger type were connected to a tee at the end of the main suction. The discharge was 24 inches with a stop valve close to the pump, and on top of the pump there was a 4-inch pipe with a valve which was always left open. On top of this valve was a nipple and a tee, from one end of which there extended a horizontal pipe that reached to the corner of the building about 12 feet distant, and from this a riser reached to within a few inches of the ceiling, the top of this pipe being 12 feet higher than the water in the settling basin. From the other end of the tee a pipe containing a valve extended to within a few feet of the sewer. This latter valve was always opened before stopping the engine, as there was usually some air in the pump. If it had not been opened the air would have gone out through the riser to the ceiling, carrying water with it. This riser should have been extended through the roof. There was no siphon on this pump as it was primed with water from the discharge pipe and was under pressure from the settling basin, the check valve on the suction preventing the water from flowing back to the river.

I had been at this plant about a month when I experienced my first trouble with the outfit. It was winter and considerable ice was coming down the river; this choked the intake and offered high resistance to the entering water which caused a greater vacuum in the suction pipe.

When the vacuum reached about 23 inches, the centrifugal pump lost its suction. I stopped the engine and let the pump fill with water from the discharge pipe, and when it was full, I ran the engine up to speed, but the pump would not regain its suction. I then went to the oiler, who had been at the plant for several years, and asked him how he was accustomed to get the suction on the pump after it had once been lost. He replied that it was necessary only to open the valve on the pipe leading from the top of the pump to the sewer, then slow down the engine and run it as slow as possible until most of the air was exhausted from the pump (which would take a couple of minutes), then speed up as quickly as possible and the pump would handle the water all right.

Another time the pump was emptied to pack the stuffing box on the shaft. After this was finished the pump was filled with water and the valve on the pipe leading to the sewer was opened to let out the air while the suction and discharge valves were open. I started the engine and ran it up to speed, but could get no suction. Then it occurred to me that I had not worked the air out before speeding up, so I slowed down the engine and did as the oiler had directed and the pump handled the water immediately. This trouble was experienced only when the vacuum on the suction was over 15 inches.

My theory of this rather peculiar performance is that there was some air trapped in some of the vanes of the impeller when the pump was primed and when the engine was started this air did not get out, unless the engine was run at a very slow speed for a while. The water being heavier than the air, the centrifugal force forced it out of the impeller, leaving the air, which expanded and prevented the pump from raising the water.

K. Lawrence.

Kansas City, Mo.

Placing the Blame for Explosions

Every time a fatal boiler explosion occurs in the United States, the papers condemn the laws that cover such subjects and call for laws to be passed that will place the blame where it belongs.

Frequently the men who write these discussions hold up the English law as superior to our own.

In a recent boiler explosion in which the fireman was killed, the coroner's jury came to a verdict of death by negligence, caused by the fireman putting ninety pounds pressure on a boiler that was good for only about forty. In commenting upon this, it was stated that this could not have occurred under the laws of England, for the English Board of Trade would have fixed the blame without fear or favor and the guilty party would have suffered, whether it were the inspector, the boilermaker, the man who owned the boiler, the engineer or the fireman.

But to show that it is about the same everywhere, that the man with the least pull is the man to shoulder the blame, I will relate a case which I observed some years ago.

At the time I was working at the plant of an oil company, about ten miles from Edinburgh, in the capacity of still tender. They had a plant of five Lancashire boilers carrying 150 pounds steam pressure. These boilers were used to supply steam to the stills and to run the pumps. The five were in a battery and connected at the front to a common blowoff pipe (as per sketch) which was located in a trench below the floor, and this trench was covered with an iron plate. In order to blow down the boilers, the fireman removed the plate, put a long-handled box wrench on the cock and, standing with one foot on each side of the trench, opened the blowoff valve.

There were two firemen, and it was their duty to blow the boilers down every shift. They would first open the valve at the far end of the main and then one would take No. 1 boiler and the other No. 2 boiler and in this way they would blow down all of the boilers one at a time.

This they had done a hundred times when one morning the elbow between the main and the blowoff valve on No. 1 boiler burst and one of the firemen was literally boiled alive. Of course, the Board of Trade was prompt in sending what they call in Scotland a "Com-a-tee" to find the cause and place the blame where it belonged. They placed a steam gage on the main between boilers No. 1 and No. 2, closed the valve at the far end of the main and then closed first one and then the other of the whole five blowoff cocks, to prove that the pipe would stand the full boiler pressure. This, however, was after a new elbow had been put in to replace the broken one. The pipe withstood the shock and this cleared the firm that installed the boilers and piping.

The committee then turned in a report that the fireman was to blame for opening the blowoff too soon. How they arrived at this verdict I cannot see, for upon investigation just after the explosion, both cocks were found closed. Furthermore, they did not look for the burst elbow nor ask any questions along that line. If they had, they would have found that the elbow gave out along an old crack, which, to my knowledge, had been reported a dozen times. But as the pipe was supposed to be under pressure only when blowing down, there was no heed taken of the report. At the inquest the other fireman stated these facts but it was of no use; someone had to be the goat and the dead man was tagged.

PATRICK MULHEARN.

Adams, Mass.

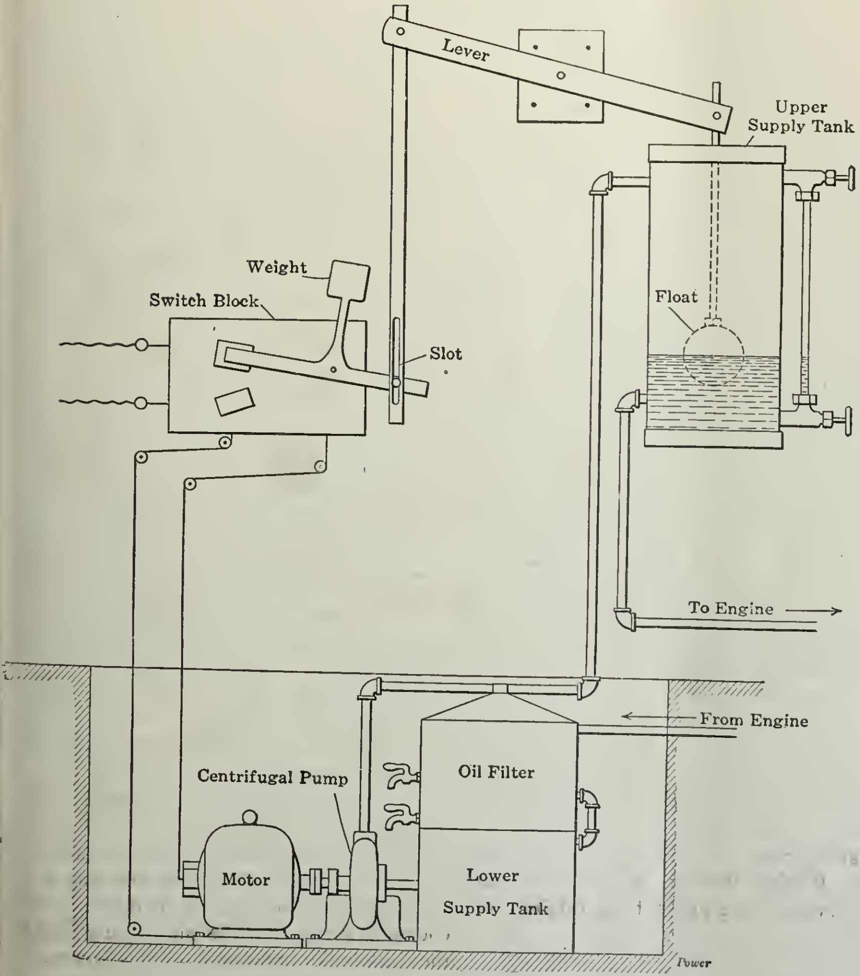
Automatically Operated Oil Pump

During a recent visit to a small power plant I came across an ingenious oiling system which was altogether homemade and seemed to operate perfectly.

A supply tank was attached to the wall near the ceiling, and from this the oil flowed by gravity to the engine. The return from the engine was piped to a filter located in a pit under the floor, and from here the oil filtered into another supply tank. By means of a motor-driven centrifugal pump the oil was forced from the lower tank into the upper supply tank. A float in the latter tank was arranged so as to make or break the contact required to start or stop the motor. The arrangements of this float and automatic switch are clearly shown in the accompanying sketch.

EDWARD T. BINNS.

Philadelphia, Penn.



DEVICE FOR AUTOMATICALLY CONTROLLING OIL PUMP

Kerosene in Boilers

Upon taking charge of my present plant, which is a small Government plant at Riverside school, Okla., I found the heating surfaces of the boilers coated with from ¼ to ½ inch of scale. In fact, I was afraid of their condition, so reduced the steam pressure from 100 to 60 pounds.

I reported the matter to headquarters at Washington, but the only answer which I received was that it was a problem for the engineers. I analyzed the water and talked boiler compounds, but received no reply, so I attached an appliance to the boilers whereby they would receive about one pint of kerosene oil every day. I started this June 1, 1910, and at present the boilers are entirely free from scale.

The superintendent had the engineer from the Fort Hill school come and look over conditions. He contended that kerosene oil should never be applied to boilers. However, I have used kerosene oil in boilers for years with success and the only objection I find is that it takes a little more valve oil. The boilers were tested with cold water at 150 pounds per square inch after they were entirely free from scale and found to be all right. The safety valves were then set back to 100 pounds, which enables me to carry the load successfully.

V. K. STANLEY.

Anadarko, Okla.

Incompressibility of Water

The popular idea that water is incompressible is true in a limited sense only and is probably due to a comparison of this property with that of the gases or other compressible substances. The compressibility of water can readily be shown when comparison is made with such materials as timber, cast iron and steel.

Using the value 0.00005 as the decrease per unit of volume of water due to a pressure of one atmosphere (14.7 pounds) and determining the ratio of this pressure on the unit volume to the deformation it produces, gives

$$\frac{14.7}{0.00005} = 294,000$$

This ratio is the modulus of elasticity of the water.

Now the average values of the modulus of elasticity of some of the structural materials, as given by handbooks, are 1,500,000 for timber, 15,000,000 for cast iron, 30,000,000 for steel.

A comparison of these values with that of 294,000 as determined for water shows that the compressibility of water is five times that of timber, fifty times that of cast iron and one hundred times that of steel.

GUY WISE.

Philadelphia, Penn.

Recently at Telluride, Colo., a mountain rat shut down the Ilium station of the power company for a few minutes and threw the city into darkness. The rat had been wandering about the station for some time, causing trouble, and one night, while standing with his hind feet on a steel rafter, he placed his forefeet on a high-voltage wire. Only the inquisitive rodent's skeleton remains to tell the tale.

Have all defects and leaks repaired without delay. It is always best to be on the safe side.

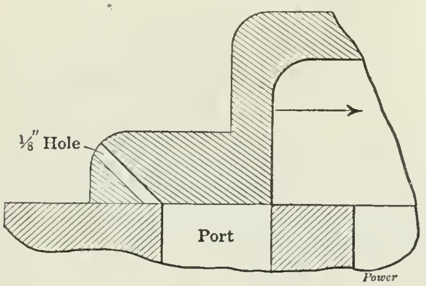
Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Condensing and Noncondensing

Referring to H. W. Benton's article on valve setting for condensing and noncondensing engines, in the September 20 issue, I thoroughly agree with him in that a difference is required, for the purpose of smooth running if for nothing else. However, he does not state the reason for his conclusions.

A number of years ago I was called to a power plant to cure a very annoying knock which occurred at both ends of the engine stroke. The engine was about 500 horsepower in size and running condensing. The knock was not only annoying, but the shocks hammered the crank-pin boxes out of shape and caused heating. A liberal advance of the eccentric cured the noise, but the shape of the diagrams was changed so much that the engineer suggested that the trouble be remedied in some other way if possible.



Location of Holes in Valve

After thinking the matter over, I suggested that a number of ⅛-inch holes be drilled in the admission edge of the valve, as shown in the accompanying figure, using care that the holes do not reach far enough on the face of the valve to allow the steam to pass out of the exhaust, before the exhaust edge of the valve covered the port. To make the matter clear, I have shown a section of a plain slide valve with the holes located as stated. We had to experiment for a time to find out just how many holes would be required, but we cured the knock.

Another case was that of an engine fitted with a slide valve with a Meyers cutoff controlled by a right- and left-hand screw, operated by a handwheel. The exhaust piping was arranged with a three-way valve so that a change from noncondensing to condensing could be made without stopping the engine.

When running noncondensing, the engine ran without the slightest jar, but as soon as the three-way valve was thrown over and the engine brought under the influence of the vacuum the knock was terrific.

To cure this case the cylinder clearance was reduced as much as possible with filling pieces, and a new main valve with additional exhaust lap was put on. The main eccentric was advanced as far as possible without interfering too much with the important operation of the valve.

A leaky exhaust valve will often cause a knock by destroying the cushion, while a leaky steam valve will stop it by creating a cushion.

H. S. Brown.

New York City.

Firing with Gas

Referring to F. P. Wilson's inquiry in the September 27 issue under the title "Firing with Gas," I wish to say that in one of our plants where the boilers are gas fired, the only change made in the furnaces was that of covering the grates with cinders and placing a single row of firebrick midway between the front of the furnace and the bridgewall. The bricks were raised at the ends toward the bridgewall so as to deflect the fire from the burners up toward the shell and over the wall. The furnace fronts were drilled and tapped for 2-inch pipes at points below the fire doors, and pieces of 2-inch pipe, 10 inches long, threaded at one end, screwed in from the inside and pointed toward the center of the firebricks.

The burners were made from two pieces of 2-inch pipe capped at one end and connected to the supply line by a 2-inch tee. These pieces were each drilled and tapped for ¼-inch pipe and ¼-inch nozzles were screwed into them. The burners, when connected up, were placed so that the ¼-inch nozzles just entered the end of the 2-inch pipe in the boiler front. With this arrangement, the gas drew in all of the air needed for complete combustion. The outside gas connections were so arranged that the burners could be turned to one side in case of the failure of the gas and the boilers could be fired with coal without disconnecting any of the burners.

There are mica-capped peek holes in the front and back of the settings so that the condition of the fires can be seen at any time.

The ash and fire doors are kept closed as the burners force in all the air needed with the gas.

J. P. Colton.

Ohio City, O.

E. P. Wilson's inquiry relative to the above subject in the September 27 issue of Power made me think that the method used at the plant with which I have been connected for the last three years might be of interest to him.

The accompanying figure illustrates the manner in which the burners are made and connected up. All of the apparatus was made right in the plant. It consists of ordinary pipe fitting. The nozzles are made from ½x3-inch nipples drawn down at one end to about a ¼-inch opening; the other end screws into a ½-inch ell for the upper row of nozzles and a ½-inch tee for the lower row. The mouths of the nozzles extend into the 3½-inch pipes, or air mixer, about 1½ inches. The air mixers are made from old 3½-inch boiler flues sawed into 12-inch lengths and bricked into the furnace-door openings, after the doors had been removed. Six burners equally spaced are placed in each opening, each furnace having three fire doors.

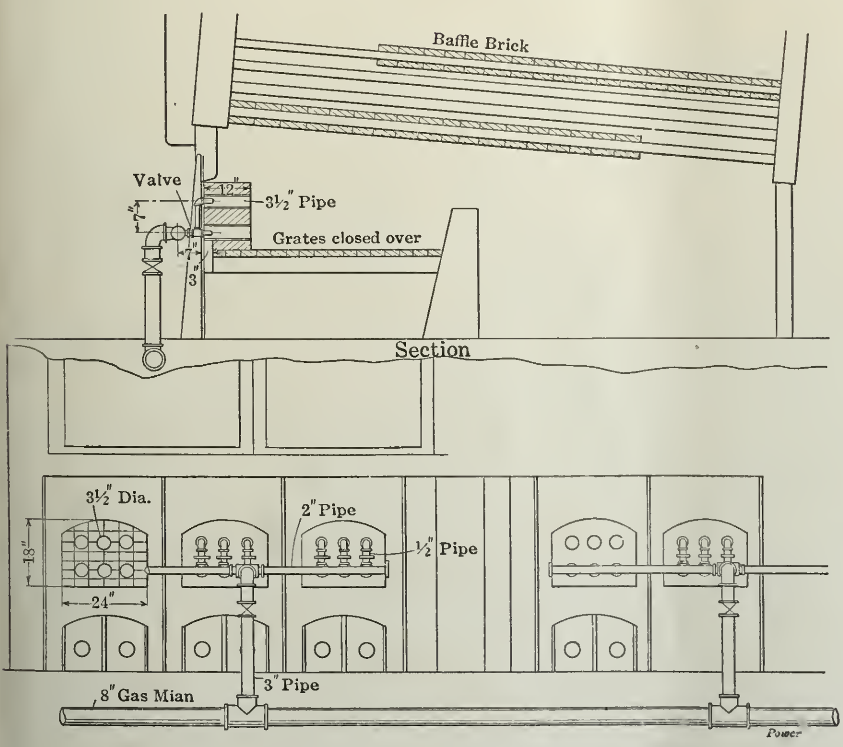
The arrangement and number of burners will, of course, to some extent, depend upon the size and make of boilers to which they are applied. In this case it happens to be 350-horsepower Heine water-tube boilers, arranged two in each battery and two batteries in each division, of which there are two, making a total of eight boilers. Each division is connected to an 80x6-foot diameter concrete stack.

An 8-inch gas main passes in front of each division. At the middle of each furnace a 3-inch pipe rises to the fire door. As illustrated in the sketch, the 3-inch pipe is reduced to 2 inches, branching in opposite directions parallel to the boiler front. Opposite each furnace door three ½-inch pipe taps are made and the burners or nozzles are made up with a ½-inch valve between the 2-inch pipe and the nozzles.

The furnace grates are covered with a course of brick to prevent cold-air currents from entering from below. No checker work has been employed. An ordinary bridgewall as used in the Heine boiler is the only obstruction to the gases except the standard baffle brick on the lower and upper bank of tubes. The gas pressure varies from 4 to 5 pounds and the fire is regulated by the 3-inch valve in the riser from the main in front of each furnace. The ½-inch valves near the nozzles may also be used for regulating the gas supply. But they are generally used merely when it is necessary to change or clean a nozzle while the furnace is in commission, which seldom happens.

The operation of this system has proved very satisfactory. The boilers have often been overloaded for a day at a time, but there has been no difficulty in keeping up steam and I have yet to see smoke issuing from the boiler stacks.

The important feature in connection with the successful burning of gas consists in having a proper "mix" or air supply at the proper time and place in the furnace. It is, therefore, important that the mouth of the nozzle be placed at a proper distance from the inlet of the air mixer. This distance will depend upon the size of both nozzle and mixer as well as the gas pressure, although the latter will tolerate a comparatively large range without affecting the mix to any great extent. When banking fires, a piece of sheet iron, cut round and a little greater in diameter than that of the mixer, with a slot large enough to fit ver the nozzle, is placed over the opening of each mixer and the gas turned own to a suitable degree.



General Arrangement of Furnace and Burners

It is not easy to say what causes the ifficulties Mr. Wilson states, as he does ot go into the details of the conditions nder which he has been burning gas nder his boilers. But it is likely that the rouble is due to either improper "mix," lack of draft or admitting cold air into the furnace at other places than where the gas enters and ignites.

T. G. Thompson.

Chanute, Kan.

Leakage through a Piston Valve

A careful reading of the article under the above title in the October 11 issue shows that its author spent a great deal of labor to *prove* that he was working with a leaky valve. He then fell into the fatal error of making a sweeping generalization from a single case. He finds this piston valve leaky; therefore, all piston valves are leaky.

And this is his process: In a small engine, he measures up the bore of the valve chest and the diameter of the valve and orders from the engine maker a special valve for experimental purposes. He may get a good-fitting valve or a bad-fitting valve. Upon test, he finds it leaks badly; therefore, he concludes, all piston valves leak badly. He expresses his conclusions in the following language:

"A significant fact, although one often lost sight of, is the enormous proportions assumed by the leakage past the piston valve of a steam engine; it reaching, in small engines, as high as 22 per cent. of the total steam consumption. When one stops to consider that this represents energy lost, its importance will at once be comprehended. The method herein described was devised by the author to measure this leakage under actual running conditions."

As to his "method" of measuring the leakage, he does not do it by direct measurement. On the contrary, a series of experiments is carried out, from which he undertakes to deduce certain laws governing leakage, and then, by a number of more or less elaborate calculations based on various assumptions, he attempts to arrive at the "actual" leakage.

His plan was to use a special valve which controlled the steam in the usual manner for the head end only, while the crank end of the valve was merely a blind plug traveling in the valve bushing. The engine operated as a single-acting engine, and the crank end of the cylinder was connected to a condenser to draw off and measure whatever steam leaked past the piston and the plugged crank end of the special valve.

He describes his calculations as follows:

"In making the calculations the assumption was made that the leakage was proportional to the pressure.

"The total leakage per hour through the various lengths of the crank end of the special valve, together with that of the piston, was reduced to equivalent leakage at 100 pounds pressure, and then the total leakage per hour at 100 pounds pressure was determined for the piston. The piston leakage being subtracted from the total leakage for each of the various lengths of the special valve through which leakage occurred, gave the net leakage through each of these various lengths, per hour at 100 pounds pressure. These results are shown by the curve, Fig. 5. Next, the equivalent distance constantly leaked through when the crank end of the special valve was at each of the various lengths was calculated, the results being shown by the curve in Fig. 6.

"The actual leakage per hour at 100 pounds pressure through the standard valve was now calculated. With the engine running normally, leakage occurred as follows:

"1. Between the steam chest and the exhaust passage, through the spaces between the steam ports.

"2. Through the spaces equal in width to the steam ports between admission and cutoff.

"3. Between the cylinder and the exhaust passage, through spaces equal in width to the steam ports between cutoff and release and between compression and admission.

"In order to determine the amount of these leaks, it was necessary to know the equivalent distances constantly leaked through and the pressure at which the leakage took place. The valve travel at each of the various horsepowers was determined, and knowing the valve travel, the equivalent distance constantly leaked through in each case, for each of the various horsepowers, was calculated.

"Leakage 1 occurred at a pressure

equal to that of the steam chest, as shown by the steam gage. Leakage 2 was due to this same pressure. Leakage 3, between cutoff and release, occurred at a pressure due to the average mean effective pressure of the expansion curves of the indicator diagrams for that particular run, and between compression and admission, occurred at a pressure due to the average mean effective pressures of the compression curves of the indicator diagrams for that particular run. Each of the above leaks was determined per hour at 100 pounds pressure by use of an equivalent distance curve.

"This same method was used for each of the various horsepowers from 5 to 15, and the results were plotted as shown by the curve, Fig. 6.

"The ratio of the total leakage through the standard valve to the total steam consumption at the various horsepowers was next determined, the results being shown by the curve in Fig. 7."

Thus it is seen how far he has strayed afield in the realm of supposition, hypothesis and imagination, away from ascertained or proved facts. Now all such roundabout and laborious "methods" are quite unnecessary. If a piston valve, or any other kind of valve, is leaking excessively, it is immediately and directly shown in the steam consumption of the engine. And that this particular engine was wasteful would be shown directly by the economy test which the author first made but does not report in his article. He says, however, that when the engine was developing 15 brake horsepower, the leakage of steam through the valve was 200 pounds (per hour, presumably), and that this leakage amounts to 22 per cent. of the total steam used. The total steam used would, therefore, be 200 pounds per hour divided by 0.22, which is about 910 pounds per hour, or about 60 pounds per brake horsepower per hour.

None of his elaborate calculations is needed to prove that this engine did indeed have a leaky valve. But what has this to do with other piston-valve engines which are commonly showing a steam consumption of about half what he found?

He calculates that the valve leakage of this engine was 22 per cent., while the total steam consumption was about 60 pounds per brake horsepower per hour. Now, if this valve leakage of 22 per cent. was stopped entirely, the steam consumption would then be 78 per cent. of 60 pounds, or about 47 pounds per brake horsepower per hour. What, then, would be the valve leakage of piston-valve engines which commonly show an economy of 35 pounds per brake horsepower per hour?

About three years ago, we made a careful investigation of piston-valve leakage, by running a series of economy tests on a single engine of average size, using in succession twenty different valves. All conditions were kept unchanged, except that a different valve was used for each test. Fairly high-pressure steam (150 pounds at the throttle) was used to develop the leakage to the full extent. Fifteen valves were made to have the ordinary commercial fit, and by direct measurement the steam consumption at full load was found to range from 26 pounds, the minimum, to 27 pounds, the maximum. per indicated horsepower per hour.

Then, a series of tests was made with valves intentionally made loose and of bad fits, to develop the loss thereby resulting. A valve two to three thousandths of an inch under size is plainly a poor fit. When moving it by hand it feels loose—any engineer would observe it. Three such valves raised the steam rate to 27.05, 27.4 and 27.8 pounds, respectively.

Two valves were then made ten thousandths of an inch under size. This is a fit so bad that no engineer would tolerate it—the valve could be rattled by hand in the valve chest. With these valves the engine tested out a steam rate of 32.1 and 32.7 pounds, respectively. This remarkable showing is explained by the long surfaces of the valve and the distance the steam had to travel through the fit in order to leak.

A. L. IDE & SONS.

Springfield, Ill.

License Laws

I am an engineer because I like engineering better than any other class of work and because, like any other human being, I must live. As there must be engineers as long as there are engines, I naturally picked out the trade I liked best, which I believe is one of the main essentials to success.

I believe in a strict license law which will allow only competent men to have charge of power plants; also, in a strict boiler-inspection law, which is of great importance to the engineer as well as the owner. Furthermore, I think that it is well worth while for the engineers to demonstrate to the owner that he is worth a better salary if the one he is getting is too small. After proving his ability an engineer should not be afraid to ask for an increase. If the owner will not pay what the work is worth, look for a man who will. However, do not leave one job until you see a chance to do better; it is of no use to cut off the nose to spite the face.

I am not posing as a "public benefactor" in demanding better laws, as a writer in a recent number of POWER called progressive engineers. It has been said by some engineers that license laws are merely an attempt to limit the number of engineers for the purpose of keeping wages up by creating a demand in excess of the supply. Assuming that this is true, it is no more than is being done in any number of other classes of work. If you go into a machine shop you will find that the old machinists have a standing grudge against the apprentice. The same may be said of all the other trades and professions, and the engineering business in all its branches is of no small magnitude.

Some writers state that the engineer receives less wages than men employed in other trades and also that the wages in license-law States are no higher than in States where there are no license laws. I suggest that POWER collect data on the wages, hours and general conditions of engineers in different States and on the size of plants, etc., and print the results in a tabular form similar to that used some time ago in regard to cylinder lubrication.

In this way some comparison could be made as to the benefit of license laws. It seems to me that some such data could be collected which would give us some definite information concerning the conditions of the engineer.

J. CASE.

Hyattsville, Md.

G. W. Johnson asks in the October 11 issue: "In a no-license State, how shall the employer decide upon the merit of an applicant who is unknown to him unless he judge by letters of recommendation which too often are worthless?"

Again, he says:

"Take the employer who believes, on principle, in having everything of the best—equipment, supplies and operatives, and is willing to pay the price. . . . Please state, you nonlicense men, how you, in his place, would judge the fitness of applicants?"

It is the business of the Government to *protect* the people in their peaceful pursuits, their homes, etc. But the Government exceeds its power when it undertakes to guarantee to employers efficient workmen. It is within its rights when it prevents the employment of a person in a position with which that person is unfamiliar, thereby becoming a menace to the public. The Government has not the power to weed out the inefficient; it has the power to weed out the menaces.

Again, I quote:

"Recently one or two engineers have expressed the opinion that license laws would have but little beneficial influence. . . . I should like to call their attention to the fact that with the enforcement of a proper law, an immense number of operating misfits would at once be obliged to leave the field clear for those who by virtue of a superior intelligence and ambition already possess the necessary qualifications."

I cite this passage in support of my statement in the September 27 issue, that the motive of the engineers behind this license-law movement is to make scarce the supply of engineers that the scarcity may fructify to those remaining.

The reason why license laws will not accomplish the weeding out of the inefficient is because it is against the law of evolution. According to this law, man collectively will improve at a given rate; this rate is regulated by two factors which are equally important, conditions and the man. Conditions present the necessity and man, individually, will or will not rise to the occasion. If he lacks the temperament, the ambition, the energy, the intelligence, the physique or any of the requirements that go to make up the human factor he cannot meet the requirements. If the requirements call for a standard which cannot be met by a sufficient number of capables, then the standard must yield and the necessity will be met by that degree of efficiency held by those who cannot attain the standard.

If there are a sufficient number of men whose knowledge and experience qualify them to *safely* operate a steam plant so that it is not a menace to the public, a license law would have the effect of putting to work those who are found to be *safe* as long as they are to be had, to the exclusion of the unsafe.

Attention is called to the emphasis on *safe* because any license law or examining board that ventures beyond the point where the public is interested and proceeds to examine into a man's *ability* or *efficiency* is working in the interest of the employer and not in the interest of the State for which it was created and is, therefore, illegal.

It is my opinion that every engineers' examining board in the country is transgressing the law in the interest of the employer and to the hardship of the engineer.

J. J. Nigh.

New Haven, Conn.

Some engineers seem to feel that it is somewhat of a hardship to have to go to the expense of paying a license fee. This may be true, but the examination will be worth the expense as a review. Many a member of an examining board could not answer nearly as many questions as some engineers. On the other hand, some very able engineers could not stand an examination, although they can do things and do them in the best possible manner, and are safe to intrust with the care of a steam plant. And, again, there are exceptions. I once met a man who was an engineer, had his license, had been examined by one of the boards, yet with all of this the man could not correct a faulty valve gear. This was not the fault of the license-law system.

The license law will cut out, to a certain extent, the opportunity for a man to say that he is an engineer when in reality he is not. You would not like to sit in a passenger coach behind an engineer that you knew nothing about unless you knew that the railroad had a man in the cab that had stood a thorough test. The railroads go so far as to have special cars in which to conduct examinations and secure practical illustrations of a man's ability. Why could not the stationary engineer be tested under similar conditions?

The United States Government has found that in order to have men to fill the places safely it is necessary to have examinations. They do not take the man on what he says but on what he can show by actual demonstration. Steam machinery is becoming more complicated each day and higher pressures are coming into use. Hence, plants should not be intrusted to those who we do not know are all right. In times gone by we may have gotten on fairly well without license laws but modern conditions necessitate suitable regulating laws. I see no reason why the engineer should complain. It will certainly better his condition. In some cases he may have to read up some but there is plenty of time for this, and those who are prepared will not object to being examined.

C. R. McGahey.

Lynchburg, Va.

Installing Globe Valves

The globe valve that controls the steam to the boiler feed pump is frequently adjusted by the attendant and is never wide open and seldom closed. For this reason it will always work better with the pressure under the disk, for with the lower pressure on the packing the valve can be closed and packed at will, and if the disk comes off the stem the pump is not thrown out of service.

A globe valve should never be put in the feed line with the pressure from the boiler under the disk as the pressure from the pump will then be on the top and if the disk should come off the stem no water can be put into that boiler until the valve has been repaired. Nor can the valve be packed while the feed pump is running.

The blowoff valve on the water column should have the pressure from the boiler on the top of the disk for the reason that it is often used by inexperienced firemen and left loose enough to leak and so destroy the seat.

A globe valve should never be used on the steam line between the boiler and the main header. If the valve is put on with the pressure from the line on the top of the disk it cannot be packed while steam is on the line. This would be bad, for steam may have to be kept on the line for months at a time. On the other hand, to install the valve with the pressure from the line under the disk would be even worse, for no sane engineer or boiler inspector would enter the boiler with high-pressure steam on the line, because often the valve yoke breaks and the stem threads strip, in which event it would not be hard to guess what would happen to the man on the inside.

When I took charge here the angle blowoff valves were attached with the pressure under the disks, a Homestead valve being between each angle-valve and boiler. When the angle valve leaked, which happened about every three days, the Homestead was closed, a new lining put in the iron disk of the angle valve and the seat turned off. When I installed new blowoff piping I took out the Homesteads and changed the angle valves so that the pressure came on the top of the disks, with the result that the valves now give very little trouble.

James Johnson.

Hackett, Penn.

Boiler Feeding by Centrifugal Pumps

The editorial in the issue of October 11 under the above heading is worthy of discussion. There are many points in favor of the use of the centrifugal pump as a boiler feeder, both motor and turbine driven. I think that in either case the speed should be variable in order to meet sudden demands. In a plant with a steady load the feed water can be fed at a constant rate, but in street-railway or hydraulic-elevator plants the water must be supplied in accordance with the demand for power, the demand varying with the number of cars running and the load they carry. Another important point in favor of the centrifugal pump is found when an open heater is used, and the exhaust steam from the pump is used to heat the water in the heater. Reciprocating pumps usually require considerable cylinder oil which is exhausted into the heater and is constantly getting into the boilers through poorly working oil separators. The centrifugal pump should be greatly appreciated for this one reason.

The first cost, however, is slightly against them, but this may possibly be reduced as the demand increases.

I have seen many centrifugal pumps used in hot-water heating systems to good advantage under low pressures and handling water at 190 degrees Fahrenheit and above. To equip the centrifugal pump with a turbine engine makes it independent as a steam pump and requires much less care in operation.

Pressures are now increasing very rapidly and most new installations consist of boilers with a working pressure of from 175 to 200 pounds. When centrifugal pumps are produced that will deliver water of from 200 to 210 degrees against these pressures at a variable speed without the possibility of the water hammering or the pump becoming steam-bound, the demand will increase rapidly.

R. A. Cultra.

Cambridge, Mass.

Comparison of Lubricating Greases

By Horace W. Gillett

> *In a series of comparative tests upon corresponding samples of mineral and tallow greases, in which both Olsen and Thurston testing machines were used with varying bearing pressures, the tallow greases showed a lower melting point and a lower coefficient of friction.*

Most engineers and users of lubricants agree that there are certain bearings on which, for the sake of convenience, cleanliness or economy, it is advisable to use a lubricating grease. But having gone that far, they consider that "grease is grease," and that one kind is as good as another.

Commercial greases differ in chemical composition vastly more than do lubricating oils, and as it was almost certain that their lubricating properties differed also, it seemed worth while to the writer to make a series of actual friction tests of the common greases on different oil-testing machines, particularly as the literature of lubricants show such a woeful lack of impartial investigations.

Before taking up the actual results of the work, it is advisable to consider the types of greases now in use, in order to see what a knowledge of their composition and properties would lead one to expect of them as efficient lubricants. Gear greases, graphite and mica greases will be left out and the discussion confined to those greases commercially used and widely advertised for service where a high-grade and efficient lubricant is demanded.

TABLE 1. RESULTS WITH THURSTON MACHINE.

Type.	Melting Point, Degrees Fahrenheit	Percentage of Soap.	Kind of Soap	Percentage of Free Acid as Oleic.	Average Coefficient of Friction.	Final Coefficient of Friction after 3-hour Run	Maximum Temperature of Bearing above that of room, Degrees Fahrenheit.	Final Temperature of Bearing above that of room at end of 3-hour run, Degrees Fahrenheit.
A, mineral, summer	167	38	Lime	Trace	0.075	0.75	70	68
B, mineral, summer	178	20	Lime	0.3	0.054	0.050	70	58
C, mineral, winter	165	23	Lime	6.1	0.063	0.063	76	65
D, mineral, winter	163	16	Lime	0.0	0.057	0.054	69	58
E, mineral, winter	142	19	Lime	Trace	0.046	0.046	58	50
F, tallow No. 3, winter	125	1.4	Potash	0.0	0.022	0.012	38	18
G, tallow No. XX, summer	120	2.1	Potash	0.0	0.029	0.018	45	32
H, lard oil	41	0 0	......	......	0.011	0.010	13	12

TABLE 2. RESULTS WITH OLSEN MACHINE.

	500 Lb. Total Pressure on Bearing = 86.5 Lb. per Sq.In.			1000 Lb. Total Pressure on Bearing = 173 Lb. per Sq.In.			1500 Lb. Total Pressure on Bearing = 253 Lb. per Sq.In.			2000 Lb. Total Pressure on Bearing = 345 Lb. per Sq.In.			2500 Lb. Total Pressure on Bearing = 431 Lb. per Sq.In.			Average Final Coefficient of Friction.	
	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	Max. Temp. of Bearing above Room, Degrees Fahrenheit.	Average Coefficient of Friction.	Final Coefficient of Friction.	All Pressures.	All Pressures and Grades.
SOFT GREASES																	
Mineral, X	28	0.0075	0.0062	46	0.0111	0.0085	68	0.0120	0.0103	86	0.0138	0.0125	83	0.0196	0.0152	0.0106	.....
Tallow, Albany No. 1	22	0.0075	0.0050	31	0.0084	0.0075	44	0.0113	0.0093	67	0.0135	0.0123	75	0.0144	0.0117	0.0092	.....
MEDIUM GREASES																	
Mineral, Y,	24	0.0126	0.0080	39	0.0114	0.0095	50	0.0123	0.0107	65	0.0140	0.0115	75	0.0140	0.0120	0.0102	.....
Tallow, Albany No. 3	23	0.0088	0.0056	36	0.0100	0.0080	50	0.0112	0.0103	64	0.0140	0.0115	75	0.0140	0.0116	0.0094	.....
HARD GREASES																	
Mineral, Z	28	0.0113	0.0090	55	0.0132	0.0108	72	0.0136	0.0113	76	0.0141	0.0117	87	0.0132	0.0116	0.0109	.....
Tallow, Albany XXX.	26	0.0090	0.0067	45	0.0131	0.0100	19	0.0124	0.0100	67	0.0127	0.0100	69	0.0121	0.0100	0.0093	.....
Mineral	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.0106
Tallow	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0 0093

There are only two types of greases that command the serious attention of the engineer, the mill superintendent or the automobile owner; these are the so called mineral greases and the tallow greases. The mineral greases consist of mineral oils, thickened with a percentage of lime soap, varying according to the consistency desired. These have a high melting point, usually from 150 to 200 degrees Fahrenheit. The tallow type of greases consists mainly of some hard animal or vegetable fat such as tallow, with a small amount of mineral oil and a small amount of soap, usually a potash soap. As a rule, they melt at from 75 to 125 degrees Fahrenheit, although at room temperature they are much harder than the corresponding mineral greases.

Any grease would be a hindrance to the bearing rather than a help, if, when in action, it were in the same condition in which it exists in the grease cup. In its comparatively hard and sticky consistency it offers resistance to the moving parts; hence before it begins to lubricate, it must melt. In other words, the grease should become an oil when it comes in contact with the bearing. There is no doubt that the actual contact surfaces of a grease-lubricated bearing are at as high a temperature as the melting point of the grease, although the whole bearing and journal seldom reach that temperature, because of radiation to the air and conduction to other parts of the machine.

When starting, the shaft generates heat by friction until the grease melts and becomes a true lubricant, at which point it begins to cut down the friction. To keep this initial friction and consequent power loss small, one should use a grease of as low a melting point as is compatible with having the grease stay in the cup without waste. This judgment is used by the engineer who uses a summer and a winter grade of grease, colder weather allowing him to use a grease of a lower melting point, or softer consistency. But the melting point is not the only factor; the friction-reducing power of the oil that the melted grease forms must be considered.

To find out which is actually the best type of commercial grease, two series of friction tests were made, one on a small Thurston testing machine and the other on a large Olsen machine. The series of tests made upon the Thurston machine will be taken up first. This machine was run throughout the series of tests at 320 revolutions per minute and with a 60-pound pressure on the bearing, which is equivalent to 240 pounds per square inch upon the projected area. The bearing was thoroughly cleaned, smeared with the grease to be tested, the cup cleaned and filled, and the machine then run for three hours, the coefficient of friction and the temperature of the bearing being read every five minutes.

Seven kinds of commercial greases were thus tested. These consisted of five samples of the mineral type; three

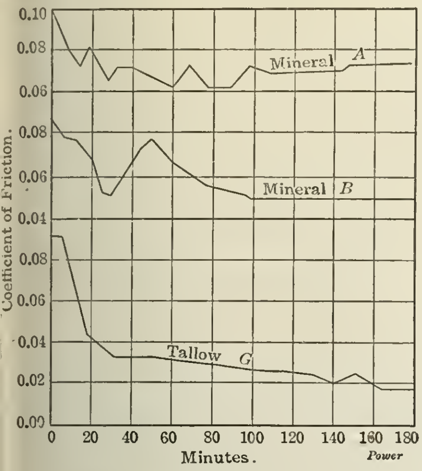


FIG. 1. SUMMER GREASES

f the winter grade and two of the summer grade, made by three different epresentative firms; and two samples of allow grease—Albany No. 3 and No. X X. Besides these greases a pure lard oil vas used as a standard of reference, so hat tests made on other machines or by ther observers might be compared with ur results.

In order to compare summer and winter reases with others of their own class, ey were divided into two sets, those alled *A*, *B* and *C* representing the summer grade, and *C*, *D*, *E*, *F*, the winter rade. The lard oil is designated by *H*. is perhaps straining a point to put lbany No. 3 or *F* in the winter class, as is quite a stiff grease and several ofter grades are made, but for this eries of tests none of those softer grades ere available.

The results of this series of tests are iven in Table 1.

The higher melting point and higher ap content of the mineral type are shown the table and it is also of interest to ote that although the tallow greases igh be expected to become more or less ncid, as they are made largely from atural fats, this is not the case, the ineral greases being the ones that show ee acid from the fats used in making e soap content. If palm oil is used in aking the stiffening soap, as is often e case because of its low price, any that oil which is not neutralized will oner or later give rise to free acid. lthough the 6 per cent. of acid found in e of these mineral greases is higher than that type will usually run, a trace of free acid is very often present in mineral greases. None of the tallow greases that the writer has examined have shown any free acid.

The coefficient of friction curves plotted against time is given in Figs. 1, 2 and 3, the first representing the summer greases, the second the winter greases and the third the lard oil.

Without a single exception the tallow greases, with their low melting point and their low soap content, gave better results than the other types—results not much inferior to lard oil itself. To check the conclusions derived from the tests made with the Thurston machine and to study the effect of varying the pressure, a series of tests was made on an Olsen oil- and grease-testing machine belonging to Sibley College of Mechanical Engineering, Cornell University.

The pressure used on the Thurston machine was 240 pounds per square inch, while that on the Olsen could be raised to 431 pounds per square inch, or 2500 pounds actual pressure on the test bearing, which had an area of 5.8 square inches. The diameter of the journal on the Olsen machine was 3¾ inches and its length 3½ inches. The journals of both machines were of steel with the bearing of the Thurston of brass, and that of the Olsen of babbitt. The latter was run at from 215 to 240 revolutions per minute, which was equivalent to a speed of from 212 to 235 feet per minute. Runs were made at various pressures, as shown in Table 2, and the speed, friction and temperatures were read every ten minutes. Each run lasted one hour, the temperature and coefficient of friction in every case becoming constant before this period was over. The greases were fed at such a rate as would maintain in each case the lowest coefficient of friction possible with the grease used.

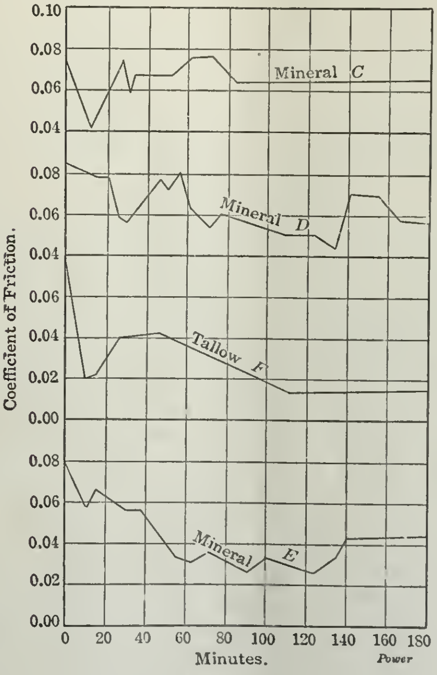


FIG. 2. WINTER GREASES

As there was not enough remaining of the samples used on the Thurston machine to make all the tests desired on the Olsen, the opportunity was seized to make a closer comparison between the tallow greases and the mineral greases of corresponding grades. To this end three grades of Albany grease, hard, medium and soft, were selected and matched by three samples of mineral greases, hard, medium and soft, which were recommended as suitable for the same classes of work for which these grades of Albany grease are recommended. Of course, the consistency of the tallow greases at room temperature was harder than that of the corresponding mineral grease, owing to their different composition. To compare with Albany No. 1, which is a soft grease, a mineral grease was selected which for convenience is designated in Table 2 as *X;* to compare with Albany No. 3, which is a medium grease, a mineral grease *Y* was chosen; and for Albany XXX, which is a hard grease, a mineral grease *Z* was selected.

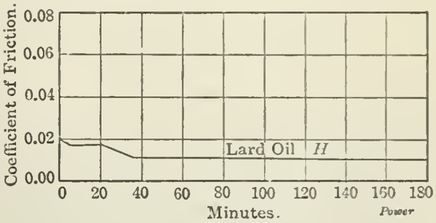


FIG. 3. LARD OIL

The results of the tests are given in Table 2, and in Figs. 4, 5 and 6 are shown graphically the final coefficients of friction, obtained at the end of the hour's run, for each pressure.

The mineral grease contained so much soap that the residue taken off the bearing after a run consisted of oil and hard particles of soap; that is, the oil had melted away from the soap, and the grease was no longer homogeneous. The grease parts with the oil first and what is finally left in the cup is practically nothing but soap. This will not show up in an hour's test, of course, but where the grease is used long enough for the oil to be exhausted, as is the case in practice, this is a vital point.

In the table are given both the average coefficient of friction during the whole run and the final coefficient. The latter is the more important, as it shows a condition in which the bearing will run all day long. But if a grease that gave a low final coefficient of friction gave a very high one on starting, then the average coefficient of friction would give valuable information as to the value of the particular grease on a bearing that was to be run only intermittently. Whether the average or final values of the coefficient of friction are considered or whether the temperature of the bearing is taken, at all pressures the tallow greases showed a distinctly greater friction-reducing power than did the greases of the mineral-oil type.

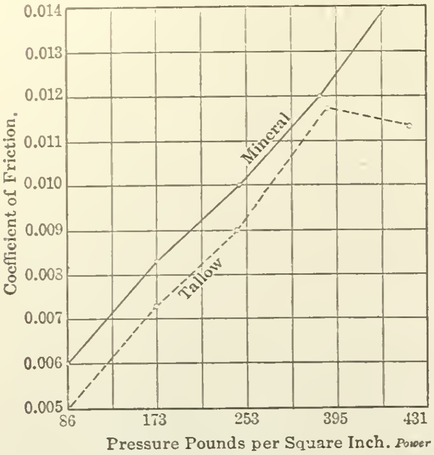


Fig. 4. Soft Greases

It will also be noted that the stiffer greases gave better results at high pressures than the softer grades, Albany XXX, the hardest grade of tallow grease, giving equally as good results at 431 pounds per square inch as at 173 pounds, while the softer greases showed an increase in friction with an increase of pressure. Hence, to get the best results not only must the type of grease be determined but also the grade best adapted to the particular problem under consideration.

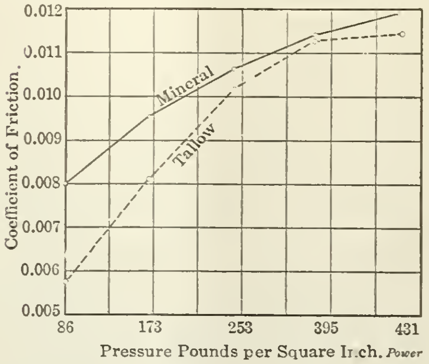


Fig. 5. Medium Greases

The difference between the tallow and mineral types, or between the hard and soft mineral greases, does not show up as great on the Olsen machine as on the Thurston, even at practically the same pressures; the value obtained for the coefficient of friction on the former is less than that obtained on the latter. This throws considerable light on the problem, and upon examining the two bearings the reason is found. The journal and bearing on the Olsen machine, which were new and well kept, were as smooth as it is practicable to make a bearing, while the bearing of the Thurston machine was not very smooth, as it had not been used for several years before this series of tests. It was ground down as smooth as possible, but was still rather rough. In other words, on the Thurston machine the grease had to do more lubricating, and in such a case the better lubricants would show an increased efficiency, as they have a wider field to work in. This shows not only the superior ability of the tallow grease to lubricate under trying conditions, but also that in making tests to show up differences in the lubricating power of oils or greases, the test bearing should not be in too good a condition. Even on the good bearing, the average decrease in friction in the whole fifteen sets of runs obtained solely by substituting a tallow grease for a mineral one is about 12 per cent.

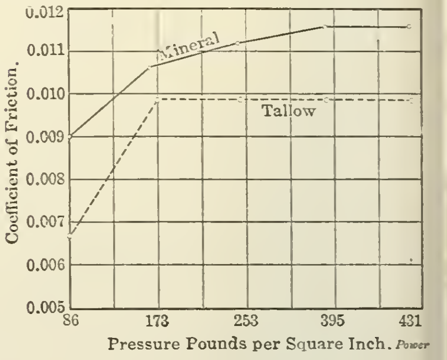


Fig. 6. Hard Greases

To sum up the results of the test it may be said that greases with a low melting point and with a low soap content should be the most efficient theoretically; the tallow greases have a lower melting point and a lower soap content than mineral greases; in actual tests on two oil-testing machines, the tallow greases consistently showed their superiority to the greases compounded from mineral oil and lime soaps, not only at one pressure on the bearing, but at all pressures.

Low Pressure Steam Turbines[*]

By H. G. Stott

> *In the Interborough plant the installation of low-pressure turbines more than doubled the economic capacity of the plant and resulted in the remarkable unit thermal efficiency of 22 per cent.*

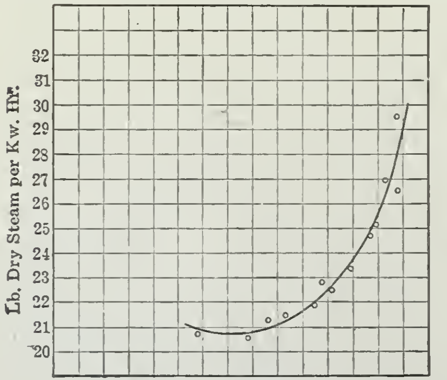


Fig. 1. Variation of Engine Water Rate with Load

Several installations of low-pressure turbines have been made during the last year with satisfactory results. The largest installation has been that of the Interborough Rapid Transit Company in New York, which has installed three 7500-kilowatt units and is now engaged in installing two more units of the same size.

Brief extracts from a very elaborate series of tests on these low-pressure units, which have already been published, follow, as they may be looked upon as typical of what may be expected in other installations under similar circumstances.

Each of the 7500-kilowatt low-pressure turbine units receives its steam from a Manhattan type of Corliss engine having two 42-inch horizontal high-pressure cylinders and two 86-inch vertical low-pressure cylinders. The high- and low-pressure pistons on each half of the unit connect onto the same crank, so that the unit consists of two compound engines with the generator between them running at 75 revolutions per minute and with a 60-inch stroke. The revolving field of the generator is built up from riveted steel plates of sufficient weight to act as a flywheel for the engines on each side of it.

The high-pressure valves are of the poppet type and the low-pressure valves of the Corliss type. These units were rated at 5000 kilowatts with 50 per cent. overload capacity, giving their most economical results between 4000 and 5000 kilowatts. As far as the heating limit was concerned it has never been reached as the engine becomes too uneconomical to justify operation before the maximum capacity of the generator is reached.

Fig. 1 gives the engine water rate when running noncondensing, and Fig. 2 gives the water rate for the low-pressure turbine, with the steam pressure entering it varying according to the load on the engine from one pound above the atmosphere to five pounds below, with a corresponding condenser vacuum of 28.5 inches when referred to a standard barometer of 29.92 inches.

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[*] Portion of the report of the Committee on Power Generation of the American Street and Interurban Railway Engineering Association, Atlantic City, October 10-14.

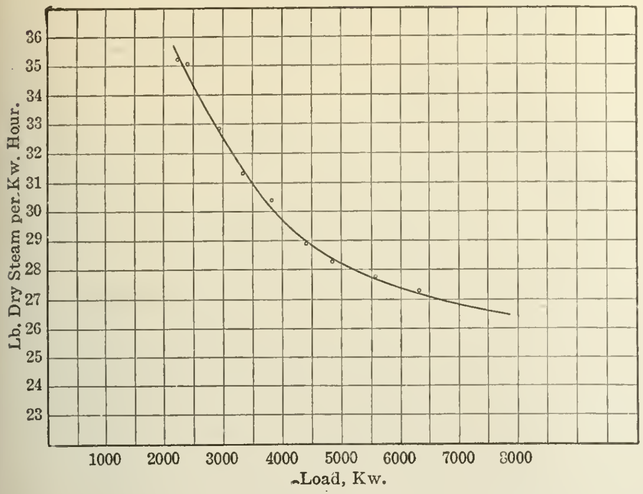

FIG. 2. VARIATION OF TURBINE WATER RATE WITH LOAD

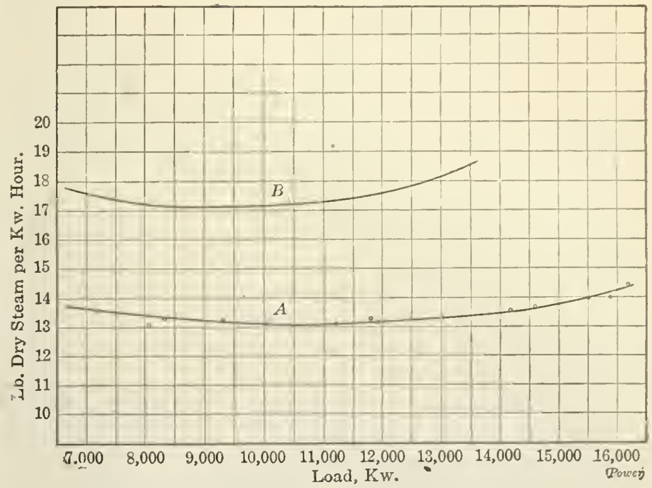

FIG. 3. VARIATION OF UNIT WATER RATE WITH LOAD

The opposite slopes of Figs. 1 and 2 result in the combined water rate shown in Fig. 3, giving an almost flat water rate for the combined unit between the wide limits of 6500 and 15,500 kilowatts.

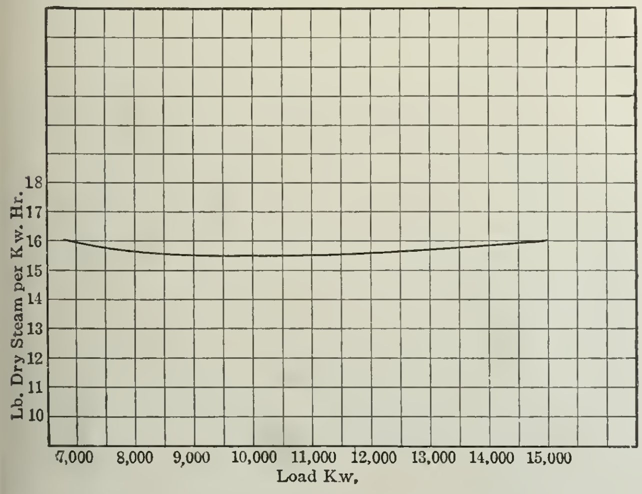

FIG. 4. VARIATION OF WATER RATE AND LOAD, HIGH-PRESSURE TURBINE

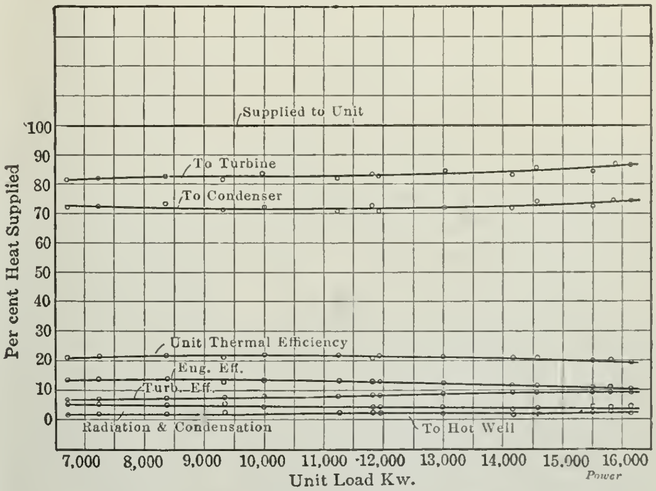

FIG. 5. VARIATION OF HEAT DISTRIBUTION WITH UNIT LOAD

To show the gain made by the installation of the low-pressure turbines the water rate of two engines is given at *B*, Fig. 3, for comparison, showing an average improvement in the combination unit, curve *A*, of over four pounds per kilowatt-hour.

Fig. 4 shows water rate guaranteed for a typical modern 12,000-kilowatt standard high-pressure turbine unit when running under the same steam pressure and vacuum as obtained for the combined unit, as shown in Fig. 3.

All these curves shown are results obtained with saturated steam only.

Fig. 5 shows the heat distribution between the various parts of the unit, showing the remarkable unit thermal efficiency of over 22 per cent.

The results obtained by the installation of low-pressure turbines are as follows:

(*a*) An increase of 100 per cent. in maximum capacity of plant.

(*b*) An increase of 146 per cent. in economic capacity of plant.

(*c*) A saving of approximately 85 per cent. of the condensed steam for return to the boilers.

(*d*) An average improvement in economy of 13 per cent. over the best high-pressure turbine results.

(*e*) An average improvement in economy of 25 per cent. (between the limits of 7000 and 15,000 kilowatts) over the results obtained by the engine units alone.

(*f*) An average unit thermal efficiency between the limits of 6500 and 15,500 kilowatts of 20.6 per cent.

One of the novel parts of this installation is the use of the induction type of generator on the low-pressure turbines. The reason for adopting this type of machine was that with the enormous short-circuit capacity of the system, amounting to over 300,000 kilowatts, the addition of large synchronous turbo-generators would result in a largely increased short-circuit capacity and it was feared that the switching apparatus would soon reach its limit of safe operation. The addition of induction generators, on the other hand, adds nothing to the short-circuit capacity and tends to help the regulation by adding so much stored energy or flywheel capacity. The results obtained have fully justified the installation and the simplicity of operation is most marked.

In general the installation of low-pressure turbines may be recommended wherever there are good engines already installed or in the case of a new installation where the load factor and the coal cost are high. In plants having a large installation of a good type of reciprocating engine, the low-pressure turbine may be added at a total cost, including new condenser, auxiliaries, foundations, piping, etc., of not to exceed $25 per kilowatt, thus bringing down the average overall investment per kilowatt of the entire plant and so reducing the fixed charges per kilowatt-hour.

Inquiries of General Interest

Split Blowoff Pipe

If the blowoff pipe on one of a battery of boilers should split would it be practicable to increase the speed of the feed pump to make up for the leakage?

S. B. P.

Probably not. The leakage from a short crack which could be prevented from extending to a dangerous length by suitable clamps, might be made up by an increase in pump speed. But in case of a serious rupture, the boiler should be cut from the line and the fire covered.

Steam Pressure in Boiler

In the issue of October 4, P. S. B. asks: Is the pressure equal in all parts of a steam boiler? A part of your reply is that the pressure of the steam is exerted equally in all directions on the water as well as on the shell.

I find in Roper's "Handbook of Land and Marine Engines" the following statement in the chapter relating to longitudinal and curvilinear strains: "The strain on a boiler subjected to internal pressure transversely is exactly double what it is longitudinally, or in other words, the strain on the longitudinal seams is double that on the curvilinear." I should like your explanation of this matter.

S. O. S.

The pressure of steam in a boiler is equal in all directions, but the stress on the curvilinear seams per inch of length is one-half that on the longitudinal ones.

The stress on each inch of the longitudinal seam is,

$$\text{Radius} \times \text{pressure} = \text{stress},$$

while that on each inch of the curvilinear seam is,

$$\frac{\text{Area of head} \times \text{pressure}}{3.1416 \times \text{diameter}} = \text{stress}$$

In a boiler 60 inches in diameter carrying 100 pounds pressure per square inch, the stress on each inch of the longitudinal seam is

$$30 \times 100 = 3000 \text{ pounds},$$

while on each inch of the curvilinear seam it is

$$\frac{2827.4 \times 100}{188.496} = 1500 \text{ pounds}$$

Area to be Stayed

What is the area to be stayed on a 60-inch boiler? Distance from tubes to shell 21 inches?

A. B. S.

The area of a segment of a boiler head to be braced is the area inclosed by lines drawn 3 inches from the shell and 2 inches from the tubes and is found approximately by the formula

$$Area = \frac{4\,H^2}{3}\sqrt{\frac{2\,R}{H} - 0.608}$$

in which,

H = Distance from tubes to shell less 5 inches;

R = Radius of boiler less 3 inches.

Substituting

$$\frac{4 \times (16)^2}{3}\sqrt{\frac{2 \times 27}{16} - 0.608} = 560.25$$

square inches of area to be stayed.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Trouble with Governor Safety Pin

I have charge of a Corliss engine upon which, as soon as the governor leaves the safety pin, it will drop and stop the engine if I do not hold it up.

This occurs only when starting up. Can you give me the cause of the trouble?

A. H. B.

You have slightly misstated the case. The governor does not leave the safety pin; that is, it does not rise. As the speed of the engine increases, the pressure of the governor rest on the safety pin is reduced, and as the pin is held in place by the friction between it and the governor rest, the force which moves the pin aside overcomes the reduced friction, and the pin is moved from under the rest before the governor has acquired the speed necessary to make it rise; therefore, it drops. The remedy is to make a shallow recess in the safety pin, out of which the governor rest must be lifted by rising from the effect of centrifugal force before the pin is moved aside.

Efficiency of Riveted Seams

What is the efficiency of joint? Thickness of plate 3/8 inch, 55,000 pounds tensile strength 13/16 inch diameter of rivets, pitch 3½ inches, joint single riveted?

E. R. S.

The efficiency of a riveted seam is

$$Efficiency = \frac{p - d}{p}$$

in which

p = Pitch of rivets in inches;

d = Diameter of rivet in inches;

hence,

$$\frac{3.5 - 0.8125}{3.5} = 0.767$$

or 76.7 per cent., the efficiency of the joint, and its strength is

$$0.375 \times 55{,}000 \times 0.767 = 15{,}819.37$$

pounds per inch of length.

Safe Working Boiler Pressure

What is the safe working pressure of a boiler 60 inches diameter, 55,000 pounds tensile strength, 3/8 inch thickness of plate, efficiency of seam, 70 per cent.; factor of safety, 5?

W. B. P.

The working pressure on boiler shells and drums is determined by the formula

$$\frac{T.S. \times t \times \%}{R \times F.S.} = \text{working pressure}$$

in which

$T.S.$ = Tensile strength of plate in pounds per square inch;

t = Thickness of plate in inches;

$\%$ = Efficiency of seam;

R = Radius of shell;

$F.S.$ = Factor of safety.

Substituting

$$\frac{55{,}000 \times 0.375 \times 0.70}{30 \times 5} = 96.25$$

pounds per square inch working pressure.

Safe Flywheel Speed

I wish to run an engine at 200 revolutions per minute; what will be the diameter of the largest cast-iron flywheel I can use with safety?

S. F. S.

The rim speed of a cast-iron wheel should not exceed 6000 feet per minute. The diameter of a wheel for this speed may be found by dividing 1910 by the number of revolutions.

$$\frac{1910}{200} = 9.55 \text{ feet}$$

the safe diameter of a flywheel for 200 revolutions per minute.

Calculating Pipe Sizes

Is there any simple rule for calculating the size of a pipe which shall have an area equal to that of two other pipes of different sizes?

C. P. S.

Lay off on one side of a square a distance equal to the diameter of one of the pipes and on the adjacent side a distance equal to the diameter of the other. The diagonal distance across the corner of the square from one point to the other is the diameter of a pipe the area of which will equal that of both the others.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for October, 1910, *was* 132,000.

November 1............ 33,000
November 8............ 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

Unification of the Steam Turbine

The steam turbine seems to be growing toward a general type built by all the large makers. We found that practically all the European builders of the Parsons type, except Parsons himself, and several builders of the Rateau type are using a velocity stage, i.e., the Curtis system, for the initial expansion. In the Parsons turbine a large part of the barrel is made up of short blades of practically the same length to effect the same expansion which takes place in the first stage of a Curtis. By allowing this expansion to take place in a single set of nozzles and using a Curtis set of two rows of running and a single row of reversing buckets to absorb the velocity so generated, the steam is reduced in pressure from, say, one hundred and fifty to thirty pounds before it is introduced to the turbine casing, decreasing the pressure upon the balancing plates or dummies, diminishing the temperature to which the rotor drum and shell are subjected, and eliminating the long section of short blades, admittedly the least efficient of the turbine on account of their large windage in the dense medium of the high-pressure steam and of the large proportion of their clearance to the active surface. The doing away with this section also greatly diminishes the length of the turbine between bearings and gives a stiffer and more manageable shaft or drum, which allows of higher rotative speeds. In the Rateau type the large initial fall of pressure brings so much less pressure upon the stuffing box at the high-pressure end. Professor Rateau himself, however, does not approve of the use of the Curtis member here, preferring to make the same drop, if desirable, in a divergent nozzle before the steam comes into the casing and against the stuffing box, but absorbing as much of the consequent velocity as possible with a single row of blades, preserving the residual velocity for use upon the second wheel by the entrance curve of the second row of nozzles, and augmenting that velocity by expansion in that second row until the work performed by that row is substantially the same as that in the first. Even he, however, uses the velocity-stage construction where lightness and first cost are the main considerations. On the other hand, the British Thomson-Houston Company and the Allgemeine Elektricitäts Gesellschaft, the builders of the Curtis type in Great Britain and Germany respectively, have used the Curtis reversing blades and second row of runners only in the earlier stages of many of their turbines, using single runners in the lower pressure stages. In our own country the Westinghouse Machine Company uses the double-runner velocity stage for the initial expansion, not of all but of some of their turbines, while we understand that the General Electric Company, the builders of the Curtis type, have built some units in which the Parsons system is used for the final stages.

The systems appear therefore to be growing together by the adoption of the other builders of Curtis initial stages and the renouncing of the Curtis builders of the Curtis principle for the final stages.

Professor Beluzzo has proposed and Gadda Finzi & Co., of Milan, have built a unit in which a Curtis velocity stage is used for the initial expansion, Rateau pressure stages for the middle range and Parsons full admission reaction stages for the final expansion.

Hotel Power Costs

For the past several months the *Hotel World* has been carrying on a controversy among its readers as to whether it is policy for a hotel to operate its private plant or to buy current from a central station.

From the several score of letters on the subject it is evident that a great majority of managers have no definite information on the subject. Their decisions are based largely on precedent and frequently have no better foundation than "We operate a private plant because we found it in the basement when we bought the hotel," or "We buy central-station current because we always have." Apparently there has been little investigation of the subject from a commercial standpoint and there is a surprisingly large amount of sentiment attached to the decisions where sentiment should not exist. Engineering matters above all others should be reduced to a commercial basis. It is easy to do this if the proper arrangements are made, but we must acknowledge that it is not always done even in plants of much greater importance than the local installation in a hotel building.

It is pointed out that the problem of a power plant for a hotel is much more involved and complicated than for the ordinary office building. There is the question of live steam for steam tables in the restaurants; power for a great variety of purposes as well as lighting and heating; refrigeration in all its modern applications, ice, water and other services which do not enter into the ordinary building requiring power. However, in just so much as the problem is complicated it should receive all the more the careful attention of the manager and engineer, and there is no excuse for not knowing in dollars and cents just what the plant is doing and why it is the best solution of the problem under the circumstances.

Out of the mass of correspondents, however, one communication looms up with exceptional brilliancy. This is from J. H. Morrow, chief engineer of the Great Northern hotel, Chicago, who gives definite figures as to operating costs both before and after installing an isolated plant.

For the year 1905, to quote from the report, the expense of the plant while buying current from a central station was:

Light and power	$38,563.01
Fuel	9,502.65
Water	1,943.40
Engineers and machinists supplies	6,291.34
Salaries	8,540.99
Total	$64,841.39

In 1906 with a private plant in operation the figures reached were as follows:

Fuel	$15,512.43
Water	984.90
Engineers and machinists' supplies	4,140.51
Salaries	12,647.87
Total	$33,285.71

These figures show a net saving of $31,555.68 or a percentage of 48.5 in favor of the isolated plant.

In the report it is stated that in the year 1906 the private plant generated 312,894 kilowatt-hours in excess of the power used in 1905, making the performance in reality still more favorable to the isolated plant. Figures for the ensuing year up to 1910 show the same character of performance.

It is stated in connection with this plant that the average cost per kilowatt-hour for the last four years has been 2.49 cents, the price of coal during this period ranging from $2.45 to $2.55 per ton. Adding the capital charges to the plant still leaves the total cost per kilowatt-hour under three cents.

This operating cost it must be understood includes every expense attached to the power plant such as fuel, salaries, electric and machinery supplies, water, lamp renewals, pumping water, elevator service, steam for all auxiliaries, including kitchen service, refrigerating plant, steam vacuum-cleaning system, air compressors and for heating the building.

It is gratifying to note that in this instance the matter of power costs has been gone into in such a thorough manner and the incident serves as a shining example to other engineers placed in similar positions or having charge of any kind of a power plant where this subject has not been given the attention it deserves.

Central Station vs. Isolated Plant

As an illustration of one of the ways in which a thing should not be done, the meeting held under the auspices of the Blue Room Engineering Society at the Engineering Societies' building, New York, on the evening of October 9, was complete.

The meeting was called for the purpose of discussing the encroachment of the central station upon the field of the isolated plant, and, if possible, of showing to the public in general, and to the operating engineer in particular, that in the great majority of cases the owners of office, manufacturing and apartment buildings can make the electric current which they use for light and power cheaper than it can be bought. Had the meeting been controlled by the most skilful manipulator in the service of the Public Service Corporation, the tendency to discourage further discussion could not have been increased.

In the language of the player folk—"It was a frost." Announced for eight o'clock, the meeting was called to order at ten minutes of nine, when the chairman introduced Charles G. Armstrong, who for more than an hour read extracts from his report to the Bureau of Municipal Research, and explained the meaning of the charts which had been distributed to the audience. The common, or garden variety of operating engineer in whose interest the meeting was supposedly held, is not a chart-and-formula-loving animal, and while he is in thorough sympathy with the argument and appreciative of the facts presented, he commenced to chafe under an hour's application of this treatment.

Charles M. Ripley, whose association with the Engineering Supervision Company as its vice-president, would have rendered it difficult for him to gain the sympathy of his audience under the best of circumstances, for the engineer dreads supervision as much as he does the central station, followed and took up another hour in the wearisome presentation, detail by detail and exhibit by exhibit, of a long argument, the essentials of which were available in print and which needed only an abstract presentation. With an utter lack of appreciation of the temper of his audience, and a woeful display of tact, he interspersed his remarks with such crass advertisements of the Supervision Company as to evoke audible expressions of disapproval, and in some cases outspoken protests.

It is unquestionable that in the majority of buildings in New York City, heat, light and power can be furnished on the premises for less than it can be bought from any outside source, however well equipped, but it is not the engineer who needs educating along this line, it is the owner and the architect who must be convinced.

Owners are much like other human beings, and are like others influenced by their economic interests. Not one will use central-station service when he can get the same service for less money elsewhere if he knows it. It is up to the engineer to show where this service can be got for the least money, and unless his is one of the exceptional plants in which this is not the case, show that he can do it in his own plant better than it can be done elsewhere.

Striking Heat Conversions

It is a striking demonstration of the conversion of heat into mechanical energy to stand beside a steam turbine and place one's hand upon the cool exhaust pipe on one side while a few inches upstream on the boiler side of the nozzles the pipe is so hot that it cannot be touched. In flowing through these few inches the steam has dropped upward of two hundred and fifty degrees in temperature and converted from ten to fifteen or more per cent. of its molecular kinetic energy into useful work.

A more striking conversion, however, takes place in the boiler where in the gas surrounding the heating surface there is a temperature of some two thousand degrees while the exterior surface of the iron has a temperature not many degrees above that of the water within.

In a film of a thousandth of an inch in thickness the velocity of the gas molecules is reduced to that corresponding (in connection with their mass) to the lower temperature, and the velocity of a mass of iron and of water molecules is accelerated until the energy required for such acceleration equals that given up by the gas molecules in their retardation.

The Tall Chimney

In discussing the report of the Committee on Power Generation, of the American Street and Interurban Railway Engineering Association, at Atlantic City recently, C. O. Mailloux expressed himself very forcibly as of the opinion that the day of the tall chimney is passing and that draft will be furnished by mechanical means. Upon the editorial page of Power for May, 1890, the opinion was expressed that "the time will come when these tall chimneys will serve no more useful purpose than that of monuments to the folly of their builders."

The fuel used in manufacturing costs too much to allow twenty-five per cent. of it to escape up the stack "to warm up outdoors," and the mechanical engineer whom we were quoting opined even at that time that "future development will be in the direction of regaining from the flue gases every possible unit of heat, and the maintenance of the draft, not by the maintenance of a tall column of hot gas, but directly by mechanical means."

Dr. Stillman W. Robinson Dead

On Monday morning, October 31, Prof. Stillman Williams Robinson, inventor and engineering expert, died at his home in Columbus, O. He was born on a farm in Reading, Vt., March 6, 1838. As a boy he took great interest in matters mechanical, but what with farm work and the poor educational facilities in a country place he had slight opportunity to follow his inclination. He studied and vented in a small way, however, and hen he became fifteen years old his pportunity to give up farming presented self, and his sixteenth year was spent operating a sawmill and erecting a rniture factory and a gristmill. At the d of that year he entered upon a four ears' apprenticeship in the machine shop F. B. Gilman, Springfield, Vt. He udied during this period, meeting his hool expenses by occasional jobs at e trade he was learning.

DR. STILLMAN W. ROBINSON

In 1860, with eight dollars and a kit stencil tools, young Robinson started r Ann Arbor, Mich., to attend the State University there. He defrayed the expenses of the trip by cutting stencils from town to town, and reached Ann Arbor with fifty dollars in his pocket. During his college career he supported himself by making stethoscopes and graduating thermometer scales. He was graduated in 1863, with the degree of civil engineer. On leaving the university he became assistant engineer for the United States Lake Survey, acting in that capacity from 1863 until 1866. In the latter year he became instructor in mechanical and civil engineering at the University of Michigan, and from 1867 to 1870 he was instructor in mining engineering and geodesy at the same institution. In 1870 he was appointed professor of mechanical engineering and physics at the University of Illinois, the first school of mechanical engineering in the country, where he remained until 1878, in the last-named year being dean of the college of engineering at this university. From 1878 to 1891 he was professor of mechanical engineering and physics in the Ohio State University, and from 1891 to 1895 professor of mechanical engineering.

In 1895 Professor Robinson resigned to become inventor and consulting mechanical engineer for the Wire Grip and Fastener Company and the McKay Shoe Machinery Company. His first invention was a thermometer-graduating machine, made while in college. Various other inventions, notably a number for shoe machines, yielded him about forty patents. He was adverse to the use of cams in machine elements and was peculiarly successful in designing link and lever motions. Many of his machines are in use today.

In addition to his other work he was State inspector of railroads and bridges from 1880 to 1884, consulting engineer to the Santa Fé road from 1887 to 1890, and consulting engineer of the Lick telescope mountings in 1887.

In 1896 the degree of doctor of science was conferred upon Professor Robinson by the Ohio State University, and in 1899 he was elected to the position of professor emeritus in mechanical engineering.

Professor Robinson was a member of the American Society of Mechanical Engineers, American Society of Civil Engineers, and of the Naval Architects and Marine Engineers; a fellow of the American Association for the Advancement of Science, and of the Society for the Promotion of Engineering Education. He was author of a number of technical works, among them "Principles of Mechanism," a college textbook, besides being a contributor of engineering literature to the technical press.

At the time of his death he was president of the S. W. Robinson and Son Company of his home city.

In personality he was kind and genial, always ready with helpful suggestions, particularly for the young men around him. His stimulating influence will be recalled with gratitude by many engineers who came in contact with him either as students or in business.

The story of his rise from the obscurity, poverty and hardship incidental to the life of the New England farm-born lad to the position of a master of engineering through his own unaided personal efforts is peculiarly American in its character and should serve as an incentive, if not an inspiration, to all who read of his life's work. Of him it can be most fittingly said:

> "Lives of great men all remind us
> We can make our lives sublime
> And departing leave behind us
> Footprints on the sands of time."

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Elliott Reducing Valves

The accompanying illustrations show the design of types *A* and *B* reducing valves made by the Elliott Company, 6907 Susquehanna street, Pittsburg, Penn. The operation of either valve is as follows:

The steam or other fluids pass through either valve, Figs. 1 and 2, in the direction indicated by the arrows, the high-pressure steam entering at *H*, and at reduced pressure is discharged at *S*. The main steam valve *A* is connected to the spindle *B* with the two case caps *C* surrounding the india-rubber tube *D*, which rests in the bottom, as shown, and is also accommodated in a similar space in the bottom of the extension or distance piece *E*. This india-rubber tube is reinforced on the outside by a number of neatly fitting rings *F*, as shown in Fig. 3. The object of these rings is to permit the tube to stretch only in the direction of its length, keeping it constant in diameter (see second illustration, Fig. 3), and at the same time eliminating friction by the tube coming in contact with the tube case *G*, Figs. 1 and 2. By making the tube *D* the exact diameter of the seat of the valve *A*, the valve is put in equilibrium to any pressure acting in the direction of the arrows.

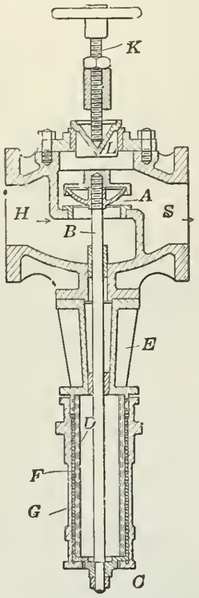


Fig. 1

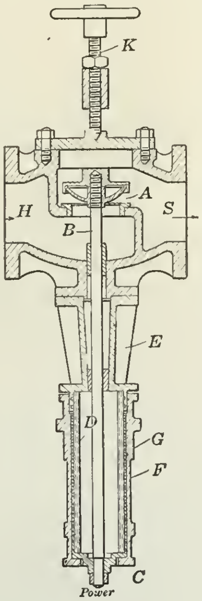


Fig. 2.

The tube-case cap *C* rests in the tube case *G*, which is provided at the proper end with wings *H*, shown in Fig. 4, to receive the upward pull of the springs *I* which are put in tension by the yoke *J* and adjusting screw *K*. The pull on the spring disturbs the equilibrium established and forces the valve *A* upward and the amount of tension determines the reduction in steam pressure.

If the area of the valve *A* is 1 square inch and the upward pull on the spring is 10 pounds as soon as the steam pressure on top of the valve equals 10 pounds, the weight will overcome the spring and the india-rubber tube will commence to stretch and allow the valve *A* to close, thus preventing further passage of steam. The india-rubber tube *D* with its surrounding wings is in effect an elastic cylinder, the bottom end of the tube-case cap *C* acting as a piston. The movement is frictionless, as the piston does not move in the cylinder, but the body of this cylinder or rubber tube is itself elongated. The movement of the tube being divided among the rings *F*, the tube even when extended is so perfectly supported as not to be distressed in the slightest, it is claimed. The tension of the springs, therefore, controls the reduced pressure which can be varied by adjusting the screw *K*.

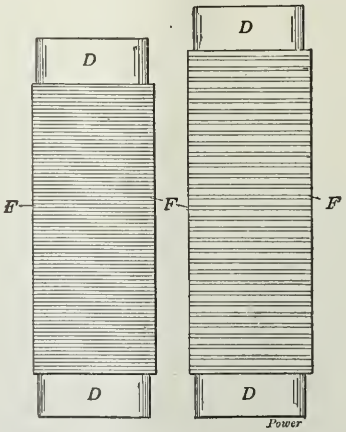


Fig. 3

The pressure on the adjusting screw *K* is received by the valve *L*. This valve *L* is made with about 20 per cent. less diameter than the valve *A*, and will lift if the pressure above the valve *A* exceeds by 20 per cent. the pressure for which the reducing valve is set. The valve *L*, therefore, acts as a relief valve against overpressure should the vale *A* fail to shut off the steam.

The rubber tube is protected from injury from steam by being filled with condensed steam. Under ordinary working conditions the tube will last for years, it is said. In case the tube should prove defective a new one can be inserted by simply decreasing the tension on the springs and removing the yoke. The tube can then be taken from the tube casing.

Type *B* reducing valve, Fig. 2, is in every way similar to type *A* with the exception that no safety valve is provided for the escape of the excess pressure.

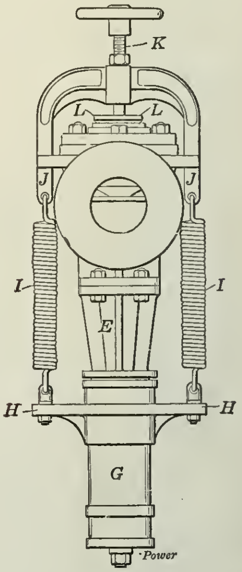


Fig. 4

These valves are made in sizes ranging from ½ inch up to and including 4 inches, with either flange or screw connections, and are capable of working under initial pressure of 250 pounds per square inch. The valves up to and including 1½ inches are made of gun bronze; above that of cast iron with bronze wearing parts.

The Manufacture of Humphrey Pumps

Brown Boverie & Co., of Baden, Switzerland, have just completed an arrangement with the Pump and Power Company, Limited, whereby they become the licensees of that company for the manufacture of the Humphrey pump and compressor. The Humphrey pump, it will be recalled, is that in which a charge of gas is exploded in contact with the water to be pumped, and the functions of compression, etc., are effected by the surge of the water column. The same arrangement is used as an air compressor by allowing the other end of the reciprocating water column to draw air into a separate chamber, situated at the distant end of the discharge pipe and then compress it and deliver it under pressure on the next explosive charge.

Siemens Schuckertwerke are making at Berlin, one of these pumps which will deliver a million gallons of water per hour to a hight of 200 feet. One of the new experimental pumps to be erected at Dudley Port, Eng., will have a suction lift of 18 feet and a direct line of 160 feet. The Pump and Power Company, Limited, has in hand for the Indian government some small pumps with suction lift, and among the new types, which they are developing, is a well pump in which practically the whole lift is suction.

Friction Test upon Keystone Grease

The figures shown in the accompanying table represent the average results of a test upon a sample of Keystone grease as recently conducted with a Cornell oil-testing machine at the testing laboratory of the William Cramp & Sons Ship and Engine Building Company.

The test consisted of runs at various pressures, each run being of one hour's duration with readings taken at five-minute intervals and fifteen minutes allowed between runs for the temperature of the journal to become constant. The journal was of a standard shape, having an area of 14 square inches and filled with Cramp's government babbitt and grooved diagonally. The grease was fed from a hand-pressure grease cup of about three ounces capacity.

FRICTION TEST.

Run.	Room Temperature, Degrees Fahrenheit	Journal Temperature, Degrees Fahrenheit	Increase in Temperature of Journal over Room.	Friction in Pounds.	Pressure in Pounds.	Average Coefficient of Friction.
No. 1	78.6	157.7	79.1	11.9	2000	0.00595
No. 2	79.2	170.3	91.1	16.4	3000	0.00547
No. 3	80.3	176.1	95.7	15.3	4000	0.00333
No. 4	83.0	185.2	102.2	16.3	5000	0.00326

The speed of the shaft was 450 revolu- ons per minute and during each entire un the grease maintained its original onsistency.

NEW PUBLICATIONS

LECTRIC WIRING. By Joseph G. Branch. Published by the Branch Publishing Company, Chicago, 1910. Cloth; 272 pages, 5x7½ inches; 93 illustrations; numerous tables. Price, $2 net.

The subject is treated in the form of a tchism; beginning with the elementary nsideration of a simple current flowing rough a single conductor, and leading to the more involved aspects of the bject. Problems are frequently emoyed to illustrate points under consideration and application is made of the various formulas commonly used in making wiring calculations. For the greater part low-potential wiring is dealt with, such as that employed for the lighting and power circuits of buildings or small towns; and only casual reference is made to high-potential wiring.

In point of criticism, it may be said that many of the definitions are extremely loose and would be apt to mislead one not already familiar with electricity. Also, the text contains numerous statements which are contradictory and do not conform to uptodate practice.

SOCIETY NOTES

During the month of November at the Modern Science Club of Brooklyn, the following lectures will be given: "The Panama Canal," by N. B. Payne, November 15; "Injectors," by Frank O'Leary, November 22; "A Modern Hydraulic Installation," by George A. Orrok, November 29.

At the regular annual meeting of the Association of Edison Purchasing Agents the present officers were elected to hold office from 1910 until 1911: H. C. Lucas, president; W. H. Francis, first vice-president; J. W. Brennan, second vice-president; A. W. Banks, Jr., treasurer; J. J. Miley, stock controller.

The monthly meeting of the Engineers of Boston will be in charge of the American Society of Mechanical Engineers, the Boston section of the American Institute of Electrical Engineers and the Boston Society of Civil Engineers coöperating. The meeting will be devoted to a topical discussion of a paper written by D. T. Randall, engineer in charge of fuel testing with A. D. Little Company, on the subject of "Smoke Abatement."

A paper on the "Rotary Kiln," by Ellis Soper, of Detroit, will be presented at the New York meeting of the American Society of Mechanical Engineers, 29 West Thirty-ninth street, on November 9. Following this paper, Charles Whiting Baker, editor of *Engineering News*, will give an illustrated lecture on the Panama canal. The date of this meeting has been changed from the customary one because the meeting would have come this month on election day.

The sixth annual entertainment and reception of the Universal Craftsmen Council of Engineers, of the Metropolitan district, including the boroughs of Manhattan, Brooklyn, Queens, Bronx and Richmond and the cities of Newark, Jersey City and Paterson, N. J., was held at Lexington Opera House, New York City, on Saturday evening, October 29. There was a large attendance, and the usual merry time was had. An enjoyable vaudeville entertainment was followed by a long dancing program.

Secretary Calvin W. Rice, of the American Society of Mechanical Engineers has just returned from a trip to San Francisco, where he attended an organization of the local members for the purpose of holding meetings in that city. A. M. Hunt was elected chairman and T. W. Ransom secretary of the local organization, with Prof. W. F. Durand, E. C. Jones and Thomas Morrin as an executive committee. A resolution was adopted inviting the American Society of Mechanical Engineers to hold its semi-annual meeting in San Francisco during the Panama-Pacific Exposition of 1915, and the suggestion was made to the council of the society that an international meeting of mechanical-engineering societies be held conjointly with that of the American society. It has not yet been decided, however, that the international exposition of 1915 will be held in San Francisco, although that city is making very earnest efforts for it, including the passage of a constitutional amendment, enabling the State to raise $5,000,000 by taxation for this purpose.

PERSONAL

Charles Guest Norris, who has been in this country for a number of months, disposing of American interests in English inventions, entertained a party at the Engineers' Club on Wednesday evening, October 26, on the eve of his return to England.

S. B. Whinery, manufacturers' agent, 95 Liberty street, New York City, announces that he has recently made arrangements with the American Well Works, of Aurora, Ill., to represent them in the New York district in the sale of centrifugal and turbine pumping machinery.

R. T. Hudson, formerly shift engineer at Norwich Electricity Company, Norfolk, Eng., and for the past five years shift engineer at the municipal power station of Johannesburg, S. A., has lately been appointed shift engineer at the Randfontein Estates Gold Mining Company's central power station, Transvaal, S. A., which has lately been equipped with Parsons turbo-alternators and has an aggregate capacity of 20,000 kilowatts, the largest units being two 6000-kilowatt sets.

Cost of Power

On Saturday evening, October 29, a public mass meeting was held in the auditorium of the United Engineering Societies' building, New York City, under the auspices of the Blue Room Engineering Society, to listen to addresses by Charles G. Armstrong, mechanical engineer for New York City, and C. M. Ripley.

Mr. Armstrong's address was based on an investigation which he has been conducting during the last three months for the Bureau of Municipal Research for the purpose of ascertaining the relative cost of light, heat and power when the electric current is furnished by the central station and when it is furnished by the isolated plant. An abstract of Mr. Armstrong's address will be offered in an early issue.

Mr. Ripley's paper on "The Cost of Heat, Light and Power," was presented in the issue of POWER for October 25.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. Theodore H. Haberkorn, Fort Wayne, Ind. 973,651.

GAS ENGINE. Charles M. Leech, Lima, Ohio. 973,792.

INTERNAL COMBUSTION MOTOR. Donald J. McKinnon, Toronto, Ontario, Canada, assignor of two-thirds to K. F. McKinnon and Ernest W. Oliver, Toronto, Canada, and William A. McKinnon, Birmingham, England. 973,800.

WATER MOTOR. Norman A. Stamm, Glendora, Cal. 973,823.

ROTARY GAS ENGINE. Herbert I. Wilber, Dorchester, Mass. 973,833.

TIDE-WATER MOTOR. Martin Logan, New York, N. Y. 973,869.

TURBINE. Gustav Engisch, Berlin, Germany. 973,929.

BOILERS, FURNACES AND GAS PRODUCERS

OIL VAPORIZER AND BURNER. Oral A. Makinson, Pratt, Kan. 973,526.

WATER-TUBE BOILER. Jiro Miyabara, Tokyo, Japan. 973,541.

DEVICE FOR INCREASING HEAT. Gamaliel C. St. John and Frederic A. D. Rankins, New York, N. Y. 973,581.

FURNACE. Gilbert R. Haigh and John R. Fortune, Detroit, Mich., assignors to Murphy Iron Works, Detroit, Mich., a Corporation of Michigan. 973,652.

STEAM BOILER. Carl Marinius Olsen, Birkeröd, near Sandefjord, Norway. 973,692.

SUPERHEATER BOILER. James P. Sneddon, Barberton, Ohio, assignor to the Babcock & Wilcox Company, New York, N. Y., a Corporation of New Jersey. 973,717.

VERTICAL BOILER. Tozaburo Suzuki, Sunamura, Japan. 973,720.

GAS-GENERATING OIL BURNER. Thomas Muehleisen, San Diego, Cal. 973,804.

FURNACE. Luther L. Knox, Avalon, Penn., assignor to Keystone Furnace Construction Company, Pittsburg, Penn., a Corporation of Pennsylvania. 973,943.

SMOKE-CONSUMING APPARATUS FOR LOCOMOTIVES. John Loftus, Albany, N. Y., assignor of one-half to Anthony Hughes, Philadelphia, Penn. 973,947.

SMOKE-PREVENTING AND FUEL-ECONOMIZING APPLIANCE. William John Storey and Gilbert Storey, Albert Park, near Melbourne, Victoria, Australia. 973,991.

LIQUID FUEL BURNER. Charles A. Swartz and Sam Finley, Frederick, Okla. 973,995.

POWER PLANT AUXILIARIES AND APPLIANCES

DETACHABLE BOILER FLUE. Ole C. Borgen, Sharon, N. D. 973,471.

HYDRAULIC PUMP. Robert Cairns, Toronto, Ontario, Canada. 973,473.

HOSE CLAMP. Milton P. McLaughlin, Wakefield, Mass., assignor of forty-five one-hundredths to H. C. McCarty, Williamsport, Penn. 973,532.

VALVE. William L. Osborne, Chicago, Ill. 973,551.

CARBURETER. Joseph G. Williams, Buffalo, N. Y. 973,602.

TUBE AND HEADER CONNECTION. James C. Alexander, Roseburg, Ore. 973,610.

STEAM SUPERHEATER. Henry W. Jacobs, Topeka, Kan. 973,662.

BLOWOFF VALVE. Henry W. Jacobs, Topeka, Kan. 973,663.

PUMP, ETC. Fitz W. Machlet, Elizabeth, N. J. 973,679.

GOVERNING MECHANISM FOR MIXED PRESSURE TURBINES. Frederick Samuelson, Rugby, England, assignor to General Electric Company, a Corporation of New York. 973,705.

TUBE CLEANER. Thomas Andrews, Rockaway, N. J. 973,740.

CARBURETER. William C. Carter, St. Louis, Mo. 973,755.

LIQUID FUEL STRAINER. William C. Carter, St. Louis, Mo. 973,756.

WATER HEATER AND PURIFIER. William Asa Gibson, Peoria, Ill. 973,773.

CENTRIFUGAL FORCE PUMP. Thomas Russell Hayton, Mount Vernon, Wash. 973,782.

CARBURETER. Percival Charles Cannon, London, England, assignor to Montague Stanley Napier and Selwyn Francis Edge, London, England. 973,855.

SPARK PLUG FOR EXPLOSIVE ENGINES. Martin T. Minogue, Philadelphia, Penn., assignor, by direct and mesne assignments, to The Superior Motor Specialty Company, New York, N. Y., a Corporation of New York. 973,873.

CARBURETER. Burt Neulon Pierce, Indianapolis, Ind. 973,877.

CARBURETER. William F. Rothe, East St. Louis, Ill., assignor to Wm. F. Rothe & Company, East St. Louis, Ill., a Corporation of Illinois. 973,882.

CARBURETER. William Ernest Haines, Cheltenham, England. 973,937.

PACKING FOR FLUID ENGINES. William F. Purcell, Brooklyn, N. Y. 973,972.

SCALE PREVENTER FOR BOILERS. John W. Stephens, Staunton, Va. 973,989.

FIRING INDICATOR AND RECORDER. Asa White Kenney Bilnngs, Habana, Cuba. 974,017.

ELECTRICAL INVENTIONS AND APPLICATIONS

INCANDESCENT ELECTRIC LAMP. Thomas E. Robertson, Catford, England, assignor to General Electric Company, a Corporation of New York. 973,703.

ELECTRIC MOTOR. Edwin S. Pillsbury, St. Louis, Mo., assignor to Century Electric Company, a Corporation of Missouri. 973,878.

CIRCUIT CONTROLLER. Winfield A. Atwood, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 973,613.

TERMINAL FOR ELECTRICAL APPARATUS. William B. Potter, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 973,560.

SELECTOR SWITCH. Edward B. Craft, Chicago, Ill., assignor to Western Electric Company, Chicago, Ill., a Corporation of Illinois. 973,762.

CIRCUIT-CLOSING DEVICE. Leslie R. Saunders, Los Angeles, Cal. 973,818.

ELECTRICAL OSCILLATOR. Frederick K. Vreeland, Montclair, N. J., assignor to Wireless Telegraph Exploitation Company, New York N. Y., a Corporation of New York. 973,826.

CURRENT CONTROLLER. Frank L. Sessions, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 973,885.

ELECTRICAL CIRCUIT BREAKER. Harve R. Stuart, Wilkinsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 973,889.

ELECTRIC BRAKE. Robert R. Dunlop, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 973,926.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power
and The Engineer

Vol. 32 NEW YORK, NOVEMBER 15, 1910 No. 46

A fireman was once placed in charge of a string of automatic stokers. His duties consisted of regulating the stoker feeds according to the load and necessitated considerable walking back and forth in the boiler room.

He soon found that he could save himself a considerable amount of trouble by setting the stokers to feed a trifle heavy and running the two end boilers which were farthest from the engine room with the side doors open.

By making his headquarters in front of these boilers and regulating the two side doors, he could remain on the bench longer than anyone who had ever held down the job.

In the vernacular of the sportsman, he was a "sidestepper."

This practice, while making it easy for him temporarily, of course impaired the efficiency of the plant.

It was not long before his little trick was discovered and he was discharged.

It is easy to fall into little labor-saving habits about the plant, which, however innocent in the ordinary run of events, may in the aggregate mean a serious loss in economy, an increase in the liability to accident.

In the engine room it is natural to curtail the number of inspection trips to a remote machine, trusting to the oiling system to do its work without attention.

Ninety-nine times out of a hundred it will do so, but the expense and trouble caused by the lack of attention that hundredth time will in most cases more than overbalance the saving in energy realized by the attendant in cutting the number of his trips down to the minimum.

Such things do not pay in the end. It is much better to give all the attention to the devices in one's charge that safe and efficient operation demands.

The better performance that will result, the freedom from hot boxes and shutdowns, will add to reputation and put one in line for the bigger job that will sooner or later come along.

In short, laziness does not pay.

Every individual is or should be actuated by self-interest. This should be the mainspring of all human action. But selfishness unless intelligent defeats itself.

It has been said that one may hold a dime so near the eye as to hide a dollar a little farther away.

It is ignorant selfishness that does this.

It was the same ignorant selfishness that influenced the fireman to waste his employer's coal in order to save a little physical effort.

If his self-interest had intelligently directed his inventive ability toward increasing the efficiency of the apparatus under his care and improving his own mental equipment and manual skill, instead of being discharged for incompetence and dishonesty his chances for advancement would have multiplied.

He saw the dime, represented by the few minutes of extra rest between trips, and held it so close that it obscured the dollar of promotion and better pay that lay just beyond.

He wanted the most that he could get for the least effort, but ignorantly gave too much effort in the wrong direction.

Modern life is full of inspiring examples of the results of intelligently directed selfishness—selfishness that wants the best for itself and realizes that it can only be had by giving the best in exchange.

You can tell by the way a man starts in

Design and Erection of Steam Piping

By Francis H. Davies

A statement of the fundamental principles to be followed in properly laying out a piping system and a discussion of the materials which should be used to conform to good practice.

The best and most uptodate practice in the layout of steam piping calls for the observance of certain well defined principles which may be briefly summarized as follows:

The lines of piping should be kept as short as possible, and their number as few as can be managed conveniently.

The diameter of the pipes should be the result of a compromise. To keep down radiation losses the pipes should be small, but not unduly small, otherwise the drop in pressure will be excessive.

The number of joints should be reduced to a minimum.

The water drainage must be ample and carefully thought out.

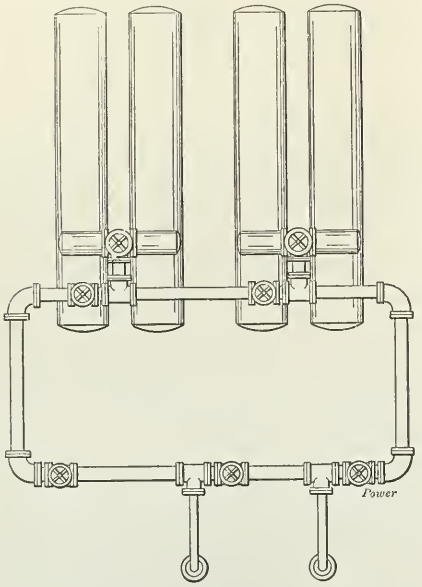


Fig. 1. Ring Main

The expansion and contraction must be provided for.

The valves must be so arranged that in cases of emergency the faulty sections may be cut out without seriously interfering with the operation of the plant.

The lagging, besides being efficient as an insulator of heat, must be applied so as not to interfere with repairs, and it should be neat and lasting.

The erection must be carefully carried out in such a manner that no part is at any time under an undue stress.

It is obvious that to fulfil all these conditions a steam-piping system, even when of comparatively small magnitude, must be carefully thought out in every detail. The subject can best be investigated by considering the preceding conditions seriatim.

Length and Number of Pipes

These considerations go to the root of the matter and primarily influence the whole layout, as the dogma "that at all costs the run must be kept as short as possible" brings into prominence the question of the ring main and the duplicate and bypass systems versus the header and other simple arrangements. The point to consider is, whether in the interests of safety it is better to provide a costly and heat-wasting ring main, as shown in Fig. 1, which will permit of any engine being fed from either of two directions in case of emergency, or, whether the engineer should place his faith on good material and workmanship, and install a simple direct system of piping, as shown in Fig. 2. Those who have followed the trend of modern practice will have little hesitation in accepting the latter course. The ring main, sometimes equipped with cross connections and numerous feeders, was once popular, particularly in large electric-power plants where reliability of service is always looked upon as the chief consideration. In the early days this was justified by the frequent disastrous failures of piping systems composed, for want of better, of comparatively low-grade material. Now, however, the position is entirely different, and experience has proven that a simple piping system of the best materials can be depended upon and is preferable in every way. By reason of the greater number of joints and the larger radiating surface, a ring-piping system is always a source of extra trouble and loss. To justify it fully, it must be used as a ring and the sections must not be shut down with a view to economy. It will, of course, give a high factor of safety under such conditions, but this is at the expense of a large waste by condensation. Realization of this point usually leads to portions of such a system being shut down and opened only in an emergency; but this policy is in itself dangerous as the act of hurriedly opening the cold-pipe line is apt to cause water hammer and blow out the joints, not to mention the risk of water in the cylinders.

Whatever the system of steam mains, it is clear that the relative positions of the boilers and engines is an important factor in determining the total length of piping. To the power-house designer this is a point that often causes trouble, and generally the efficiency of the piping system is sacrificed to considerations of space and the convenient arrangement of the plant. This, in most cases, is essential to some extent, but it frequently happens that by a comparatively slight alteration in the general layout of the plant considerable economy may be effected both in the cost of the piping and in the condensation losses. For instance, it is not unusual to place the main flues and economizers between the boilers and the engines, thereby materially adding to the length of the steam feeders. This may be necessary in some cases, but consideration should be given to the question of whether it will not pay to place them either above or below the boilers and thus reduce the length of the steam piping. Such a course has the additional advantage of reducing the cost of the land and buildings which are at times very important items. Another consideration in the case of large power plants is the relative positions of the boilers or batteries of boilers. Instances are not wanting where a little forethought would have resulted in a much better arrangement of the various batteries at no extra cost and at the same time an appreciable saving in piping and fuel would have resulted.

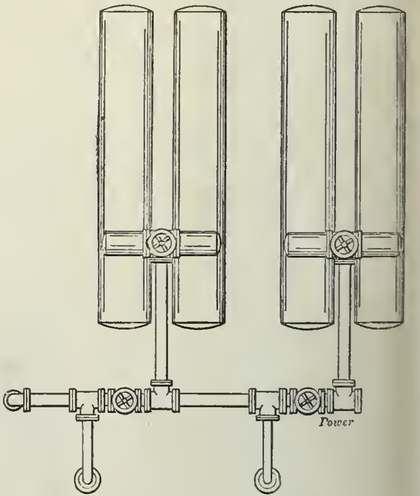


Fig. 2. Direct System

Those who have not minutely studied the question seldom appreciate, to the full extent, the heat loss that goes on in piping even when well lagged. A reliable authority has stated that under average conditions a well covered steam pipe continually under steam requires 372 pounds of coal per year per square foot of pipe surface to make up for the heat lost through condensation. In the case of a bare pipe the loss is placed at 1120 pounds of coal per year. This, of course, is an approximate statement as the heat loss will vary directly with the pressure and temperature of the steam, the location of the pipes, and the quality of the covering. Still, it serves as a rough indication of how serious this loss is and how well it pays to reduce the length of piping. For example, accepting 372 pounds per square foot of surface as correct, a 4-inch steam-pipe line 200 feet in length will have a radiating surface of approximately 200 square feet, and will waste 37 tons of coal per year. Even a 25 per cent. reduction in this coupled with the reduced initial outlay and the reduced losses from joints would be worth having. There are many steam lines where more than 25 per cent. could have been saved in the length of run if the matter had received a little more forethought.

Diameter of Pipes

It is advisable to keep down the diameter of steam pipes for two reasons: first, because of the saving in cost; second, because of lower heat losses as a result of a smaller radiating surface. On the other hand, the drop in pressure has to be considered and must be kept within reasonable limits. One of the most important features in the design of piping is the velocity of flow; the maximum permissible rate is quoted by various authorities at from 6000 to 8000 feet per minute. These, however, are figures which cannot be accepted definitely, as circumstances will sometimes allow them to be increased, and at other times it is necessary to use lower velocities. The allowable drop in pressure, the initial pressure of the steam and the diameter of the pipe, all have to be taken into account in determining the best and most economical velocity. With low-pressure steam a high velocity of flow is permissible, because the pressure being low, means a comparatively small loss in friction between the steam and the pipe. The curves in Fig. 3 show the amount of steam, in pounds per minute at 150 pounds per square inch, that will pass through pipes of various diameters at different velocities. Given the pounds of steam an engine is using per minute and the diameter of the pipe, the velocity in any case may be ascertained approximately from these curves; if the pressure be other than 150 pounds, the variation is practically proportional.

A general rule to allow for the drop of pressure in steam pipes is 1 pound per 100 feet of length. This is sometimes exceeded, and when the extra cost and wastefulness of large pipes is considered, it is better to err upon the smaller side. It should be recognized that a drop in pressure due to friction is not entirely a bad thing when within reasonable limits, as much of the energy absorbed is transformed into heat which is given back to the steam.

Before leaving the subject of drop in pressure, mention will be made of a phenomenon which is perhaps seldom appreciated. When several engines are fed from one main header and happen to take their steam at the same instant, a momentary but excessive drop in pressure takes place. In itself, it may not mean much loss of energy, but it is obvious that it must recur frequently under ordinary conditions, and in the aggregate may affect the running of the engines, especially if they are fully loaded. In some plants this trouble has been so pronounced as to require steam receivers to be placed at intervals in the pipe line. By providing a local reserve of steam this arrangement reduces the pressure drop and at the same time minimizes the vibration which is liable to arise under these circumstances. It further allows a smaller diameter of pipe to be employed, in which case the reduced radiation losses will compensate for those arising from the receiver. A vessel of any reasonable size will help matters, but it is usual to provide one for each engine, having a capacity of about fifty times that of the high-pressure cylinder up to the point of cutoff. This may appear excessive and probably is for many cases, but it is a receiver capacity that is actually employed in many plants. Steam separators help matters by equalizing the flow in the main pipe, but generally speaking, they are too small to have sufficient effect.

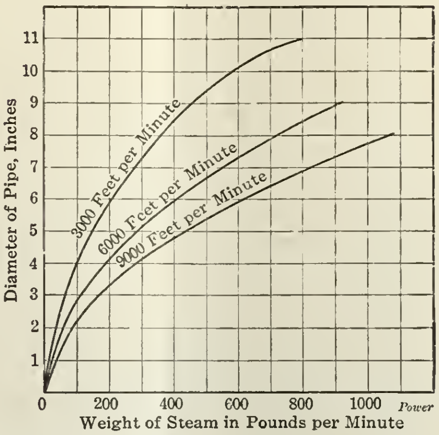


Fig. 3. Velocity of Steam at 150 Pounds Pressure in Various Sized Pipes

Joints

It is desirable to minimize the number of joints as each is a potential source of waste and trouble. As steel pipe can now be supplied in lengths up to 20 feet, it is the number of valves and tees which determine the number of joints; therefore, the designer is left without much leeway in this direction. However, the practice of riveting branch pipes to the main instead of installing tees is to be recommended as it makes a much more satisfactory job. The branch should be shaped so as to fit the pipe closely and should be double riveted and calked.

It is perhaps hardly necessary to call attention to the elementary principle that flanges should meet each other truly; yet, without doubt, many leaky joints are due to carelessness in this respect. No amount of tightening up the bolts on one side will suffice where the two faces are much out; this expedient, which is used too often, places severe stresses upon a part which is not designed to withstand them. Care and perhaps a little extra expense on the flanges, will be compensated for by freedom from repairs.

The flanges should be made of mild steel, stamped from the solid piece and provided with strengthening collars. Pipes up to about 6 inches in diameter, usually have their flanges attached when hot, by screwing on with a fine thread and expanding, calking and brazing to complete the job. In the larger sizes it is best to fasten the flanges by means of rivets, the holes being drilled radially to the center of the pipe after the flange is placed upon it; the burs are removed and the rivets are hydraulically driven. A facing strip projecting from $\frac{1}{16}$ to $\frac{1}{8}$ inch from the flange is essential with large pipes, although with small ones a good joint can be made if the whole surface is accurately faced. With pipes of large diameters the pulling up on the bolts distorts the flange, and only the inner part near the pipe, which is not under the influence of such stresses, is of use in preventing leakage. It is with the idea of reducing flange distortion to a minimum that the bolt circle is made as small as possible.

Drainage

An efficient drainage system is by no means an easy thing to devise. Where superheated steam is used this feature loses much of its importance, but in plants operating with saturated steam too much consideration cannot be given to it. A good system is one which may be depended upon to keep all piping, which is under pressure, free from water when no steam is flowing to the engines, so that when a valve is opened there is no danger of a rush of water into the engine. Further, it must be arranged so that in the event of condensation taking place when steam is flowing or the boilers priming, the water is cleared from the pipes immediately and without water hammer. To secure effective drainage of a pipe in which there is no flow of steam, it is essential that horizontal pipes should be slightly pitched, permitting the water to flow by gravity to a point from which it can be drained. Where steam is flowing, the water should be drained by gravity in the same direction as the steam. It is obvious that water in a horizontal pipe carrying high-velocity steam cannot be expected to force its way against the latter, even when the pipe is sufficiently out of level to bring gravity into effect.

It follows, therefore, that many of the small drains frequently placed haphazard in a steam line are really almost useless, because much of the water never reaches them; and if it does, it cannot flow away freely owing to the pipes being too small. Hence, the steam carries the bulk of the water past these drains into the engine or the separator. There seems little doubt that in the majority of plants, designed without due regard to the foregoing, the bulk of the water finds its way out through the engine separators. Many engineers are now realizing the existence of this state of affairs, and are doing away with the numerous small drains and wasteful traps, and are relying more upon large separators, properly placed. In an installation planned upon these lines the feeders from the boilers will slope toward the main header, and the boiler stop valves will be at the highest points. The header will be horizontal, for if it feeds several engines it is useless to give it a slope with the idea of the water draining by gravity, as with certain engines running the direction of the steam will oppose such a slope. This header will have numerous small drain pipes leading to traps, but will rely for draining entirely upon the large pipes carrying the steam to the engines; these will be connected to the bottom of the header and will pitch downward to separators of ample size communicating with traps. The steam pipes to the engines will rise from the tops of these separators, and with a 180-degree bend will meet the engine stop valves. Such a system entirely fulfils the conditions; in no part of the system can water remain to any extent in pockets, and when it drains, its direction of flow is always the same as that of the steam.

Expansion

Efficient expansion bends or joints are a necessity in a steam-piping system of considerable length; for ordinary purposes the bends are preferable. Often with expansion joints there is trouble through sticking and weeping at the glands, and unless the pipe is well anchored and the expansion joint is thus forced to do its work, it will jam and throw the stress upon the ordinary joints. The adjustment is delicate and it is often difficult to arrive at the exact point where leakage at the gland is stopped without interference with the free movement. The linear expansion of a steel pipe is 0.36 inch per 10 feet of length with superheated steam at 550 degrees Fahrenheit, from which it is evident that the expansion in a long line is considerable. This large expansion, alternating with contraction, is sure to result in trouble if proper measures are not taken to meet it. Each section of piping containing an expansion piece should be securely anchored at each end, as this will insure that each of the bends receive their share of the work and no more. The best method of anchoring is to use a heavy casting securely fixed to the walls or columns of the building; to this the pipe is clamped by heavy wrought-iron straps. It must be borne in mind that the branch pipes leading from a main header will tend to move with the latter; therefore, they should be arranged, as far as possible, in such a manner that this tendency is not interfered with. If they are very short and are coupled direct to the engines they will act practically as anchors, with resulting stresses on the joints which the expansion pieces may not be able to relieve. When the latter are in the form of U-bends there are two important points to remember when erecting them: first, they should be placed horizontal, as in any other position they will interfere seriously with drainage; the mistake of placing them vertically is frequently made, and it cannot be too strongly condemned; second, the bends should be sprung into place, so that their movement when hot or cold will be about equal. In erecting steam piping, generally, it is good practice to have it under a certain amount of initial tension, so that when steam is turned on the expansion will compensate for this and leave the pipe unstrained when in its normal hot condition.

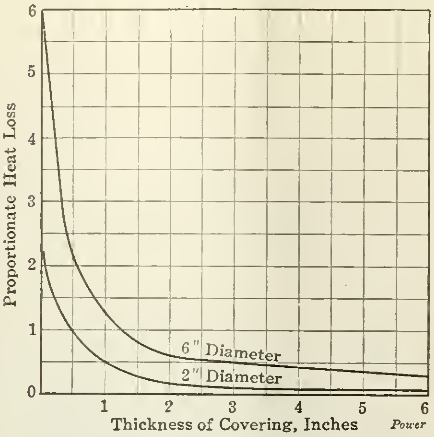


FIG. 4. HEAT SAVED BY VARIOUS THICKNESSES OF COVERING

Valves

The arrangement of the valves in a steam line is more a matter of common sense than anything else and it would be difficult to go far wrong. The choice of the valves, however, is another question; and it is very easy to fall into error, particularly if cheapness is a prime consideration. For high-pressure work a cheap valve is the most expensive in the long run, as the cost of frequent repairs soon outweighs the original saving. This is particularly true with small valves, of which there are many on the market that should be classed as rubbish. The characteristics of a good valve are that it be of the "full-way" type, so constructed that water pockets will not form in it; the seat, in addition to being ample in size, must not become distorted under superheated steam, and the construction must be such that there is no likelihood of the valve jamming if left unused for long periods. As a matter of fact, in a well ordered engine room all valves are periodically opened and closed to keep them free, but in too many instances this precaution is neglected, hence, the designer must reckon with this possibility. Valves over 7 inches in diameter should be fitted with a bypass in order that the pipe may be warmed before opening the large valve; this costs very little more and is a great convenience. Wherever possible, it is well to place valves horizontally so as to avoid the formation of water pockets. This is perhaps not usual practice, but it is one of those points which it is well to observe in the interests of safety.

Pipe Covering

The prevention of condensation by the use of efficient lagging is a subject upon which much might be said. In deciding upon the relative merits of various coverings it is necessary to appreciate fully the basic principle upon which their action as heat insulators depends. Much depends upon the material itself, but its efficiency as a lagging depends largely upon the presence of air in minute particles in the fibers. Air in the ordinary sense is not a good heat insulator, but when split up in this manner it is exceedingly efficient; for this reason materials such as hair, wool and felt rank high in the list of good nonconductors. In practice, however, such substances are quite unsuitable for steam-pipe covering owing to their inflammability; therefore, it is necessary to use substitutes such as fossil meal, carbonate of magnesia and slag wool. The latter stands first in efficiency, having an insulating value of about 75 per cent. of that of hair felt. It forms the basis of many good coverings, being combined, according to the fancy of the manufacturer, with other substances which give durability and binding power. A lagging of this class, if well applied, will save from 80 to 90 per cent. of the heat which would be radiated from the bare pipe, and although it may cost more than one of cheaper materials it will give greater satisfaction and will prove a better investment.

The usual thickness to which a lagging is applied is 2 inches, and it is seldom necessary to exceed this; on the other hand, it should never be less than 1 inch as even the best of materials require this as the minimum. The curves in Fig. 4 show the proportionate amount of heat that is saved by first-class coverings of various thicknesses in the case of 2- and 6-inch pipes. From this it will be seen that while the first inch has by far the greater effect, anything over 3 inches scarcely pays. The function of the third inch is more of a mechanical protection than anything else and forms a reserve of insulation to allow for external and internal wear. As is well known, the covering of pipes subjected to vibration, wears to a considerable extent internally, the inner part being gradually ground to powder until ultimately the lagging works loose and breaks off. There is no preventative of this, other than first-class material and workmanship; and the same may be said of external wear. For instance, a canvas wrapping of good quality will be a much better protection to the outside of the lagging than the coat of black varnish with which cheap work is often finished. Wear is more pronounced in small vertical pipes than in any others, owing to the fact that they are more susceptible to vibration. They require special care, and cheap work in this instance is simply money thrown away.

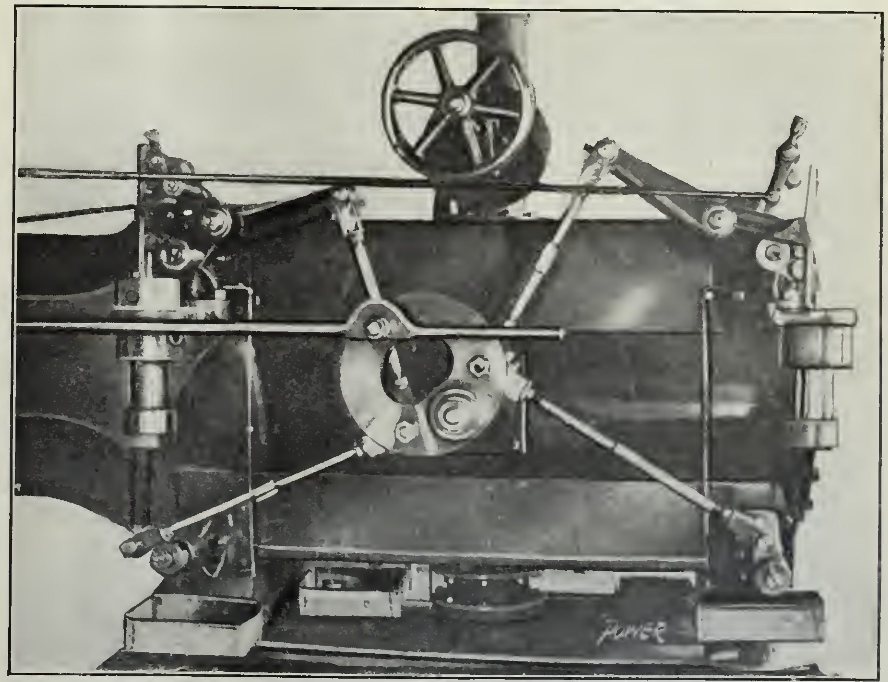
The practice of covering flanges is fortunately becoming more general, but it is still more the rule than the exception to find power plants in which, from a mistaken idea of the value of convenience, the flanges are left quite bare and the covering beveled down so as to leave room to get at the nuts. There is no excuse for this practice in view of the numerous removable flange covers on the market, and it is strange that so many engineers fail to appreciate that every foot of uncovered radiating surface represents a yearly cost in fuel many times the sum that would suffice to cover the flanges and stop the waste.

Material for Pipes

The question of the best material for high-pressure steam piping was settled long ago in favor of steel. A solid-drawn or lap-welded steel pipe insures the greatest safety, being in this respect far ahead of copper or cast iron. The former is now seldom used, and the employment of the latter is rapidly being discontinued except for low pressures, for which it is alone suitable. The advantages of copper are its durability, toughness and the fact that it will to some extent compensate within itself for expansion. For these reasons it is now often used for expansion bends, although for all practical purposes steel is quite as good and far more reliable. Modern solid-drawn copper pipes are infinitely preferable to the brazed pipes formerly used. With the latter it is only a question of time for the zinc, which forms an essential part of the solder, to disappear under the influence of chemical action and the high temperature, leaving at the seam a cellular structure of copper which is weak and liable to give out. The solid drawn copper pipe, while free from this defect, still suffers from another which it is impossible to cure, namely, the hardening of the pipe near the flanges. This is the result of the continual bending due to the vibration; once the pipe hardens and loses its ductility it also loses the only valid reason for its use. Recognizing the inherent weakness of copper pipe, it is the practice in some classes of work, particularly in marine work, to reinforce them either by brazing a spiral copper band around the pipe or by winding over it a copper or delta metal wire at a tension of about 1.5 tons per square inch. Another method is to space circumferential steel bands at short intervals either by shrinking or by means of cotters. So far as added strength goes, all these means are effective, but they largely increase the cost and at the same time add materially to the heat-radiating surface. In short, copper pipe is not worth considering, although many engineers of the old school still cling to it.

Cast iron is in a somewhat different position by reason of its cheapness, but such pipes have been the cause of many bad accidents and should be barred for pressures over 80 or 100 pounds per square inch. It is particularly susceptible to water hammer and herein lies its chief danger. Cast-iron elbows and tees are still often used in conjunction with steel piping, but it would be better if they were entirely abolished, especially in those positions where the effect of water hammer is often severe. Elbows and tees to be safe under high pressure should be constructed of a specially tough mixture of cast steel.

Peculiar Corliss Valve Gear



An Odd Type of Valve Gear

In a power plant at Augusta, Ga., one of the engines is equipped with the peculiar design of Corliss valve gear shown in the accompanying illustration. The engine bears no nameplate and no one in the plant knew much about it.

It has a 26½x60-inch cylinder, runs at a speed of 68 revolutions per minute and is belted to a railway generator. The wristplate is actuated by a single eccentric and reach rod; the exhaust valves are operated in the usual way. The steam valve gear, however, has peculiar features. The valve stem has an arm on which is hung the dashpot rod and plunger. A catchblock is fitted to the upright piece, shown extending out of the dashpot, and is attached to the plunger at a point about 2 inches from the top. A spring holds the catchblock on the upright arm against the catchblock on the arm actuated by the wristplate, or, in other words, always forces the vertical catchblock stem toward the center of the cylinder, where it is in a position to engage with the steel block of the reciprocating arm.

On the stud of the arm, actuated from the wristplate connection, is an A-shaped arm working on a stud, the shorter end of the arm containing a roll. As the governor balls assume a higher plane of rotation, this arm and roll are moved toward the disengaging rod attached to the dashpot plunger, and as the speed of the engine increases the rod comes in contact with the roll and is disengaged from the catchblock of the arm, when the arm and dashpot plunger drop and close the valve. The operation of the valve gear at the other end of the cylinder is, of course, precisely the same.

Under-Feed Automatic Stokers

By J. F. Springer

In the Jones under-feed stoker coal is forced into the furnace under the fire by means of a steam-driven reciprocating ram. In the gravity under-feed stoker the fuel is forced underneath the fire and onto inclined grates by two mechanically operated rams.

In the eighties of the last century, E. W. Jones attacked the problem of burning the Oregon fir. This fuel contains an excessive amount of moisture which hindered its use on a commercial scale. Of course, the users might have sun-dried the fuel themselves. But this would have been quite an undertaking, requiring both space and time. There is another way and this is the fundamental germ of the stoker which this inventor evolved. Why not use the fire of one moment to prepare the fuel for the fire of the next?

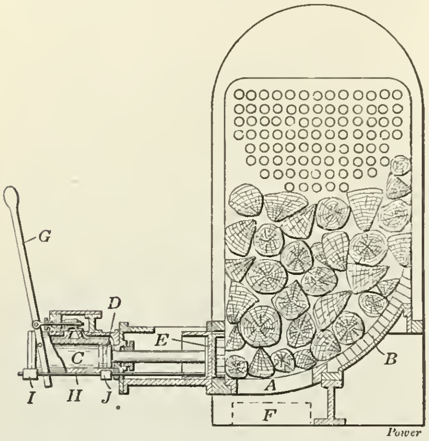


Fig. 1. Jones' Wood Stoker Patented in 1889

Jones conceived the idea of putting the green wood right up underneath the fire. It would then be near the source of heat and advantageously situated for drying. Further, such a position was an admirable one for the next operation, that of the actual introduction into the fire itself. He worked out this idea in such a way that the green wood was gradually carried from the outside supply to the fire. As the wood approached the point of combustion it became more and more dried.

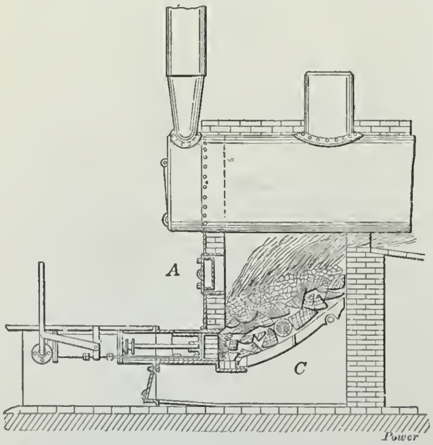


Fig. 2. Jones' Improved Wood Stoker Patented in 1892

But how were lengths of wood to be introduced into the fire from the bottom? The inventor's answer to this question is the basis of the United States patent issued to him in 1889. This, however, was not, apparently, his first thought in so far as the mechanism is concerned. He conceived the idea of using levers to force in the fuel, but abandoned this method in the patent for that of a ram performing thrusting movements at intervals. In Fig. 1 is shown a vertical section of a stoker, furnace and boiler. At *A* is a floor of grate bars. At *B*, and continuing the upward incline begun by the grate bars, is a dead surface formed of firebrick. The fuel, in suitable lengths, lies in this wedge-shaped receptacle. The stoker consists of the steam cylinder *C*, the piston *D* and a ram *E*. By alternately admitting steam at the front and rear of the piston, the ram is operated back and forth. The head of the ram is protected by a covering of firebrick. There is an ashpit at *F*. The lever *G* is connected with a slide valve which controls the admission of steam through the two inlets which open into the cylinder at the front and rear. This lever is operated by hand. The rod *H* is connected with the ram; on this rod are two tappets *I* and *J*. It is the back and forth motion of this rod with its tappets which automatically closes the steam valve before the piston reaches the end of the stroke. The wedge shape of the fire chamber tends to support the mass of fuel in it regardless of the position of the ram.

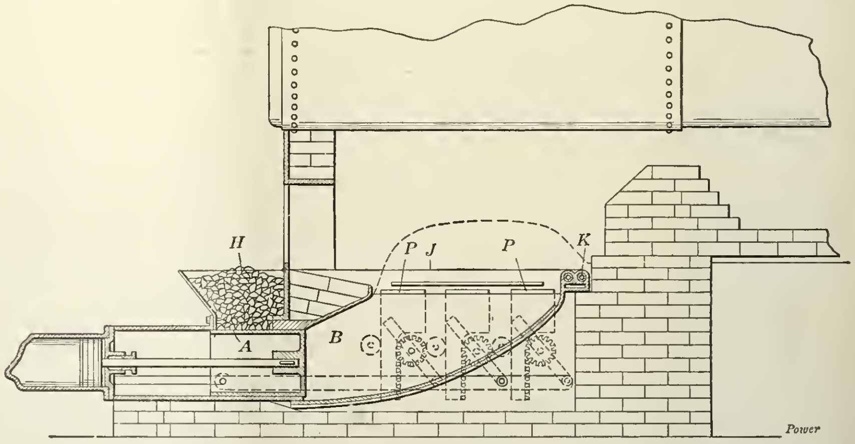


Fig. 3. Longitudinal Section of Jones' Coal Stoker Patented in 1892

A fire burning at the top of the pile and supplied with air from below by way of the grate bars *A* will not only heat the boiler tubes above but dry the green fuel underneath. This drying operation distils from the wood, gases which are valuable as combustible materials. Instead of being wasted, these are carried upward by the current of air coming from below, and so pass through the intense portion of the fire.

This stoker proved to be a gratifying success. The patent specification of 1889 includes coal as a possible fuel, but it is scarcely probable that the stoker, as originally designed, was well adapted to its use. About two and one-half years later, Jones took out another patent. In Fig. 2 is shown a longitudinal section of the stoker covered by this patent. The principal difference between this and the original machine lies in the arrangement of the dead plates. In Fig. 1 the dead plate is inclined, is quite wide and lies at the rear, while in Fig. 2 it is reduced in width, is practically

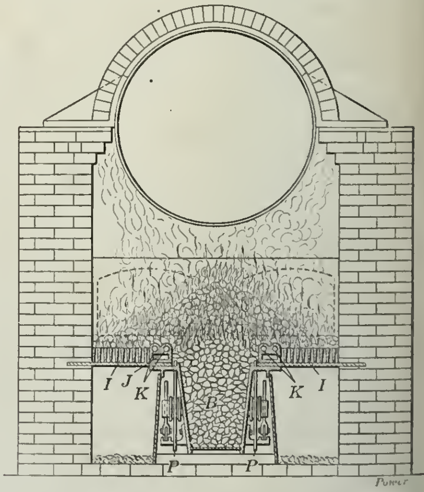


Fig. 4. Cross-section of Jones' Coal Stoker

horizontal and lies in front. The supply of air is markedly changed by thus providing the long grate bars *C*. Further, this arrangement is no doubt better for the disposition of the ashes. The door *A* gives access to the upper part of the fire.

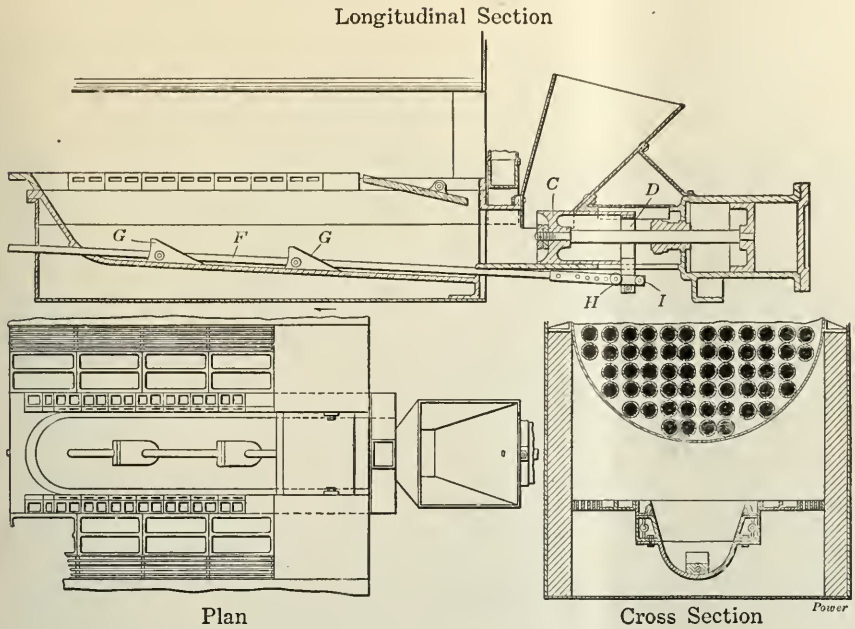


FIG. 5. STOKER PATENTED BY ROE IN 1897

Although it was a very excellent apparatus, very few people wanted a steam-operated stoker to use in connection with wood. It is a principle sometimes forgotten by inventors that there is no immediate advantage, at least, in inventing something which nobody wants. But it was not quite that bad with the Jones wood stoker. There was some field for it although a very limited one. What was really needed in the industrial world was a stoker that would do for coal what this stoker had already done for the moist Oregon fir wood.

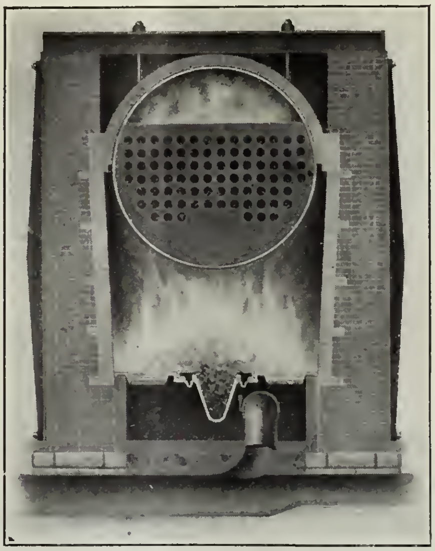


FIG. 6. CROSS-SECTION OF JONES' STOKER AS IMPROVED BY F. A. DALEY

This problem the same inventor had already set about to solve. Upon the same day on which his second patent for a stoker especially designed to feed wood was issued, he obtained a patent for a stoking apparatus conceived with especial reference to coal.

By referring to Figs. 3 and 4, the construction of this apparatus may be understood. The fuel magazine *B* is a long box, narrower at the bottom than at the top and sloping upward from front to rear. The bottom and sides of this magazine are solid; there is no grate at all. In fact, none is needed as this receptacle is not intended to be the seat of combustion. The arrangement for forcing in the fuel remains much the same. The ram plunger is not protected by firebrick, since it is not brought into close proximity to the excessive heat of the fire. The head of the ram is of such form and size as to close the delivery opening *A* in the hopper *H* when the ram is forward, thus preventing clogging at the rear. To provide air, the tube *J* surrounds the fuel magazine *B* at the top on both sides and at the rear. Slots along the sides of this tube, open toward the center. By forcing air through this slotted pipe, a supply of air is furnished to the top of the fuel magazine. An advantage in providing the supply of air from the sides rather than from the bottom lies in the fact that ashes are less liable to clog the openings.

It was at first feared that the tuyere pipe would be subject to two disadvantages. Its proximity to the fire seemed to warrant the expectation that it would be burnt out. It was therefore suggested that the pipe be of wrought iron, which has a melting point much higher than that of cast iron, and that a circulation of water be arranged by a return tube *K*. In actual practice, however, it was found that the water tube deteriorated faster than the tuyere pipe. In fact, the tuyere pipe seemed to receive very fair protection from the air current forced through it. The other disadvantage which was feared related to the clinkers which it was thought might clog the tuyeres. Consequently the inventor provided an elaborate arrangement of slice bars *P*. The intention was that these should be automatically operated up and down in front of the tuyere slots, and thus prevent any tendency to clog.

Upon referring to Fig. 4, it may be noticed that there is an arrangement of grates *I* at either side and above the fuel magazine. They were intended to assist combustion. However, they could not be depended upon for this purpose. The air supplied through the tuyeres was under pressure, while that in the ashpit was not. Consequently, if the draft of the flue were not strong, the air coming in through the slots would find an easier path down through the nearby grates into the ashpit; hence, the grates were ineffectual. This does not seem to have been at once discovered, for the grates were continued in one or two later designs. In the patent issued to the same inventor in 1897, side grates were used, but they are separated by wide dead plates from the tuyeres. At the present time, they are not used at all. But this is, perhaps, partly due to a change in design by which the ashpit was made the reservoir for the air supply.

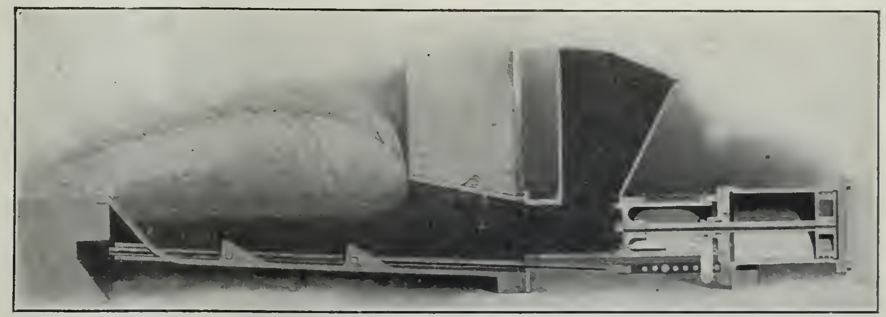


FIG. 7. LONGITUDINAL SECTION OF THE IMPROVED JONES' STOKER

It was found as the development went on, that the sharp upward incline of the floor of the fuel magazine was objectionable, because then the depth of the fuel at the rear was too small. In fact, the fuel at this point was too quickly consumed to permit its prompt replacement. It was also found that the ram plunger did not always supply the fuel where and when it was wanted. It was likewise found that long tuyeres were subject to distortion. And further, it became necessary to augment the air supply. The difficulties with the tuyeres were easily remedied by increasing the cross-section of the supply tube, breaking up long lengths into sections and the like.

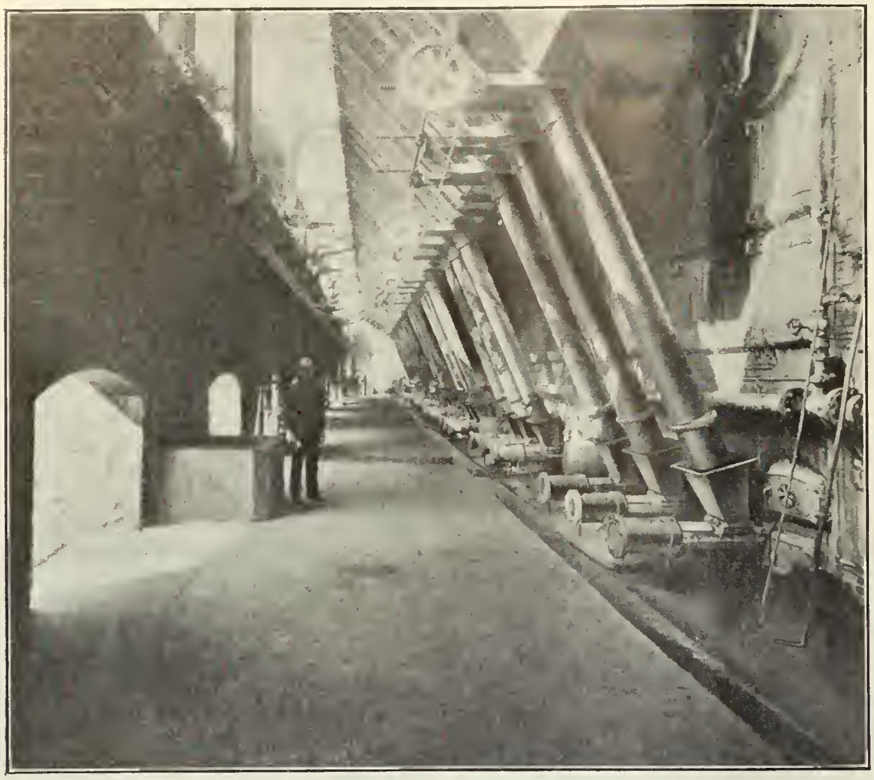


Fig. 8. Typical Installation of Jones' Stokers

The design of the fuel magazine *B*, as shown in Fig. 4, was an idea of permanent value. The fire was prevented from penetrating downward by the exclusion of a supply of air from below. But, while actual combustion was not permitted to take place, the heating effect of the active fire at the top of the chamber was highly beneficial. The strong heat above, combined with the exclusion of oxygen, brought about a distillation of the coal. Gases, valuable as fuel, passed off and upward and soon found a current which carried them, burning as they went, against the boiler. As the coal was forced up by the injection of raw coal from the outside, it became more and more coked. Coke occupies more space than does the coal; hence, when the fuel needed more room, on account of its expansion, it found it at the top of the fuel magazine. The ram plungers, forcing in new instalments of coal, had to overcome considerable friction and weight. If the sides of the fuel magazine were vertical, the friction would be much greater.

J. M. Roe provided a great improvement in this stoker. There was, of course, no necessity of abandoning underfeeding or the lateral inclination of the sides of the fuel magazine. It was essential, however, if this stoker was to go on to very great success, to lengthen and give more uniform depth to the magazine and to secure the introduction of the fuel into this chamber in a more thorough manner than was possible with the use of the original form of the ram plunger. In 1897, Roe received a patent for a device in which these difficulties were overcome. In Fig. 5 are shown a plan and sectional views. The fuel magazine is long and has approximately the same depth throughout. This design made the old process of introducing fresh fuel by means of the simple reciprocation of a plunger inadequate. To move the contents of the magazine throughout the entire length, Roe provided an auxiliary feeding mechanism. To the plunger *C* is attached the lug *D* which projects downward through a slot in the ram chamber. This lug operates the long bar *F* which passes through holes near the bottom of the front and rear walls of the fuel magazine. Within the fuel magazine, several blocks *G* are secured to the rod *F*. When the ram is reciprocated, the rod *F*, and consequently the blocks *G*, participate in its motion. The amount of this participation may be varied by changing the positions of the pins *H* and *I*. The movement of the blocks *G* forces the coal in the magazine upward. By using as many blocks as are required and by adjusting the extent of their motion, this lifting of the fuel is accomplished throughout the length of the magazine. Further, the caking of the upper layers is broken through by the action of the reciprocating blocks.

The plan view in Fig. 5 shows the wide dead plates at both sides which separate the tuyere section from the side grates.

The next step in the development of this type of stoker was taken by F. A. Daley. An improvement devised by him relates to the entire elimination of the side grates. In Fig. 6 is shown a transverse sectional view of the stoker as modified by this inventor and Fig. 7 shows a longitudinal section. The space beneath the fire chamber and the fuel magazine is sealed except for its communication with the compressed-air supply and the tuyere sections. Previous to this advance in design, in the effort to secure an efficient air supply, the tuyere area had been constantly increased until the boxes reached the floor. Mr. Daley now did away with this cumbersome arrangement and substituted the entire underlying chamber as a compressed-air reservoir. A great deal of trouble had been experienced with the warping of the dead plates. The provision of a circulating supply of air under the entire dead-plate area has overcome this, apparently, in a pretty thorough manner, for experience with this rearrangement now covers several years and has been very satisfactory.

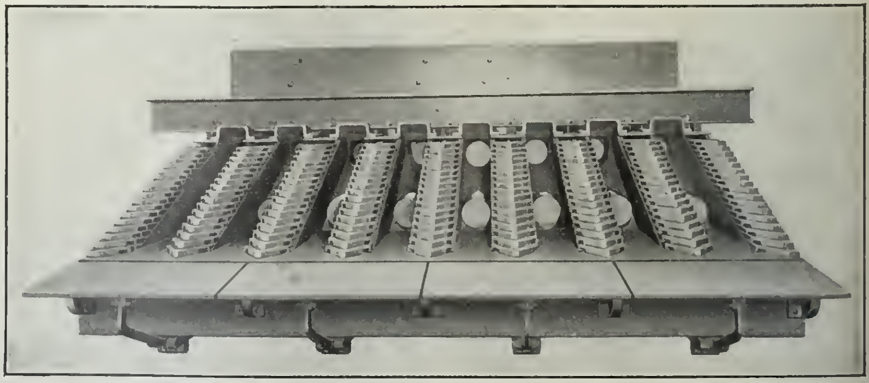


Fig. 9. View of Gravity Under-feed Stoker from Rear

Fig. 8 shows a view of a typical installation of the modern Jones stoker.

THE GRAVITY UNDER-FEED STOKER

The combustion area of the gravity under-feed stoker is divided into longitudinal strips which are alternately grate surfaces and coking sections. Each coking section is quite deep and is provided with two horizontally acting rams, one above the other. These advance and retreat together. Fig. 9 is a view of this stoker looking from the rear end of the combustion chamber. There are eight coking chambers, in some of which the faces of the rams can be observed. Alternating with these are the grate surfaces proper. These consist of flights of V-shaped steps. The effect of this arrangement is to secure for each coking chamber a flaring grate at each side. In the vertical faces of each step are air outlets. The air is thus supplied through longitudinal rows of holes that pretty evenly divide the whole surface.

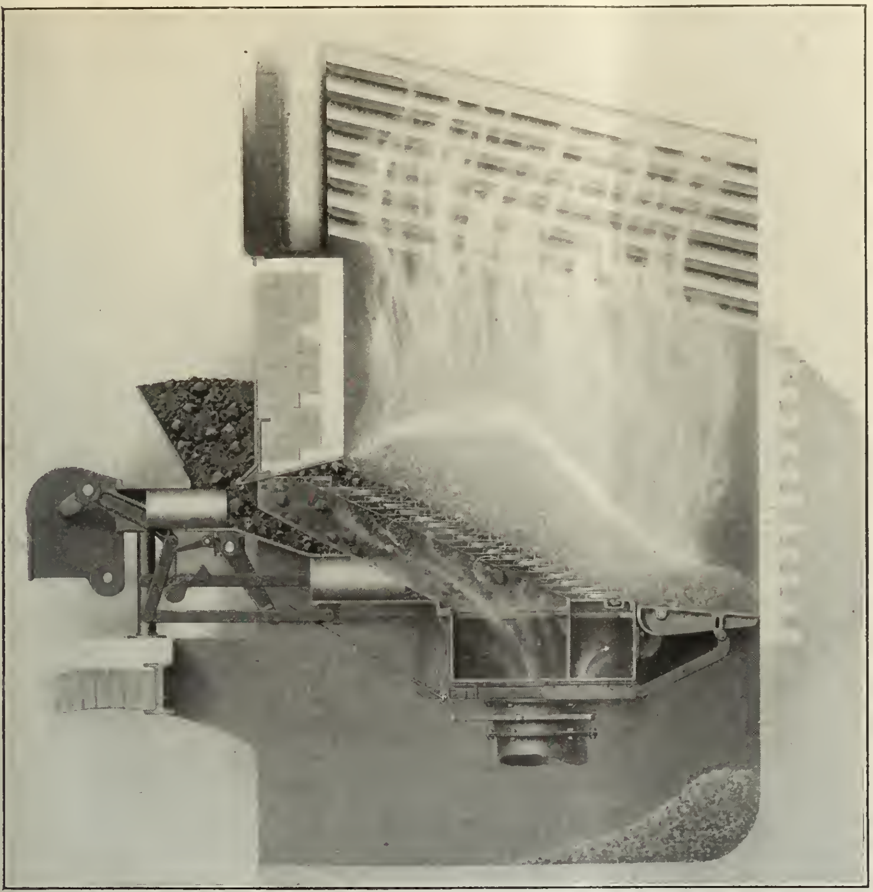


Fig. 10. Longitudinal Section of Gravity Under-feed Stoker, Showing Principle of Operation

By referring to Fig. 10, which is a sectional view, the operation of the stoker may be understood. The coal is automatically or otherwise put into the hopper at the top. The hopper opens into a cylindrical chamber in which the upper ram works. This ram is operated by the crank shaft at the right which is driven by a worm and gear. A single-speed shaft drives a series of worms. The same crank that drives the upper ram is connected to the lower one. The withdrawal of this ram lets drop not only unburnt and partially consumed coal but also a part of the fire itself. When its motion is reversed, it moves on both coal and fire. The two rams work together, so that the depression of the fire surface and the recovery extend along the whole coking section. The series of sections is simultaneously in different stages. This is accomplished by arranging the cranks at different angles on the shaft.

In this stoker, the air supply is furnished by a volume blower. The air is forced into the wind box (see Fig. 10) and passes upward beneath the grate surfaces to the tuyere boxes. The pressure necessary to supply just the right amount of air and the proper draft will vary with the fuel used and details of each individual installation. The tuyere boxes alternate with the coking chambers and thus both the grate surfaces and the retorts themselves are cooled by the moving air. Further, this air, as it cools such parts, becomes more or less heated. This is a distinct advantage, for thus the gases passing off from the coking fuel as well as from the fire itself are not chilled.

It is necessary that the particles of oxygen and those of the combustible gases be thoroughly intermingled. It is claimed that the air supply of this stoker provides a thorough mixture because of the low air pressure and large area of the tuyere opening, which are numerous and arranged for this intermingling. The method of furnishing the air under pressure is well shown in Fig. 10, which is a vertical section through the stoker and fire. The air rises from below, cooling the metal and being itself heated.

The most exposed portions of the stoker are the tuyere sections. These are interlocking and no bolts are employed. They are cooled below and even above somewhat by the air. Further, there are no moving parts exposed to high temperatures. Frictional wear is inconsiderable, largely because of the snail-like movement of the stoking mechanism; the crank shaft makes a rotation once in about two minutes. The reducing gears run in a bath of oil. A shearing pin is employed on the speed shaft between the driving clutch and the sprocket wheel to disconnect the drive when from accident or other cause the resistance becomes greater than a predetermined amount. In Fig. 11 is shown a transverse vertical section through the furnace. It will be seen here that air is supplied through one pipe for a battery of grate sections. When mention is made of grate sections and the like, the ordinary grate is not meant where ash, and even fuel, drop through to a compartment below. In this respect, the tuyere sections do not form a grate at all, for no substances drop through. The waste is carried on down to the dump plates. The power required to operate the stoker need not be very great because there is but little to move.

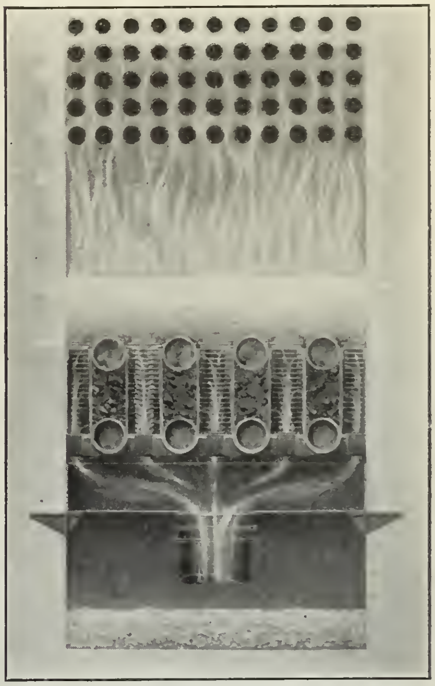


Fig. 11. Cross-section of Gravity Under-feed Stoker

Corrosion of Copper and Brass*

By W. B. Tardy †

> *Of the various causes it is believed that vagrant electric currents escaping from the electric plant of the ship produce by far the greater part of the rapid copper, copper-alloy corrosion. Uniform composition, absence of carbonic-acid gas, etc., and elimination of grounds will lessen corrosion.*

The following theories and notes are chiefly a brief résumé of a paper read before the Institute of Metals, by E. L. Rhead, and the discussion thereon. Nothing of value has been added, the object being to condense the various theories.

In the above-mentioned paper the discussion and recorded experiments tend to establish the following facts; that corrosion of copper and brass is due:

(1) "To methods of manufacture."

(2) "To selective chemical action."

(3) "Corrosion resulting from chemical action set up or promoted by the electrical conditions resulting from the presence, side by side, in the alloy, of different components containing the constituents of the alloy in varying proportions, or from the presence of included impurities."

(4) "The effect of vagrant electric currents escaping from the electrical equipment."

It is conclusively shown that with copper of a fixed purity that part which is hardest, either from processes of manufacture or from operations in working it into shape, is more readily attacked and suffers more rapid deterioration. The evident remedy is uniform annealing, using copper or brass of the softest quality compatible with the service required of it. In this connection it was admitted in the discussion that copper-corrosion troubles in serious form date from 1899, it being stated that this corresponds to the time when the first electrically produced copper, manufactured in 1898, was put on the market. It is difficult to see how this method of producing copper would make it more susceptible to attack, provided, of course, that it is remelted afterward. It would not differ from ordinary smelted copper, except that it might be somewhat purer. This immediately raises the question whether electrically deposited copper has not had certain beneficial minute impurities removed. This question might be very readily answered in a general practical manner by having two similar pipes installed close together to do the same duty, making all conditions identical except having one of the pipes made of electrolytic copper and the other copper purified to the same degree by smelting processes.

In regard to causes (2) and (3) it is stated that it does not seem possible for cause (2) to operate unless affected by (3), but that the electromotive force resulting from the contact of copper with zinc alloys containing copper in large amounts is very feeble, never exceeding 0.08 volt when immersed in sea water. A long series of experiments was made, immersing hard and soft copper and various copper alloys in different saline solutions, from which it is shown that the harder the metal the more susceptible it is to corrosion or to solution, that the relative order of the solution tensions of metals varies with the liquid in which the metal is placed; but that in every case the corrosive action is increased by the presence of carbonic-acid gas. In this connection the discussion brought forth descriptions of the rapid failure of condenser tubes due to lodgment of particles of cinder, ash and coal or other forms of carbon in the tubes. The obvious manner of avoiding troubles due to lodgment of gas-producing solids is to so locate suctions that they are well clear of ash discharges, have pipes, condenser tubes, etc., as straight and with as few pockets as possible, then increase the velocity of flow to the point where all silt or solid matter will be scoured out and swept through pipes or tubes.

It is shown in the discussion by the members of the Institute of Metals that the tubes of a condenser were protected and their life materially lengthened by making electric contact from the main injection pipe to the overboard discharge pipe with a large-capacity cable, thus electrically short-circuiting the condenser. This was the main condenser of an electric-power plant in which there were known to be serious grounds.

From the above case, from our own experience and observation, and the experience of a large number of officers in the service, we offer the opinion that theory No. 4, vagrant electric currents escaping from the electric plant of the ship, furnish both the cause and the electromotive force that produces by far the greater part of the rapid destructive copper, copper-alloy corrosion that occurs on board our ships. It is admitted at the outset that the fact that the ship is purposely made the ground for the wireless has no direct bearing or effect; since its electromotive force, and consequently its current, is alternating, there can be no injurious effect other than the possibility of this alternating effect assisting a direct current to flow where otherwise the resistance was too great for it to be established or maintained by the small direct electromotive force due to leaks from the plant.

It is a matter of common experience and observation that copper pipes most frequently fail at or near bends; one side or other, sometimes both sides, of flanges. It occurs so frequently next to the flange at the outboard end of the line where the suction pipe connects to the stool pipe as to give rise among certain officers to the belief that the corrosion is, or may be, due to thermoelectric effects.

The location of the most destructive corrosion is where one would expect it, reasoning from the first three theories. The greatest differences in character of metal, due to working, are formed at bends and flanges. The greatest differences in composition are found near flanges due to brazing material, and the most active liberation of corrosive gases would naturally be at or near bends. Therefore postulate No. 4 is needed only to account for the rapidity of corrosion. The theory is that vagrant currents will take naturally the easiest path, that the copper pipe, where it is straight and continuous, offers this path, that at a flange the resistance is enormously increased, due to discontinuity of pipe and interposition of gasket, so that the current finds its easiest path through the salt water by the joint. It leaves the pipe just short of one flange and reënters the metal of the pipe just beyond the joint, the point of leaving becoming the positive pole, the metal there going into solution and plating out further along the pipe or being swept out altogether. It is believed that some such action takes place at bends. So far as we know, nothing practical has ever been done to test this theory. With two similar or exact pipes doing as nearly as possible the same duty, one might have an uninsulated copper conductor of large capacity laid parallel, with one end in close mechanical contact with the pipe outboard of the outboard flange and the other end connected to a pump or whatever the pipe is attached to; or a copper sleeve might be slipped over each joint, making electric contact with the pipe each side of the flange, while no such precautions were taken with the similar pipe, long observation ought to give an indirect proof or determination.

With a Siemens-Halske thermocouple galvanometer and proper leads and contacts it ought to be possible to determine any potential differences from point to point in the body of a pipe, also the direction of current—at least such difference as would be produced by grounds—and from these determinations see whether

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*Abstract of a paper published in the *Journal* of the American Society of Naval Engineers.

†Lieutenant commander, United States Navy.

corrosion does always, or usually, occur on the positive side of an interposed resistance.

To Prevent or Limit Corrosion

(1) Uniformity, physical and chemical, in copper and copper alloys is necessary.

(2) Prevent entrance of sewage, carbonic-acid gas, ashes or coal; if latter cannot be prevented, make pipes and tubes as straight and free of pockets as possible, and increase velocity of water so that they will be scoured out.

(3) Eliminate or minimize grounds if the theory of vagrant currents be accepted.

If this be not accepted, make tests on board ship to prove or disprove its validity; those suggested previously, as well as others that will suggest themselves to the practical investigator.

The Combustion of Gas Carbon*

By W. A. and R. L. Clarke

> *By the production of gas from crude oil a byproduct of carbon in the form of lampblack is produced. Many attempts have been made to successfully burn this under boilers but in most cases a poor efficiency has resulted. By the adoption of a special type of furnace, herein described, it has been possible to burn this carbon with an evaporation of four pounds of water per pound of carbon.*

One of the most aggravating propositions confronting a gas engineer is the accumulation of byproducts for which there is practically no market and very little use about the plant. This was the case for many years in the manufacture of coal-gas in which the byproducts of coke, tar and ammonia were necessarily considered in the profits of the plant. The problem has now been worked out by the coal-gas companies through the introduction of water-gas apparatus to consume the coke, and the placing of tar burners under the coal benches and boilers to consume the tar, leaving only the ammonia as a byproduct for commercial sale.

With the advent of the oil-gas industry, which originated on the Pacific coast, a new byproduct was produced, which is an important factor in the economical manufacture of the gas; this is the carbon recovered from the manufacture of gas from crude oil. The first plants manufacturing gas from crude oil considered this carbon a nuisance and expense, but as the industry increased and a larger number of gas plants adopted the system, it came to be recognized generally that the carbon was at least good for fuel under the steam boilers. Later its value was increased by its being briquetted and sold to the public for fuel and, further, by its use in water-gas generators in place of coal or coke. In these generators it produced a gas superior to the oil- or water-gas, even showing an increase over the B.t.u. of the mixed oil- and water-gas. This is at present the hight of economy in the use of this gas-carbon, but all gas plants are not equipped with the necessary water-gas apparatus or the necessary carbon coke-ovens with which to carry out this system; in fact, most of the plants of the Pacific coast are burning this carbon under their steam boilers at a very great loss in efficiency compared with the use of the carbon in a water-gas plant. In view of this the present discussion will be confined to only the practical combustion of gas-carbon under boilers as found in the ordinary gas plant, and will not enter to any extent into the technical and chemical points in the combustion of this carbon.

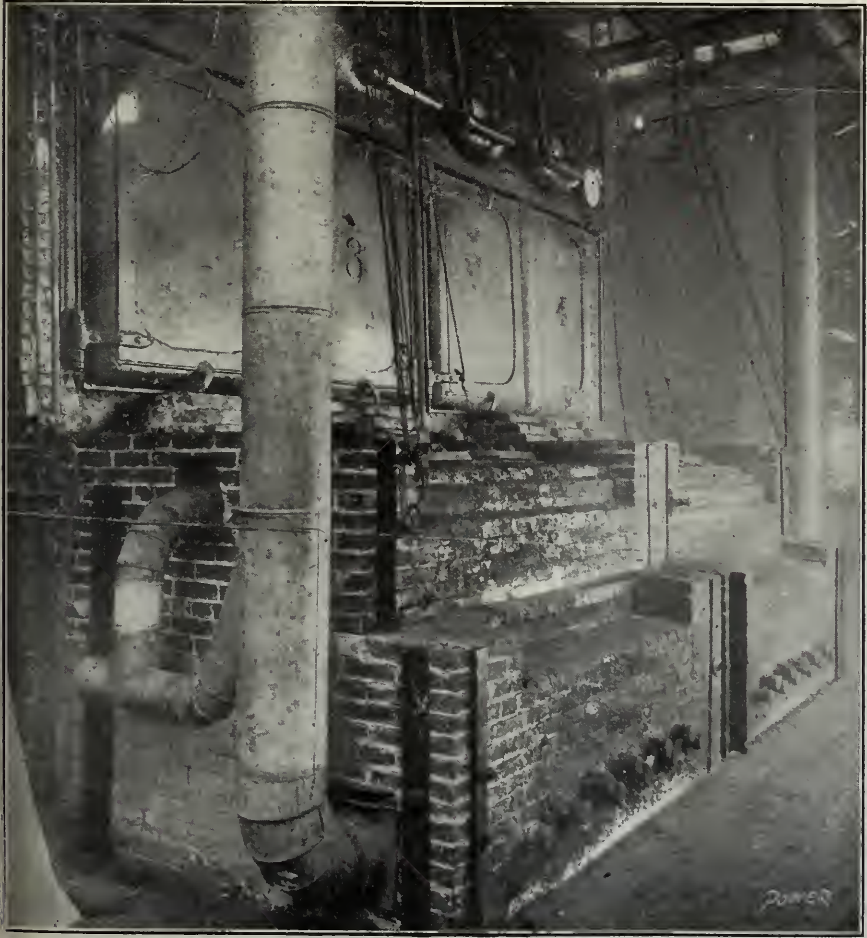

Fig. 1. Exterior View of Furnace

It has been the custom in most gas plants to burn the carbon under boilers of the ordinary form, on cast-iron grates or on improved forms of wrought-iron grate bars, also on water-cooled grate bars. All of these methods, with either natural or forced draft, have been tried out and have resulted in the destruction of the grate bars in a short time due to the excessive heat. Each plant has its individual design of grate bars and a system by which the carbon is burned under the boilers, varying from the simple process of throwing the carbon onto the grates to an exacting method of coking the carbon on the front of the grates and then distributing it over the surface of the grate bars, after the carbon has become incandescent. There has also been some economy gained by the introduction of an oil or tar burner above this layer of carbon.

These methods of burning carbon appear to be absolutely wrong and in opposition to the principles of combustion. It is impossible to bring a fire up to the point of incandescence and then, by throwing on top of it a blanket of wet carbon, to obtain the proper combustion from the fuel. It is radically wrong, even to the extent in which it is carried out in a water-gas plant, where the carbon is burned down by several runs of the generator.

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*A paper read by R. L. Clarke before the Pacific Gas Association.

With this in view, the writers have endeavored to design a furnace in which the principles of combustion, of not alone gas-carbon but also coal and coke, can be carried out with more economical results. In the case of coal and coke, the resultant ash and clinkers are a factor to be considered, but in the particular furnace described this has not been considered, as the presence of ash and clinker in gas-carbon is so small that this point can be overlooked to a great extent.

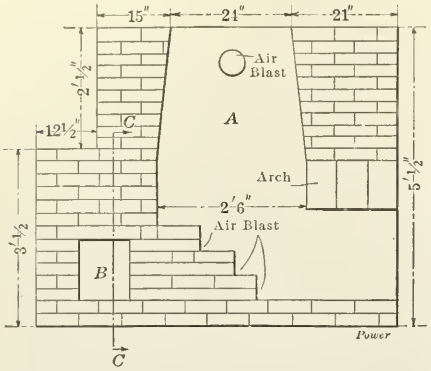


Fig. 2. Section through Furnace Looking Toward the Side

The first attempts to properly burn the gas-carbon were made by the use of an underfeed stoker of a type commonly used in connection with the burning of coal. The results obtained were so encouraging that they led to further experiments and resulted in the design of a gas-carbon furnace similar to a dutch oven. The ordinary evaporation obtained from the burning of gas-carbon on grates, either with forced or natural draft, was not over two pounds of water per pound of carbon, hence the boilers could not be worked to their rated capacity with this system of burning the carbon. By the use of the underfeed stoker, the evaporation was increased to four pounds of water per pound of carbon and the boilers were worked to their rated capacity without the least trouble. Encouraged by this result, a furnace was erected similar to a dutch oven, such as is used in a great many plants which burn sawdust, except that the operation of the furnace is reversed. Fig. 1 shows an exterior view of this furnace for which patents have been applied for.

By referring to the sectional elevation, Fig. 2, it will be noticed that this furnace is designed on the lines of a base-burner coal stove. The main compartment *A* of the furnace, which contains the carbon, is rectangular and is shaped like the frustum of a pyramid, it being several inches wider at the bottom than at the top where the doors for the admission of the carbon are placed. This is to facilitate the feeding of the carbon which is burned at the bottom of the furnace. Experience with this furnace has shown that if the walls were perpendicular, the carbon would "hang up" and not feed down unless some mechanical means were used to force the carbon down into the air blast. Next to this compartment is located a blast flue *B* into the end of which is connected a blast pipe delivering air from an ordinary blower at the pressure of 8 inches of water. The wall dividing this air flue from the main compartment of the furnace in which the carbon is burned is checkered with air flues (see Fig. 3) so that the air is driven in at the base of the main compartment, and after passing through and combining with the body of carbon, the resultant flame passes under an arch into the boiler setting at a point which would ordinarily be the ashpit. The heat of the flame when passing through this archway into the boiler setting is intense and the combustion is very complete, all smoke being eliminated; and after a few hours' run the brickwork of the furnace acquires an almost white heat. It has been necessary to caution the firemen as to the limit of heat that can be maintained on this arch, the walls of the boiler setting and the tubes of the boiler. If care is not taken in regulating the blast, the arch will be melted, necessitating the rebuilding of the furnace. An air blast is also introduced just below the filling doors of the furnace for the double purpose of forcing the smoke and gases, that are formed in the top of this compartment, down into the incandescent carbon at the bottom and also to gradually dry out and remove the moisture from the carbon and force it down through the fire. Air ducts are also placed at the top of the arch for the dual purpose of cooling the arch and aiding in the combustion of the gases which pass the arch; this is similar to the system used in an ordinary regenerative coal-gas bench.

By referring to Fig. 2 it may be seen that the air ducts in the dividing wall between the carbon compartment and the main air chamber are extended by the use of firebrick, so that the lower air ducts extend about half way across the main compartment. This is necessary because of the increased width of the bed of carbon at the base of the furnace, the idea being to give an equal depth of carbon in front of each series of air ducts. There are no grates used in the furnace, it being constructed entirely of firebrick. On account of the accumulation of a slight amount of ash and clinkers, it is necessary about once every ten days to burn down the carbon in the furnace and clear out the accumulation of clinkers and melted brick.

Careful tests made on the boilers to which these furnaces were applied have shown an evaporative efficiency of five pounds of water per pound of carbon. The carbon contained 40 per cent. of moisture when introduced into the furnace, and it has been found necessary to have from 30 to 40 per cent. of moisture present in order to obtain the best results and also to add to the life of the furnace. If carbon with a smaller percentage of moisture is used, the same amount of heat is not obtained and the fire burns well back into the furnace compartment, melting the openings to the air ducts. This seems to prove conclusively that each particle of carbon with its absorbed moisture is really a small water-gas plant in itself. The splitting of the water into its component gases and their combination with the carbon form a water-gas which, at the high temperature of combustion, will melt anything that it comes in contact with, unless handled with caution.

By the use of these furnaces, together with storage hoppers located above them, and proper handling of the carbon by the latest improved mechanical appliances, the dread that every gas man has for the pile of carbon should soon be eliminated and the profits of the gas companies increased in accordance with the increased efficiency of these furnaces over the ordinary method of burning the carbon.

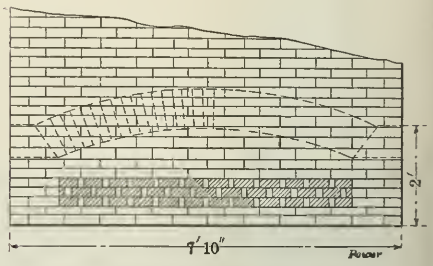


Fig. 3. Brickwork Showing Air Ducts

The annual lecture season at the Modern Science Club, Brooklyn, N. Y., opened on the first Tuesday in October. Lectures will be given regularly each Tuesday throughout the season which ends with June, 1911.

The subject for the October 25 lecture, delivered by A. R. Maujer, was "Peat Fuel." It was shown that great progress is being made in the solution of the problem of preparing peat for use as fuel. Numerous stereopticon views of the Canadian Government's experimental peat plants were shown. At one plant, located at the bog, the peat is machined and air dried. At the other plant, located in Ottawa, the peat fuel is gasified in a double-zone producer and the gas used to operate a 50-kilowatt gas-engine generator. Both of these plants were described in Power for September 13, 1910.

In addition to the Tuesday night lectures, a subscription course is proposed for some other night.

Experiments with Tile Roof Furnaces

By A. Bement

The results of a series of tests to determine the effect of roof tile upon furnace conditions, and a discussion of the report of similar tests performed at the experimental station of the University of Illinois.

Some experiments were made recently for the purpose of reducing furnace temperatures. One of these consisted in the removal of two sections of the roof tiles for the full width of the boiler, see Fig. 1, putting in their place the well known T-baffle tile, shown at *B*. This was for the purpose of exposing the heating surface directly to the fire so that the furnace temperature would be lowered, and to determine whether equally good combustion could be obtained without attaining such a high temperature, as would occur with a full roof. The results proved that the temperature was reduced by a considerable amount and that combustion was good. It was found, however, that the water supply to the lower tubes, with a mechanically fired boiler of the Babcock & Wilcox type, was not sufficient, resulting in the bottom row of tubes being burned in a very short time.

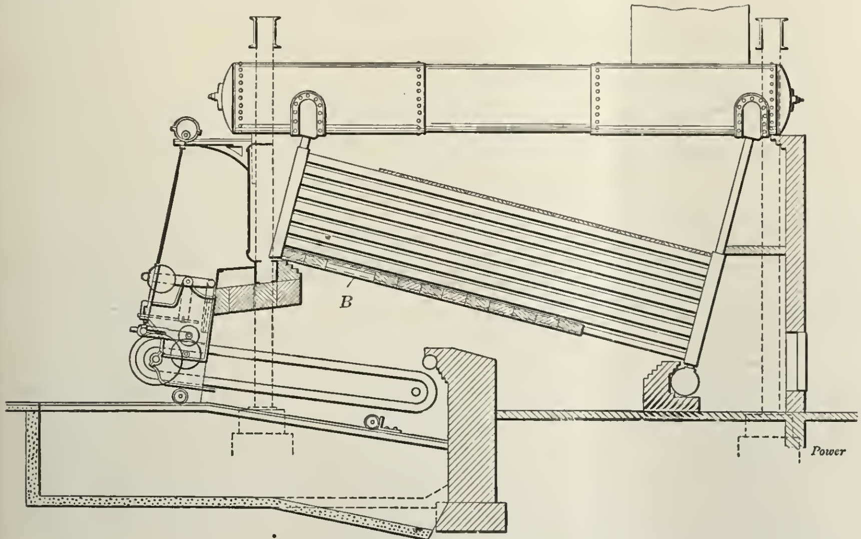


FIG. 1. TWO SECTIONS OF TILE REMOVED

In the case of a Heine boiler, fired by hand, it was found that the presence of the tile roof caused more smoke to be produced than when T-tiles were used, exposing the lower half of the bottom tubes to the fire. This was because the high furnace temperature with the roof caused more sudden and complete volatilization of the coal on account of the high prevailing temperature than did the other condition. For the purpose of correcting this condition, sections of the tile roof were removed as shown at *A*, Fig. 2, and their place filled with T-baffle tile which left the bottom of the tubes exposed as far back as the bridgewall. This was accompanied by a decided reduction in smoke, giving results much better than the tile roof or the entirely exposed tubes. Attention should be directed to the fact, however, that this applies only to hand firing and not to stoker firing.

It appears that the best results with hand firing may be had when the furnace temperature is moderately high, combined with provision which prevents the gases from reaching the tubes too soon; but with stoker firing, high furnace temperature produces more complete combustion. This is because the supply of coal is uniform, likewise the supply of air, hence the combustion capacity is constant. Therefore, it makes no difference how quickly or thoroughly volatilization occurs, because it is uniform; the air supply and mixing capacity are always present. With hand firing, a large amount of fresh fuel is introduced at one time and if the temperature be very high, there is a rapid evolution of volatile gases, producing sudden masses of dense smoke which find exit at the chimney, because the air supply and mixing capacity are not adequate at the moment. Thus a moderate cooling of the furnace represses the volatilization and it is found that a harmonizing of the elements involved tends toward a better result. A frequent and uniform supply of coal by hand to a tile-roof furnace gives results like those obtained with the stoker.

The tile roof owes its origin more particularly to a necessity for reducing an excessive maintenance cost, a number of years ago in a large power plant in Chicago. The trouble was excessive burning out of the bottom row of tubes, which were exposed directly to the fire for their full length owing to being equipped with the T-tile. In seeking a remedy for the trouble it appeared probable that if the tubes were insulated and protected from the action of the fire they would not burn, so it was thought that by covering a considerable portion of the exposed surface, the water supply to the tube would be adequate for the remaining exposed portion. Therefore a firebrick encircling tile was applied, forming what is known as a tile roof, with the result that the burning of the tubes ceased entirely. The application of the roof, however, served to retain the heat in the furnace; in other words, it delayed the impingement of the gases upon the heating surfaces, during the brief period required for their passage under the roof. The result was that the actual temperature in the furnace was higher than it had been with the tubes of the boiler exposed and T-tiles used. This resulted in an increased wear upon the firebrick linings of the furnace so that the cost of maintenance of the brickwork was increased. This increase, however, was very much smaller than that made necessary by the replacement of burned tubes, so there was a net gain due to the tile roof. This has always been the experience with boilers worked to or above their rated capacities when properly equipped with a tile furnace

TABLE I. DISTRIBUTION OF HEAT ON BASIS OF COAL AS FIRED.

	Tile Roof.		Exposed Tubes.	
	B.t.u.	Per Cent.	B.t.u.	Per Cent.
Heat usefully employed in making steam	6,500.5	61.53	6,986.5	66.57
Loss in moist gases, due to				
Moisture in coal	150.8	1.42	154.9	1.47
Burning available hydrogen	309.1	2.93	294.7	2.81
Water of combination	94.5	0.90	90.0	0.86
Moisture in air	25.0	0.23	23.0	0.21
Loss in dry gases	2,009.4	19.02	1,530.6	14.59
Loss by undeveloped heat in refuse	661.0	6.26	303.0	2.88
Loss by sensible heat in refuse	32.0	0.30	25.1	0.24
Loss by radiation and heat unaccounted for	781.7	7.41	1,087.2	10.37
Total	10,564.0	100.00	10,495.0	100.00

roof. In this connection, however, it should be stated that considerable trouble has been experienced in the use of an improperly designed tile. The original one was called an "encircling tile" and its design was such that its thickness was sufficient to prevent it from breaking, and it wore away gradually during a long period of service. Owing to the success of the roof, the maker of the boiler adopted what was designated as a "C-tile" of quite a different design and of much less thickness where exposed to the fire than the encircling tile. The result has been in many cases, especially where the C-tile was made of poor material, that it broke and exposed a portion of a tube to the intense heat of the fire, with the result that the tube burned out. But in no case, where a properly designed tile of good material has been employed, has the cost of maintenance increased; but, on the contrary, it has been materially reduced.

Concerning the comparative behavior, as far as smoke is concerned, in the Chicago plant previously mentioned, which was equipped with chain-grate stokers under Heine boilers and fitted with T-tiles, exposing the lower half of the bottom row of tubes to the fire, there was a continuous and uniform discharge of rather dense yellowish-brown smoke, which, however, entirely disappeared when the tile roof was installed.

In bulletin No. 34, issued by the experimental station at the University of Illinois, which contains the report of a series of tests upon a Heine boiler with and without a tile furnace roof and equipped with a chain-grate stoker, it is intimated that a better condition of fire prevailed in the tests with the exposed tubes than with the tile roof. The report, however, states that this was due possibly to other causes. The coal used in the tests with the exposed tubes was of a more favorable size than that used with the tile roof, and for this reason the condition of the fire was better. As a matter of fact, the effect of the tile roof in producing better conditions of fire is quite marked, because the higher furnace temperature causes the coal to ignite faster and burn with greater freedom, overcoming in a large measure, the difficulty experienced in obtaining prompt ignition of the entering fuel with chain-grates or other stokers.

TABLE 2. DISTRIBUTION OF HEAT ON BASIS OF COAL AS BURNED.

	Tile Roof.		Exposed Tubes.	
	B.t.u.	Per Cent.	B.t.u.	Per Cent.
Heat usefully employed in making steam	6,499.5	65.63	7,011.0	68.63
Loss in moist gases, due to				
Moisture in coal	150.8	1.52	154.9	1.52
Burning available hydrogen	309.1	3.12	294.7	2.89
Water of combination	94.5	0.96	90.0	0.88
Moisture in air	25.0	0.26	24.0	0.23
Loss in dry gases	2,009.4	20.29	1,530.6	15.02
Loss by sensible heat in refuse	32.0	0.32	25.1	0.24
Loss by radiation and unaccounted for	781.7	7.90	1,061.7	10.59
Total	9,903.0	100.00	10,192.0	100.00

Regarding the question of relative heat efficiency it may be said, briefly, that the condition of combustion in the two series of experiments differed in respect to the air supply; in other words, the CO_2 was higher in the tests with the exposed tubes than with the tile roof. Therefore, it would necessarily follow that the efficiency secured would be influenced by this condition. However, the influence that this difference in CO_2 had on the result was not considered in the report.

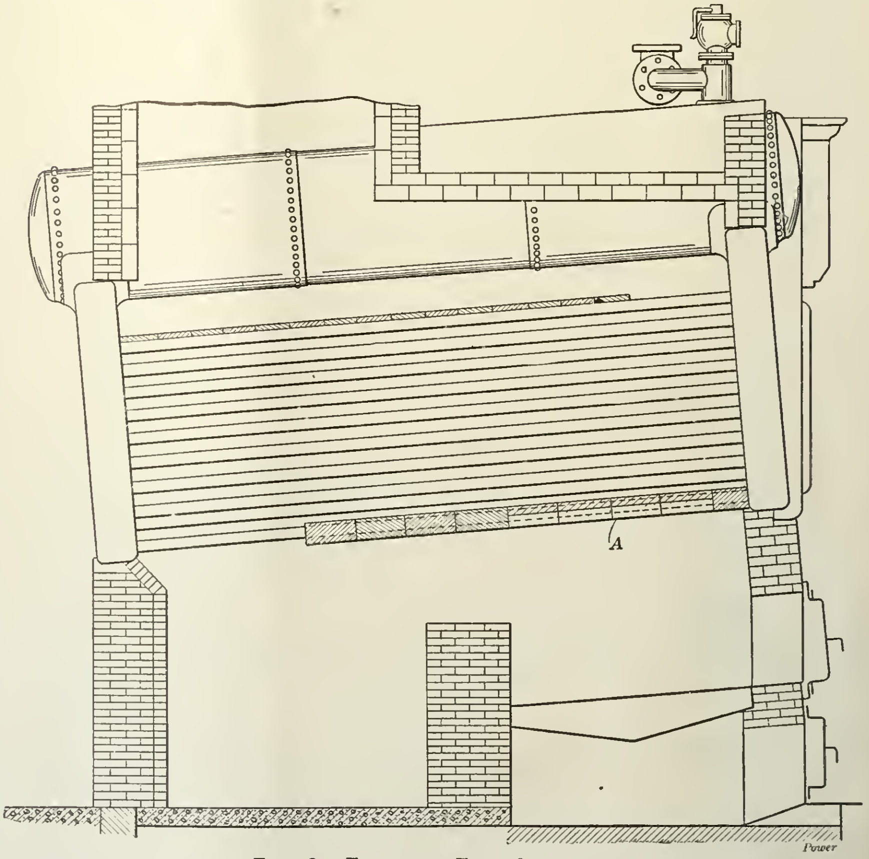


FIG. 2. T-BAFFLE TILE INSERTED

In an experiment of this kind, assuming that the same kind of coal is used in both series of tests, four elements could influence the result: First, the capacity at which the boilers operate; second, the quantity of air supply, that is, the percentage of CO_2, would have an important bearing on the heat loss in the escaping gases; third, the completeness of combustion; and fourth, the ability of the boiler to absorb heat passing through it. As the horsepower developed in the two series of tests was practically the same, the necessity for the first consideration is eliminated and the subject is confined to the matter of air supply, combustion and heat absorption. If the tests had been properly conducted, the same percentage of CO_2 would have been maintained in each series and in this way the actual relation between the two types would have been at once apparent; as the combined effect of the two elements involved, that of combustion and heat absorption, would have appeared. The condition of air supply being more favorable in one case, however, introduces an unfortunate complication and makes an accurate diagnosis of the performance difficult, but not impossible.

The first step in an analysis of the results requires that an account of the heat distribution be made, which, upon the basis of coal as fired, is shown in Table 1 and upon the basis of coal as burned, which is the significant one, is shown in Table 2. From this it appears that the loss due to radiation and heat not accounted for, with the tile roof is 7.90 per cent. and for the exposed tubes 10.59 per cent., a difference of 2.69 per cent., an amount which can be explained only upon the basis that it represents loss in the form of smoke; that is, more incomplete combustion occurring with the exposed tubes than with the tile roof. On the other hand, the loss in dry gases, which takes into consideration the question of air supply, is 15.02 per cent. with the exposed tubes and 20.29 per cent. with the tile roof, a difference of 5.27. Therefore, in the case of the exposed tubes, there was a considerable loss due to incomplete combustion and with the tile roof a still greater loss through heat being carried away by the hot gases. The greater loss in hot gases occurring with the tile roof was due to two causes; the first and most important being that with the exposed tubes the boiler absorbed heat much better than with the tile roof, for the reason that not only was there an exposure of a larger amount of heating surface, but its distribution was such that the gases were in contact with the bottom row of tubes the full length of the boiler before they began to act upon the main portion of the heating surface. In this respect, the plain boiler, from the standpoint of heat efficiency has a superiority over the boiler having a tile roof.

The other cause is found in the difference in air supply, or as might be expressed, in the percentage of CO_2. The temperature of combustion depends upon this, the temperature being higher when the percentage of CO_2 is higher, and conversely. Thus the plain boiler was supplied at a higher initial temperature, with the result that more heat necessarily passed into it than when equipped with the tile roof. It necessarily follows, that if more heat enters a boiler, there is less to go to the chimney. Hence, part of the loss in hot gases with the tile-roof boiler was due to its inferiority as a heat absorber and the remainder to the unfortunate fact that it was not served with the same initial temperature as the other. It is apparent from this that to arrive at a just and accurate comparison, it is necessary to make a correction in the loss in the gases escaping from the tile-roof boiler. This has been done, as shown in Table 3 by recalculating the loss in the dry gases, assuming that the percentage of CO_2 had been 7.5 instead of 6.8. The final temperature of 558 degrees from the tile roof was not obtained with conditions of combustion represented by 7.5 per cent. of CO_2, but by 6.8. It is evident that if the boiler had been served with the higher initial temperature, the final temperature would have been lower. How much lower it would have been is not known, but assuming that the difference of 90 degrees in the final temperatures is two-thirds due to the superiority of the boiler with the exposed tubes and one-third due to the inferiority of the initial temperature applied to the tile-roof boiler, the loss in the moist gases is reduced for the tile roof by one-third of their difference. The loss in dry gases has been calculated upon the basis of a final temperature of 558 degrees and, assuming 7.5 per cent. of CO_2, gives 18.79 per cent., which is 3.77 per cent. more than with the exposed tubes; one-third of this is 1.25, which subtracted from 18.79 leaves 17.54 per cent. as the loss in the hot gases which would have occurred if there had been 7.5 per cent. of CO_2 with the tile roof. Hence, Table 3 shows the true comparison of the two equipments. Upon the basis of the correction employed the difference in the temperatures of the escaping gases should have been 60 degrees instead of 90 and it is for this reason that the reduction for the loss in the moist gases is given in Table 3.

TABLE 3. DISPOSITION OF HEAT ON BASIS OF COAL BURNED, WITH AIR SUPPLY THE SAME IN BOTH CASES, GIVING 7.5 PER CENT. OF CO_2 FOR EACH.

	Percentages.	
	Tile Roof.	Exposed Tubes.
Heat usefully employed in making steam, as reported.	....	68.63
Heat usefully employed in making steam, by difference	68.90	....
Loss in moist gases, due to		
Moisture in coal..........	1.42	1.52
Burning available hydrogen	2.81	2.89
Water of combination.....	0.89	0.88
Moisture in air............	0.22	0.23
Loss in dry gases............	17.54	15.02
Loss by sensible heat in refuse.	0.32	0.24
Loss by incomplete combustion, by difference..........	....	2.69
Loss by incomplete combustion.....................	0.00	....
Loss by constant radiation..	7.90	7.90
Total...............	100.00	100.00

As there are no data making it possible to separate the radiation and incomplete-combustion losses, it is assumed that that unaccounted for with the tile roof is all lost by radiation. If this is true, it follows that the greater unaccounted for loss with the exposed tubes shows an incomplete-combustion loss of 2.69 per cent. If combustion is not complete with the tile roof, it necessarily follows that the radiation loss of 7.90 per cent. is too high. If so, it also follows that the incomplete-combustion loss with the exposed tubes is greater than that charged to it; also, if, as the bulletin states, "radiation is greater with the tile roof than with the plain boiler," the radiation, as given, is too high. Hence, the incomplete combustion with the exposed tubes must be greater than 2.69 per cent. So it is apparent that there are certain slight differences against the exposed tubes, which are not taken into account. To sum up briefly, the efficiency of the two types is practically the same, one losing about 2.5 per cent. more in smoke and the other an excess loss of 2.5 per cent. in hot gases. As far as heat efficiency is concerned there is no difference, but with the three other features under consideration, that of maintenance, smoke and the performance of the fire, the superiority lies with the tile roof.

TABLE 4. MISCELLANEOUS DATA.

ANALYSIS OF COAL ON PURE COAL BASIS.

	Per Cent.
Carbon..............................	81.411
Hydrogen............................	4.925
Oxygen..............................	9.217
Nitrogen............................	1.690
Sulphur.............................	2.757
Total	100.000

	Tile Rcof.	Exposed Tubes.
Temperature of escaping gases, degrees Fahrenheit.....	558	468
Excess in temperature of escaping gases, degrees Fahrenheit.................	90	......
CO_2, per cent. by volume....	6.8	7.5
Furnace temperature as measured, degrees Fahrenheit...	2066	1883
Excess furnace temperature, degrees Fahrenheit........	183	......
Temperature of combustion, degrees Fahrenheit........	1959	2131
Excess temperature of combustion, degrees Fahrenheit	......	172

Some explanation relative to the temperature of combustion should be made. Therefore, it may be said that the temperature as calculated is the one influencing the performance and not the actual temperature realized in the measurements by a pyrometer. Therefore, although the temperature measurements taken in the two furnaces are 2066 degrees for the tile roof and 1883 for the plain tubes, the temperature of combustion calculated according to the formula given on page 29 of "Kent's Steam Boiler Economy" is 1959 degrees for the tile roof and 2131 for the plain tubes. The reason for the actual measured temperature being so much lower with the exposed tubes, considering the higher temperature of combustion, is that a greater quantity of heat flows immediately from the furnace into the lower row of tubes than with the tile roof, where it accumulates and gives a higher reading of the pyrometer.

Table 4 presents certain data taken from the bulletin, which have a bearing upon the foregoing. However, as the coal analysis was not given and a heat balance could not be made without it, this data, on the basis of the composition of the pure coal, is given in the first part of the table, which is calculated from the ultimate analysis of Vermillion county, Illinois, coal, published in bulletin No. 31 of the engineering-experimental station. This is the coal reported to have been used in the tests.

They put in an alternater et th' nursin'-bottle factery over t' Jayville. Th' feller thet's tryin' t' make a noise like an ingineer writ t' th' factery thet furnished it an' told 'im it didn't work right; sed it run hotter 'n Texas, an' wudn't keery th' load. They writ back an' ast 'im ef th' power facter wuz all right. He writ 'em again an' told 'em thet he'd looked th' dumd masheen all over an' cudn't find anythin' thet looked like a power facter an' gessed thet they hed fergot t' send it out. Told 'em ef they'd send out one he'd put it on an' try it. Haw! Haw! Haw!

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Grounds and Overloads on Induction Motors

By R. H. Fenkhausen

Most induction motors of moderate size are supplied from two-phase or three-phase circuits of either 220 or 440 volts, and single-phase circuits of 220 volts. The 220-volt single-phase circuit is usually derived from the outer wires of a 110—220-volt three-wire system and therefore, as a rule, the neutral point is grounded. Polyphase circuits of 220 and 440 volts, however, are usually operated without any permanent ground on the system, not only because the underwriters object to a grounded power circuit, but also because better results are obtained when the system is completely insulated, owing to the decreased strain on the insulation of the motors. The majority of the induction motors now on the market are so well insulated as to be practically immune from grounds, but owing to the fact that most manufacturers use one motor for either 220 or 440 volts, the factor of safety of the insulation on a 220-volt machine is double that for 440 volts. It will therefore be found that 440-volt motors in exposed locations are a great deal more liable to insulation breakdowns than 220-volt machines. In order to permit one type of winding to be used on either voltage, all the coils of one polarity are connected in one group and all the coils of the opposite polarity in another group. For 220-volt service the two groups of coils are connected in parallel, and as this voltage is more widely used, most motors are connected in this way at the factories. In case an order for a 440-volt motor is received, it is only a few minutes' work to open the parallel connections between the two groups of coils and reconnect them in series.

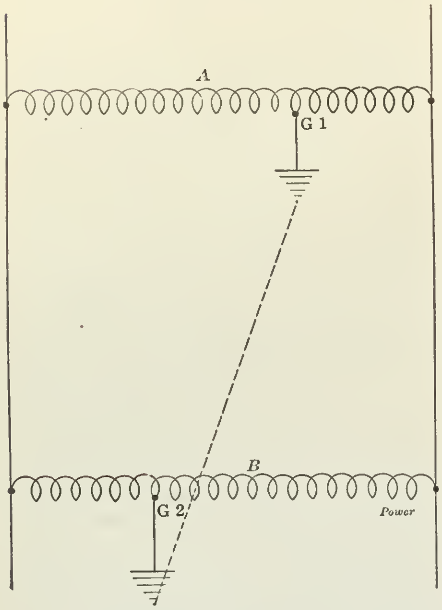


Fig. 1. Partial Short-circuit by Two Grounds

On an ungrounded system, one ground on a motor does not necessarily cause a shutdown; the motor may be kept in service temporarily. If, however, two grounds exist simultaneously either on the same or on different motors, it results in a partial short-circuit of the windings of the motor or motors. Suppose, for instance, that a ground $G1$ exists on the motor A, Fig. 1, and at the same time another ground $G2$ occurs on the motor B. It is evident that most of the current will flow through the earth, as indicated by the dotted line, from $G1$ to $G2$ and most of the windings of both motors will be heavily shunted by this path. The coils represented in heavy lines will receive the full-line voltage, and not having sufficient impedance, will allow excessive current to flow, in all probability burning out themselves and by conduction some of the adjacent coils. It therefore follows that if a ground occurs on a motor it should be located as soon as possible, and for this reason a suitable ground detector should be installed in a prominent place to indicate as soon as a ground occurs.

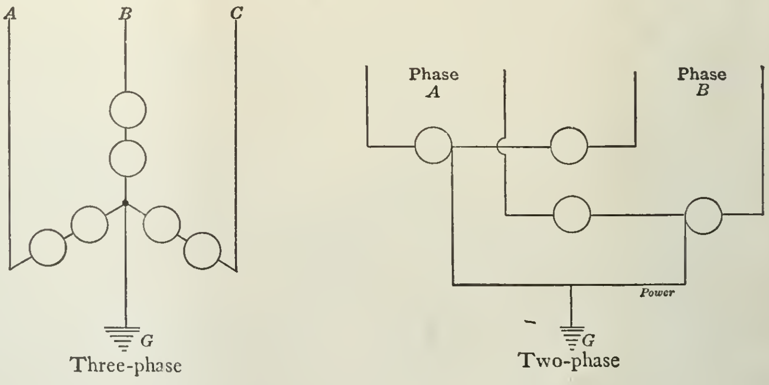


Fig. 2. Ground Detector Lamp Connections

Fig. 2 illustrates simple ground detectors for two-phase and three-phase motor circuits which give satisfactory indications and require only a few lamps and sockets for their construction. Ordinarily all lamps will burn dimly and at equal brightness, but as soon as a ground occurs one set of lamps will burn more dimly and the others will brighten up. A dead ground, such as would be liable to occur on a line wire will extinguish the lamps connected to it, and raise the others to full candlepower. A ground on a motor, however, usually indicates by a partial unbalance of the lamp banks, on account of the autotransformer action of the motor winding; this is illustrated in Fig. 3. With a little practice, however, it is possible to approximately locate the group of coils in which the ground exists, by merely watching the ground-detector lamps.

In the plant under my charge there are several very heavily loaded 440-volt two-phase three-wire feeders, each having a large number of motors connected. Upon the appearance of a ground, steps are immediately taken to locate it. Each feeder is momentarily "killed" at the switchboard, until the grounded one is indicated by the detector. In case it proves to be one of the heavily loaded circuits where the large number of motors would render its location a difficult and tedious job, nothing is done until the noon hour, when all motors are shut down. If the ground still exists with all the motors shut down, all the lines are looked over and if no ground is found it is assumed that the ground is on a small line, so it is burned off by grounding the opposite phase through a small fuse and oil switch. If the ground blows a 50-ampere fuse, no further effort is made to burn it off and a careful search is made. It usually happens that the ground disappears when the motors are shut down, so the detector lamps are closely watched at starting time, and by use of the following indications an approximate location of the faulty motor is made.

All motors above 5 horsepower are equipped with autostarters, some of which have one starting point and some two. Motors of 5 horsepower and under are thrown directly on line.

The permanent appearance of the ground indicates a motor of 5 horsepower or under.

The appearance of the ground followed by one or two momentary disappearances indicates a motor of over 5 horsepower, as the ground will clear when the controller is moved from step to step, and the number of disappearances indicates whether a two- or three-point starter is on the faulty motor.

A ground indication shifting from phase to phase indicates a reversible motor.

By applying these "propositions" it is possible to locate the grounded motor as one of a possible four or six. One man then remains at the switchboard and watches, while a second goes to each of the suspected motors and flashes its shop number by means of the main switch. As soon as the number of the grounded motor is flashed it will be identified by the switchboard operator by the flashing of the ground lamps. This method of locating grounds may seem lengthy, but it really takes far less time than is necessary to describe it, and is far preferable to making tests of each individual motor, because it does not interrupt the service seriously and can be done to a great extent right at the main switchboard.

Prevention of Overloads

Squirrel-cage induction motors will occasionally fail to start a load having high static friction, but will carry the same load with ease once it is started. This limited starting torque is an inherent characteristic of all motors of the squirrel-cage type. It is poor policy to install a large motor to take care of the starting duty, because it means a large investment, poor efficiency and, worse still, a low power factor, which is a great disadvantage, especially if the motors are supplied with current from an isolated generating plant. If the load cannot be reduced during the starting period, or part removed by means of a friction clutch, a wound-rotor motor should be substituted. A slight increase in starting torque may often be obtained by slotting the end rings between bars, or by boring them out in a lathe to reduce the cross-section; but this should be attempted only by a competent electrical engineer because, as pointed out in a previous article, additional starting torque obtained in this way is at the price of poorer efficiency, power factor and speed regulation when running at full speed.

To protect motors from overloads, reliance must be placed on fuses and circuit-breakers, but to obtain adequate protection either two sets of fuses must be used, as pointed out in a previous article, or else the fuses must be so connected that they will be short-circuited while the motor is coming up to speed.

The general tendency in all progressive plants is to install new machinery, and as this is usually belted to existing line shafting, the only safe method of preventing overloads is to take periodical readings with a wattmeter in order that any dangerous overload on a motor will be detected. On two-phase three-wire systems the fuse in the common return wire must be 1.414 times as large as those in the separate-phase wires, and if the fuses are removed by a machine operator there is great liability of their being replaced incorrectly, which would leave too large a fuse in one main wire and too small a fuse in the common return. When the motor is started there is liability that the common return fuse will blow just before the motor comes up to full speed. It will, however, have enough speed to fall into step as a sort of single-phase motor with both phases in series across the outside wires, and is sure to overheat. It is my practice to use fuses 25 per cent. larger than necessary in the common return, which renders it less liable to blow, and still leaves both phases of the motor adequately protected by the other two fuses. To prevent starting and running fuses from being interchanged and also to stop operators from mixing the center and outside fuses, the control panel for each motor is arranged, as far as possible, so that each fuse will fit only its own block. The National Code rating for fuses makes this quite easy. Take for instance a motor requiring 30- and 50-ampere fuses for running and 50 and 80 for starting. The 30-, 50- and 80-ampere fuses fit each a different block and cannot be interchanged. If, however, 35- and 50-ampere fuses had been selected, both sizes would fit one block and an error could easily be made.

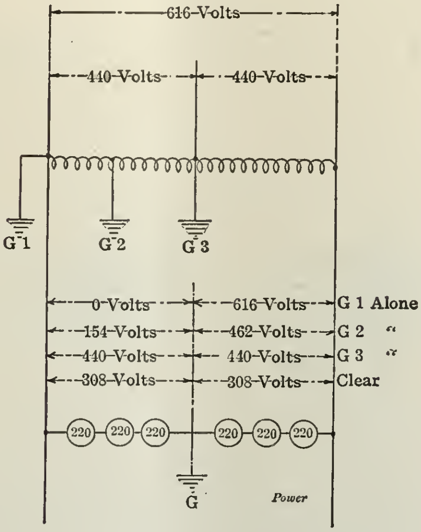


Fig. 3. Effects of Different Grounds

In some cases it is not possible to select fuses that will require different blocks and in such cases it will be found advisable to purchase each capacity of fuse with a different-colored fiber casing, so that any wrong arrangement of fuses can be detected from a distance. This color scheme is also valuable in identifying fuses the label of which has become illegible or been removed. Some fuse makers stamp the ampere capacity on the metal ferrule but they are few in number. Fuses may be procured in red, black and buff fiber casings from different makers, each of whom seems to prefer a certain color. This gives three capacities for each size of block and should prove sufficient for all practical purposes.

Different makes of fuses are pretty uniform, especially as regards the indicators, which usually indicate about once in a hundred times. I have yet to see a cartridge fuse that can be relied on to show the condition of the fuse without testing; most fuses blow without indicating, but some indicate that they are blown while still in good condition. This is probably due to the indicating wire being of a lower resistance than the main fuse wire and therefore carrying more current per circular mil at full load than the main fuse in parallel with it.

Most types of autotransformer starters do not give overload and no-voltage protection to a motor, and auxiliary apparatus has to be installed for this purpose, as explained in my article on motor starters in the November 1 issue.

A Most Puzzling Condition

By C. L. Greer

Recently in our plant there arose one of the most puzzling situations due to trying to feed separate railway circuits by different rotary converters all of which were supplied in parallel on the alternating-current side without the interposition of transformers between the busbars and the converters.

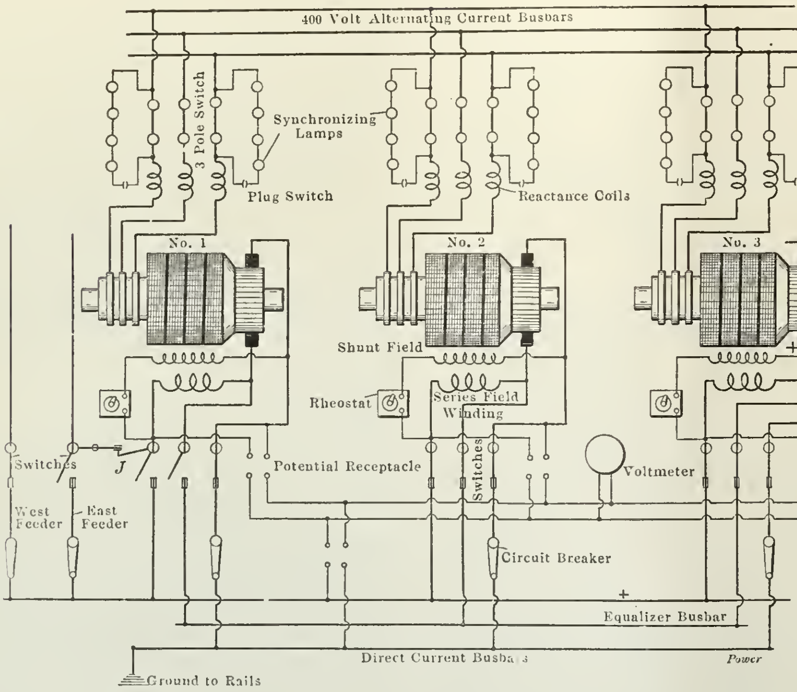
A large converter had just been installed to take the place of a smaller machine which had been taken out. The large machine had been connected up but would not properly divide the load with the two smaller machines. While waiting for the proper adjustments to be made, a one-unit substation nine miles east of the plant became disabled, which considerably increased the load on the two small converters in the plant.

Since the large converter would not

work in parallel with the smaller ones without taking all of the load, we tried to utilize it by running it on the east-end feeder alone, pulling the west end with the two smaller machines.

The accompanying diagram shows the connections and how it was intended to adapt the large converter (No. 1) to the east-end load. As shown at *J*, a temporary connection was made from the lower stud of the positive main switch to the line side of the east-end feeder switch, there being a switch included in the connection *J* to disconnect the converter from the feeder.

The feeder switch was left open to separate the feeder from the positive busbar and the positive switch of the converter (also the equalizer switch) was also left open to keep No. 1 from being connected in parallel with the other two machines. The negative switch and circuit-breaker were closed to complete the return circuit from the track rails. No. 2 and No. 3 were run in parallel as usual, supplying the west-end feeder from the main busbars. All of these conditions are shown in the diagram of connections. (No ammeters are shown.)



CONNECTIONS OF MR. GREER'S ROTARY CONVERTERS

Nos. 2 and 3 were in operation when No. 1 was thrown on the east end and it was then that the puzzling phenomena occurred. Nos. 2 and 3 immediately dropped their load and their direct-current ammeters flew back against the pegs, indicating that current was backing in—that No. 1 was sending current back through the other two machines. The load on No. 1 picked up to an amount far in excess of the east-end load, indicating that No. 1 was carrying the load formerly taken by Nos. 2 and 3.

The busbar voltage was that of No. 1, but dropped to that of the distant west-end substation when Nos. 2 and 3 were disconnected from the busbars, thereby showing that there was no leak anywhere between the east and west feeders. Moreover, the load on No. 1 dropped each time Nos. 2 and 3 were cut out on the direct-current side.

Since there was no connection between the two feeders, why did the machines behave in the manner described?

Some Additional Points on Installing Induction Motors

By George Treater

In the article on "Installing Induction Motors," by R. H. Fenkhausen, in the issue of August 23, the matter of current-carrying capacity of motor-supply conductors appears to have been ignored. The selection of wire size is discussed solely on a voltage-drop basis. Voltage drop is the determining factor when a motor circuit is relatively long; but, in a great majority of cases, motor circuits are short and the currents carried by them determine what sizes of wire should be used. In selecting the size of wire to use for connecting an electric motor to the supply main or transformer, both the voltage-drop and the current-carrying capacity of the wire should be considered. If the voltage drop in the wires is excessive, there will be an undue loss of energy, and the voltage at the receiving or motor end of the circuit may be so low that the motor will not develop sufficient torque to do its work. Hence the voltage drop should be considered most carefully where long circuits are concerned.

Where circuits are short, however, it is usually true that if the conductor were made of a size that would give the allowable voltage drop, the wire would be so small that it would be overheated by the full-load current of the motor. Even if the heating were not severe enough to damage the wire, it might easily damage the insulation and set fire to any inflammable substance against or near the wire. Rubber insulation, such as that used on most indoor electrical wiring, deteriorates very rapidly when subjected to only moderately high temperatures. Consequently the current density in conductors insulated with rubber must be kept lower than when the wire is covered with some other insulation.

TABLE OF CARRYING CAPACITIES OF INSULATED WIRES.

B. & S. Gage.	Amperes.		Circular Mils.
	Rubber Insulation.	Other Insulation.	
18	3	5	1,624
16	6	8	2,583
14	12	16	4,107
12	17	23	6,530
10	24	32	10,380
8	33	46	16,510
6	46	65	26,250
5	54	77	33,100
4	65	92	41,740
3	76	110	52,630
2	90	131	66,370
1	107	156	83,690
0	127	185	105,500
00	150	220	133,100
000	177	262	167,800
000	210	312	211,600
Circular Mils.			
200,000	200	300	.....
300,000	270	400	.....
400,000	330	500	.....
500,000	390	590	.....
600,000	450	680	.....
700,000	500	760	.....
800,000	550	840	.....
900,000	600	920	.....
1,000,000	650	1000	.....
1,100,000	690	1080	.....
1,200,000	730	1150	.....
1,300,000	770	1220	.....
1,400,000	810	1290	.....
1,500,000	850	1360	.....
1,600,000	890	1430	.....
1,700,000	930	1490	.....
1,800,000	970	1550	.....
1,900,000	1010	1610	.....
2,000,000	1050	1670	.....

It is possible to calculate from certain formulas (which take into consideration the radiating surface of the wire, its insulation, the material of the wire and the nature of its surroundings) what the temperature rise in a given conductor will be with a given current flowing, but such calculations are not ordinarily necessary in practice. Tables have been compiled by the insurance authorities giving safe current-carrying capacities, and they are nearly always used. The official table of the National Board of Fire Underwriters is reprinted herewith. This table can be used directly in selecting wire sizes of proper carrying capacity for electric-lighting circuits and for circuits not subject to overloads. But for motor circuits the following supplementary rule should be observed:

The motor leads or individual circuit must be designed to carry a current at least 25 per cent. greater than that for which the motor is rated. Where the wires under this rule would be overfused in order to provide for the starting current, as in the case of many of the alternating-current motors, the wires must be of such size as to be properly protected by these larger fuses.

The diagram illustrates the reason why the current-carrying capacity of conductors should be considered in wiring motors. At *A* is represented a motor circuit presenting the conditions given by Mr. Fenkhausen in the example at the lower left-hand corner of page 1515. The current times distance is

$$150 \times 50 = 7500$$

and this value at a 3 per cent. drop corresponds in the table on page 1514 to a No. 6 wire. Then, using, as is recommended, the next size larger wire, No. 5 is selected. This wire, from the Underwriters' table, has a capacity of 54 amperes with rubber insulation and 77 amperes with other insulations. Rubber-covered wire is always used in conduit or molding and the other insulations, "slow-burning" and "slow-burning weatherproof," are run "open." No. 6 wire is amply large to safely carry the normal current of 33 amperes. It is also large enough to meet the insurance requirement of a 25 per cent. overload allowance which would require a wire good for

$$33.2 \times 1.25 = 41.5 \text{ amperes.}$$

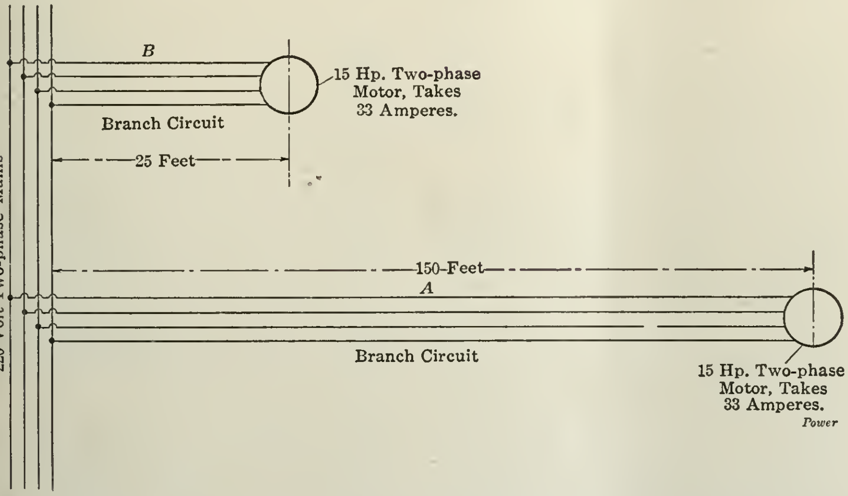

B
15 Hp. Two-phase Motor, Takes 33 Amperes.
Branch Circuit
25 Feet
220 Volt Two-phase Mains
150-Feet
A
Branch Circuit
15 Hp. Two-phase Motor, Takes 33 Amperes.
Power

Diagram Illustrating Distinction between Heating and Drop Considerations

But now refer to the branch at *B* in the diagram, where the motor circuit is but 25 feet long. This is representative of many industrial-plant motor installations. Here the distance times current is equal to

$$25 \times 50 = 1250.$$

For a 3 per cent. drop, No. 12 would be the wire size, and following Mr. Fenkhausen's recommendation, No. 10 wire, the next largest size, would be selected for the installation. This has a "rubber-insulation" capacity of 24 amperes and an "other-insulation" capacity of 32 amperes, but the normal current of the motor is 33 amperes and the 25 per cent. excess carries the required capacity up to 41¼ amperes, so that No. 10 wire would be too small from the insurance standpoint; No. 6 would have to be used.

A plan that can be followed to advantage is to select the smallest size of wire allowed by the Underwriters' Code and find out whether the drop with that size would be greater than is desirable; if not, use it, but if the drop is excessive, then figure the size of wire on the basis of voltage drop.

LETTERS

Undercutting Commutator Mica

Up to the present time I have been successful in treating commutators that sparked as a result of high mica by cutting it below the copper. My former experience was on electrical coal-cutting machinery. It is well known that electrical coal-cutting machines are given the most severe treatment, one feature of which is the saturation of every part about the motor with oil and fine coal dust; the dust and oil, however, did not cause any trouble in short-circuiting the commutator bars. With this experience in mind I cut down the mica in a motor now under my charge, but in this case the experiment was a failure; the commutator sparks badly.

In the same room with the motor are several pipe-threading machines on which large quantities of oil are used to keep down the temperature of the pipe and cutters while threading. The heat of cutting vaporizes the oil and it settles on the surrounding parts. The commutator of the motor condenses the vaporized oil that falls on it and the oil collects a dust full of sharp molding sand and fine iron parts which rises from half a dozen tumbling mills in the same room. This dust accumulates in the grooves between the commutator bars, and in a short time causes rings of fire to surround the commutator.

By means of a little gasolene I cleaned the commutator and the sparking ceased, but it was only a short time until the crevices were again filled and sparking was renewed. The mica in places showed signs of burning so I felt it would be safer to cut down the commutator flush with the mica, and it is now running more satisfactorily.

J. L. Hart.

Chicago, Ill.

Utilizing the Good Coils of a Damaged Transformer

We are supplied with power over a three-phase 6600-volt transmission line through step-down transformers lowering the pressure to 440 volts for power and through two transformers to 110 volts for lighting, the lighting transformers taking the full 6600 volts at the primary terminals. Both of the lighting transformers burned out, and as lights were absolutely necessary something had to be done, repair parts not being obtainable up here on short notice.

One of the transformers was taken down, when it was seen to have four secondary coils. The damaged primary coils were removed and the secondary coils, being found undamaged, were put back on the core and connected in series across one phase of the 440-volt power circuit as an autotransformer. Then by taking a circuit from the terminals of each coil four 110-volt circuits were obtained, each good for nearly one-fourth the rated capacity of the original transformer.

J. M. Wauchope.

Ketchikan, Alaska.

When condensing engines are liable to be lightly loaded for any length of time, it is especially desirable to provide a safety device that will break the vacuum when the water rises to a certain hight in the condenser. If this is not done, the condenser may be flooded and the engine wrecked, should the speed of the air pump slacken enough to allow the water to back up in the cylinder.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Some Hindrances of Producer Gas Power

By Charles S. Palmer

Some years ago, Power saw the enormous opportunity of producer-gas power, and has constantly indicated the opportunity by the distinct recognition of gas producers and gas engines as actual factors in the industrial world. The history, the present status and the possible future of gas power have been described, thoroughly and continually. The public have been informed, in a sane and reliable way, of the great installations and of the smaller ones, as well as the developments in producer and engine construction. Full information has not been lacking; and some parts of the country have been quick to appropriate and use this newer and more economical form of power. But, taken as a whole, some parts of the manufacturing world, and New England in particular, have been slow—perhaps foolishly slow—in taking producer-gas power for what it is worth. Perhaps there is a reason for this seeming paradox.

The condition of New England as a manufacturing center is unusual. She has the inventive ability, the traditions, and the plants for making goods of almost every description; but she is dependent on coal for most of her power and she has no native coal mines. It would seem axiomatic that a factory section so situated would be the first to exploit and develop such a tempting chance as that offered by producer-gas power. Such, however, has not been the history of recent years. To be sure, some few progressive or far-sighted manufacturers have taken hold of this means to power economy; some few more have experimented with plants of various types; but, on the whole, the inability of the carefully rated plant to meet the demands of enormous emergency overloads has contributed to keep back the progress which might have been anticipated. Moreover, the remarkable efficiency of some steam plants comprising large units has kept many smaller manufacturers from seeing that *no small steam plant can compare with the small gas-power plant in economy*. This much seems to be certain; namely, that the producer-gas plant is desirable for establishments of small to medium requirements, say from 30 to 200 or 300 horsepower (perhaps larger) in its decided economy over steam. But even granting this there seems to be some other reason why we see only a few scattered illustrations of the new source of power. What is it? It must be something more than the partly fancied and partly real slowness of the East. All the manufacturers in New England are not asleep, nor timid, nor foolish. Perhaps the stumbling block is the practical question of reliability. Is the producer-gas power plant really as reliable as the good old wasteful steam plant? And, if not, why?

Pat and Mike down in the boiler and engine room do not know it, but they are walking encyclopedias of practical information—practical information only—on the use of steam. There is a century of practice behind these sooty and hard-headed firemen and engineers, while there is only a quarter of that behind the untried and unskilled producer-gas operator. If anything happens to the boiler or engine, Pat goes around the corner and talks it over with Mike, if he cannot fix it himself; at any rate, between them they get over or around or through the trouble; the wheels go round, and the boss never knows anything about the hitch. Bless you, the boss supposes that the boiler and engine never have any troubles.

It is an entirely different story with the producer and gas engine. There is so much of practical chemistry and physics, and mechanics, and electricity, and what not, that if Pat or Mike is left in charge of this new creature that eats only a quarter of the food it ought to, and sometimes runs quietly, sometimes snorts loudly, and sometimes lies down and stops short—there is so much that is new and strange that when troubles come, and they do come everywhere, then Pat and Mike both sigh for the man who installed the blamed old thing, and lie down and whine that the thing is no good. The superintendent hears of this, but it never seems to occur to him that the real trouble is that he needs to give Pat and Mike several years of experience with the producer and gas engine before setting them to the task of taking care of them. Men have learned to run steam plants and men can learn to run gas-power plants; the one is just as reliable as the other, but both must be understood. The point, then, in adapting the small-sized plant to the requirements of the small manufacturer is to get a supply of competent firemen and engineers.

The writer has had some hard and practical experience along this line. Several years ago, in company with two or three friends, he tried to work out a good clean campaign of producer-gas power for New England. Between us we had the physics, the chemistry, the mechanics and the business ability requisite for the undertaking. We picked out what was as good a type of plant as could be found; and we had no prejudice in the matter, for one of the firm wanted the best obtainable plant, not only as a demonstration plant but also for service in his own factory (where it is still saving him thousands). We could sell plants to small manufacturers—a few of them; we could install them, we could run them; but as soon as our backs were turned and the local engineer tried to shoulder the responsibility for keeping the plant running, then troubles began. Reliability was largely an illusion. Why? Because, it takes months and years to get familiar with the tricks of the producer and engine; and no one can guarantee reliability unless he knows in advance of most of the impending troubles.

But experience is growing, and the troubles peculiar to gas power are coming to the light. When a trouble is known, either in steam or gas machinery, it is a comparatively simply matter to adjust things. That has taken some time; but the manufacturers have accumulated a fund of practical information which is almost as complete and certain as that relating to steam practice. The only trouble is to get this into the heads and hands of the coming crowd of practical operators of the producer and gas engine. Now who is to do this? The schools, the local owners of the power plants, or the builders of the producers and engines? Probably each will have his part to do in ironing out theory into reliable practice; but if a manufacturer wants to use this means of conserving the coal pile, yet hardly dares to intrust it to any of his engineers or firemen, why not consider this plan? Why not take some young and promising mechanics and tell them to go right out into some good factory where good types of gas producer and engine are built, guaranteeing them their wages in whole or part, as long as they need to fit themselves for practical operators, under an agreement to come back at the end of their apprenticeship and work a certain term of years? At least that would be one way. The only way to learn anything is to try to *do* it. Two or three years of shop experience, with good evening study, will put the average mechanic where he can be sure that he can "deliver the goods" in handling gas producers and engines.

This lack of experienced men seems to me to be one of the greatest of the hindrances to the rapid adoption of gas power. How can we expect to see the common use of producer-gas power until we have an abundant supply of producer-gas engineers? And, how can anything hinder the universal adoption of this source of great economy as soon as we shall be provided with the indispensable supply of competent gas-power engineers? Is it anything more than a matter of practical systematic education?

Breakdown Statistics in England

The following data relating to engine breakdowns are extracted from the annual report of Michael Longridge, the chief engineer of the British Engine, Boiler and Electrical Insurance Company, Ltd.

The rate of breakdown during 1909 was, among steam engines and turbines, 1 in 10.2 of the machines insured, and among gas and oil engines, 1 in 10.4, slightly lower in both cases than in 1908, but slightly higher than in 1907.

The causes of these breakdowns may be roughly classified as follows: Accidents and causes unascertained, steam engines, 27 per cent.; gas and oil engines, 34 per cent.; old defects or deterioration by wear and tear, steam, 41; gas, 19; weakness, bad design, workmanship or material, steam, 18; gas, 23; negligence of owners or attendants, steam, 14; gas, 24.

GAS AND OIL ENGINES.

Description of parts which are believed to have given way first.	Average six years to 1908.	During 1908.	During 1909.
	Per cent.	Per cent.	Per cent.
Valves and valve gears	34.3	31.3	29.6
Cylinders and cylinder ends	16.1	19.4	15.4
Pistons	10.2	4.8	2.7
Connecting rods and their bolts	10.7	10.8	9.4
Main shafts	5.0	9.1	13.4
Governors and governor gear	4.9	4.8	5.4
Silencing boxes and exhaust pipes	3.9	2.4	1.3
Flywheels and driving drums	4.4	5.4	0.7
Clutches and couplings	2.9	0.0	0.0
Frames and pedestals	3.0	1 2	1.3
Bolts*	2.4	2.4	2.0
Total wrecks, cause not ascertained	0.8	0.0	0.0
Main driving ropes or belts	0.0	0.0	0.0
Miscellaneous	0.9	5.4	7.4
Gas-producer plants	0.5	3.0	11.4
	100.0	100.0	100.0

*These bolts do not include bolts in connecting rods, valve gear and other moving parts, the bolts in these parts being included with the parts themselves.

The percentage in analysis for 1909 supports the view that technical education and foreign competition are influencing British methods of design. The decline in the number of casualties resulting from neglect is also satisfactory.

Turning to the figures for gas engines, the large percentage of breakdowns from unknown causes is unsatisfactory, but difficult to alter, because so many of the gas engines insured are small, and on the occurrence of a breakdown are either repaired or dismantled for repair before the inspector can examine them. The cases of weakness and bad design are also rather numerous. The weakest parts are undoubtedly the crank shafts; as many as 13 per cent. of the breakdowns for the year were breakages of shafts. That about one-fourth of the breakdowns resulted from neglect, arises from the practice of leaving small gas engines without skilled attendants, or without attendance altogether.

CORRESPONDENCE

"Restoring" a Piston Ring Groove

Some time ago I was called in to look over a 2½-horsepower two-stroke-cycle gasolene engine, which a friend had in his boat. The engine had been running very badly ever since it was put in the boat (it was bought second-hand). Upon taking the engine apart to give it a thorough cleaning, the owner had found that the piston had only two rings at the working end and none at the other end (see Fig. 1) instead of three, as it originally had (Fig. 2); the extreme front end *A*, Fig. 2, had been broken off. As the two rings which were there were somewhat worn, the owner had three new ones made, one for a spare. Two new rings were put on and the engine set up again, but it could not be made to run satisfactorily. This was the state of affairs when I first examined the engine, and I found that the engine had very little compression, in spite of the two new rings that had just been put on the piston, so I suggested that a third ring be put on.

The problem was how to put it on. We were out in the country where the only tools at our disposal were a blacksmith's outfit. However, we filed the outside edge *B*, Fig. 1, true around the circumference and made four little iron brackets shaped like an L and riveted them on as shown in Fig. 3, spacing the brackets 90 degrees apart around the circle. The rivets were countersunk on the outside and the third ring sprung over the ends of the brackets, on the neck of the piston. When the engine was assembled again it was made to run pretty well, after a few trials.

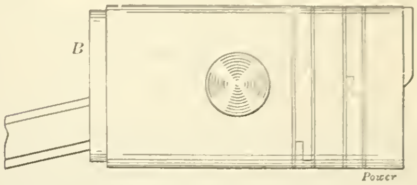


FIG. 1. PISTON AS TAKEN OUT OF CYLINDER

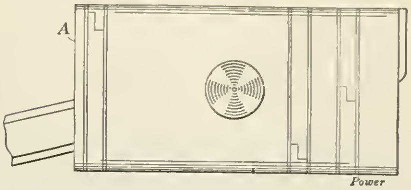


FIG. 2. PISTON AS ORIGINALLY BUILT

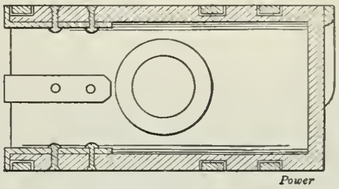


FIG. 3. THIRD RING AND HOLDING LUGS

J. E. Poché.

Plaquemine, La.

Another Diesel Manufacturer

Willans & Robinson, of Rugby, England, who have for the past five years manufactured Diesel engines for the Diesel Engine Company, of London, have made arrangements for placing the engine on the market themselves, and for dealing direct with customers, both in Great Britain and for export. They are manufacturing the engines in sizes ranging from 120 up to 600 brake horsepower.—*Gas and Oil Power.*

Ike Jinks, th' ingineer at th' hobble-skirt factery, wanted t' larn t' run gas'line ingins so he bot a '98 model ottermobeel t' practice on. It started fer th' fence with 'im an' Ike set in it an' hollered "Gee," "Gee," but th' dumd critter wud'nt "Gee" wuth a cuss. Th' last thing Ike sed 'fore he hit th' fence wuz "Gee-sus," an' he didn't say nuthin' more 'till th' docter hed woke him up.

Readers with Something to Say

Washing Out Boilers

In a small lighting plant where all the boilers are in service at once, it is often hard to get time in which to clean them. A boiler that is blown down and cooled as soon as the pressure is off and then washed out with cold well water is often badly strained and is apt to leak in a very short time.

To overcome this I placed a live-steam coil with an open end in the storage tank. As soon as a boiler is cut out for cleaning, I first draw the fire and then turn the steam from the boiler into the water in the storage tank. This serves the double purpose of reducing the steam pressure quickly and of heating the water in the tank as hot as it can be handled conveniently. Washing the boiler with this hot water does not set up any severe strains, and when the boiler is refilled with the hot water it takes only half the time to get the boiler under steam again.

CHARLES FENWICK.

Qu' Appelle, Can.

The Power Plant Owner

The engine was an old-time Corliss, 12x36 inches in size, installed in 1884. It had never been indicated and had always been run by starter-and-stoppers. I ran the plant for three days after I came to take charge and then told the owner that the engine was using too much steam. I said that I could set the valves so that 1000 pounds less coal would be consumed in the 11-hour run. He laughed at me and I told him to come the next morning and we would weigh the coal and he could fire the boilers, for he had run the plant occasionally. He came the next morning and fired and we weighed the coal. He burned 4090 pounds during the 11-hour run. I set the valves after I had indicated the engine. The next day, the engine handled the load so easily that the miller put on a feed of 1½ bushels more wheat per hour. When the day's run was over, I had used 2880 pounds of coal. This was a saving of 1210 pounds. The coal cost $4.40 a ton or 22 cents a hundred. This meant a saving of $2.66 each day. This man was paying me $2.50 per day. By virtue of the saving I was working for nothing and giving him 16 cents a day.

I asked for a raise and did not get it. I left. Since then this man has been after me to go back but I have another position now.

All engineers look alike to some power-plant owners.

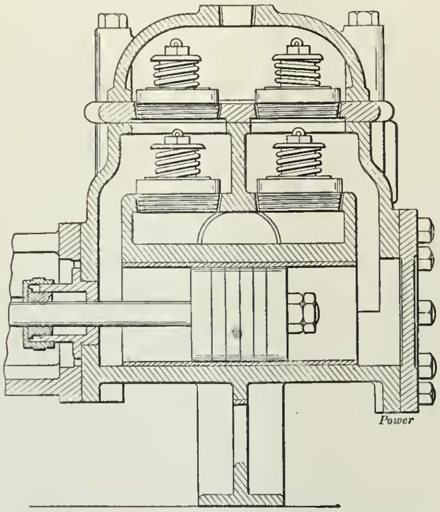
C. D. COOK.

Midford, Okla.

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Operation of Steam Pumps

There are many different styles of pumps in use but the most common are the piston pump and the plunger and ring pump. The clearance in the former is small, but the passages are indirect and cause more or less friction at high speeds. In the latter type the water passages are direct, owing to the suction valves usually being below, and the discharge valves above the piston, thus reducing the friction to a minimum.



SECTION THROUGH PISTON PUMP

One of the most important operations in repairing a piston pump is packing the piston and rods. In this a good engineer can save many a dollar for his employer, and considerable work for himself by being careful. The rings to be right should be cut about ⅛ inch too short to form a complete circle, which allows room for expansion when the packing has become wet. If the pump is fitted with a device for expanding the rings, care should be taken to avoid making them too tight. They should be expanded a little at first and then given a trial after which they may be expanded more if necessary. It requires the closest attention to obtain the best economy in packing the rods. Many engineers take out all the packing in the stuffing box when the rods leak and replace it with new packing, whereas possibly an additional ring is all that is required. Again, there are other engineers who add an extra ring without examining those already in the box, endeavoring only to stop the leak, and never giving a thought to the expense of replacing the rods at an earlier date than would be necessary if the rod packing were given the proper attention, as nothing scores the rods more than hard packing.

Trouble with a pump is invariably found in the water end; therefore it is a good idea to have a petcock tapped into the discharge pipe, and by opening this when the pump does not work properly it is easy to ascertain where and what the trouble is. If air blows steadily from the petcock, the trouble is usually with the suction pipe or supply, as the pump is drawing air from some source. If the air blows out on one stroke, and is drawn in on the next, it is well to look for an obstruction under the discharge valves; for, as the piston makes its stroke, instead of forming a suction in the suction pipe by raising the suction valves, it will draw air through the partly raised valve by way of the discharge pipe and force it out on the next stroke. If no air comes out of the petcock and still no water is being handled, the trouble may be one of two things; either the pump requires packing or something may be under the suction valves. Sometimes the air will come out of the petcock in jerks, which signifies that one side only is working, and, as a general thing, it will be found that the fault is with the packing.

In the case of a pump, similar to that shown in the accompanying sketch, failing to discharge sufficient water, when everything else seems to be in first-class condition, it is well to look at the gasket between the suction and the discharge chambers. Sometimes this gasket breaks, and, if such is the case, the water being discharged by one side of the pump will be drawn into the cylinder of the other side, and on the return stroke the piston will force the greater part of the water back to where it started. This continual churning will greatly reduce the quantity of water discharged. The engineer should be on the alert to detect this trouble as quickly as possible, for it will not take long for the water to wear a groove in the casting where the gasket is broken; if it does the pump will be a constant source of trouble as the gasket will break at this low point quite often. If trouble of this sort is experienced it will help matters considerably to put a small piece of sheet rubber, beveled at each end, under the gasket to fill in the groove, and use thick sheet packing for the gasket.

Walter C. Edge.

Philadelphia, Penn.

Experience of an Indicator Man

A company doing a large manufacturing business has an engine that is 100 per cent. overloaded. One day this engine refused to carry the load, so part of the plant had to be shut down at a heavy loss to the company. The engineer was asked to locate the trouble and, if possible, to fix it. He changed nearly all the valve rods, moved the eccentric, removed the valve bonnets and claimed to have set the valves, but everything went from bad to worse.

The manager finally decided to send for an indicator man who lived in a city some 200 miles distant. When the latter arrived and took a diagram he found that the mean effective pressure was 68 pounds on the head end and only 15 pounds on the crank end, and that the eccentric was set 1½ inches too far ahead. He made the necessary corrections and the engine ran all right, for which the company had to pay him $25 and expenses. Everything went along smoothly for about ten days when the engine again refused to carry the load and everyone was at a loss to account for the trouble.

About this time I happened to call at the plant and the manager asked me to take a diagram, telling me that he had just had the engine indicated but that something must have happened since that time. He said everyone had been questioned but that no one seemed to know anything about it. The plant is operated 24 hours a day, so I asked him about the night engineer. He replied that the latter did not know anything about an engine, hence could not have done anything to it.

I did not say anything but picked up my indicator and started for the engine room. The engineer gave me all the assistance he could and we soon took a diagram which showed that all the work was being done on the crank end. The engineer could not understand how the engine had gotten so badly out of adjustment in so short a time. I adjusted the governor rods and then obtained a good diagram, after which we threw on the load and the engine ran as good as ever.

After supper I walked over to the plant and waited around for a while until everyone was out of the boiler room except the old negro fireman. I then questioned him and our conversation was as follows:

"Are you on duty every night?"

"Yessah, I done bin on dis same job for fo' yeah."

"Who is your night engineer?"

"Oh, dat's Mars Frank."

"Well, who is master Frank?"

"Weel, you see Mars Frank done been de son ob de cap's brudder."

"Oh, yes, you mean that Frank is a nephew of your general manager."

"Yessah, I spose so; anyhow, I knows de Cap and Mars Frank been powerful good friends."

"Does Master Frank ever handle the engine at night?"

"Sometimes he stots up for de day man and he been always lookin' about for anything what might happen. He's an awful caful man, and is always workin' at sometin'. De oder night I thought he gwine t' kill hisself. He work on dem two little rods up da mos all night."

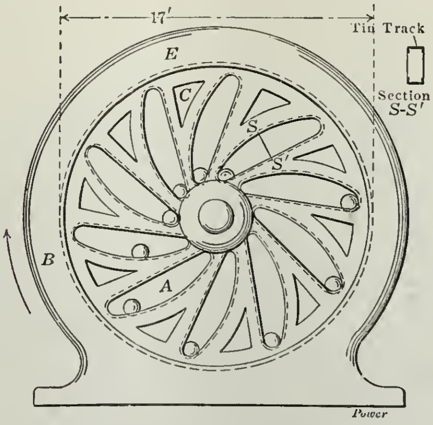
I did not say any more, but went on back to the house and went to bed. The next morning I told the manager he had better get an indicator and have his chief engineer indicate his engine eve   morning.

H. T. Fryant.

Macon, Miss.

A Perpetual Motion Artifice

A perpetual-motion device was recently exploited here for advertising purposes and, although much elaboration was manifest in a preparation to astonish the populace, it fell short of inciting the desired interest through a miscalculation in the concealed working mechanism—the only sane assumption. A short, narrow street in the business district was chosen, and two riveted-steel columns were erected upon either side of this thoroughfare, giving a span of about 35 feet and a hight above the street level of approximately 20 feet. Upon these posts the device shown in the accompanying sketch was mounted so that its action could be viewed from either end of the street.



Elevation of Wheel

The device was constructed of wood; the revolving wheel, about 17 feet in diameter, was divided into ten pockets *A* arranged as shown, and each containing a wooden ball running on a tin track as indicated in the section. The action of the balls was to impart rotation to the wheel in the direction of the arrow. The exterior rim of the wheel was concealed by a wood casing *B*, which contained a weekly change of advertising of local merchants. The hollow sections *C* were to all appearances used for ornamentation.

The implied principle is one of great simplicity, but it was noticeable that spasmodically the device would cease to rotate until overhauled and recharged. This creation has now disappeared from local view and advertisers who reserved space in advance, with payments, have lost interest in "perpetual motion."

In removing the device it was observed that a pocket in the casing, about the location of *E* in the sketch, was arranged to house a motor. If such an agency fails to excite "perpetual motion," then what?

R. W. Lawson.

Los Angeles, Cal.

Blowoff Pipes

Several years ago I was engaged to take charge of a 500-horsepower plant the boilers of which were of the horizontal-tubular type. Two of these were recent additions to the plant and had been bought secondhand, having been in use about a week before I took charge. I gave orders that these two boilers be cooled down the following Sunday in order that I might look over them, as the fireman had informed me that the handholes and also the manhole plates had not been taken out before the boilers were put in operation, and as they were not insured, no inspection had been made.

When I arrived at the plant at about 8 o'clock the following Sunday morning, I was informed by the fireman that he had been unable to empty the boilers as the blowoff pipe seemed to be stopped up. I took off the bonnet of the valve, but found that the obstruction was inside or between the valve and the boiler. As no means had been provided other than the removal of the pipe, other means had to be resorted to in order to empty the boiler. After considerable trouble we succeeded in removing the manhole plate and then procured 50 feet of 1½-inch hose, one end of which we placed in the boiler and the other end in the connection to the sewer, thus forming a siphon for removing the water. After all the water had been drawn out of the boiler, we attempted to take out the front handhole plate and in order to do so had to procure a piece of 4x4-inch timber 6 or 8 feet long to use as a battering ram. Having driven the plate in we found the whole inside of the boiler covered with a very hard scale, in some places 2 inches or more in thickness, and at the rear of the boiler the blowoff pipe was filled up and a large heap of scale over the hole.

After removing as much of the loose scale as we could with a hand pick we turned our attention to the blowoff pipe. This pipe was covered, beneath the boiler, with plastic asbestos, the removal of which disclosed a union inside the wall. One can imagine what the result would have been had this plastic cement cracked and broken off, exposing the union to the action of the hot gases.

We removed the pipe and installed a new pipe of the full size, and placed the union where it belonged (outside of the blowoff valve). The plates were then put on, and about one gallon of kerosene emptied into the boiler before any water was admitted. By so doing an opportunity was given for the oil to rise to the surface of the water and thus coat the sides and all parts in contact with the water, and thus afford an opportunity for the oil to work in between the scale and the iron and loosen the scale.

During the following week the connections to the blowoff were changed. Each individual blowoff was connected to a main pipe which extended to the river, and a gate valve was installed in this main for emergency use. For instance, if a blowoff pipe on any boiler became clogged, it was necessary only to lower the pressure on this particular boiler, close the valve in the main pipe and open the blowoff of one of the other boilers having a higher pressure, also opening the valve on the blowoff pipe that was clogged; in this way the obstruction would be forced back into the boiler. Of course, care and common sense had to be used in opening the valves slowly, although it was not necessary to use a great difference in the pressure, a difference of five pounds being sufficient.

Charles H. Taylor.

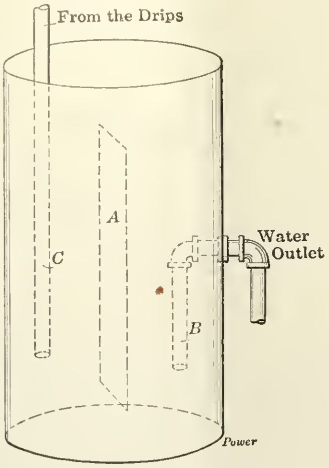
Bridgeport, Conn.

Engine Room Kinks

The following ideas, while not new to all, are probably new to many readers of Power, and as they have been found to be very helpful about a power plant, those readers to whom they are new may appreciate them.

To keep tools in order on a tool board or in a tool box, trace the outlines of the tools with a pencil and fill in the outlines with paint, thus making a rude picture of each tool in its proper place. It is no trouble at all then when putting tools away to put each one where it belongs. When a tool has been left out, the picture of it is a constant reminder of its absence and will cause one almost instinctively to get the tool and put it away, thus often preventing tools from becoming lost.

The figure herewith shows an oil trap which anyone handy with a soldering iron can make from a lard can or a deep tin pail and which it is well worth hiring a tinner to make if one does not care or has not time to make it himself. The part *A* is a partition which causes the oil and water, which enters through the pipe *C*, to rise to the top, thus permitting the oil to separate from the water before the water passes out through the overflow pipe *B*. If the drips from the oil trap on the heater or exhaust head are passed through such a trap, a large percentage of the cylinder oil will be recovered and by passing it through the filter with the engine oil will result in quite a saving.



Homemade Oil Trap

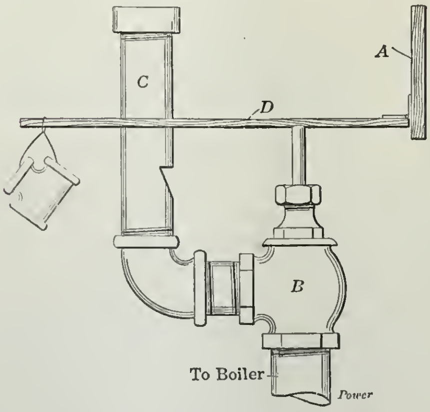
Any drainage from the engines which contains oil may be passed through such a trap and the oil recovered.

G. E. Miles.

Salida, Colo.

A Safety Whistle

While visiting a small plant located in the country, I noticed the device shown in the accompanying sketch and questioned the engineer (who also acted as fireman) concerning it. He stated that when the old safety valve gave out after 60 years' use on various boilers, his employer would not purchase another as there was no compulsory boiler inspection at the time. He therefore became apprehensive as he was often obliged to leave the boiler room after making a new fire, and as a result frequently stopped work in another part of the building and waited for the crash.



Valve and Whistle

Becoming tired of this he made the alarm, shown in the sketch, and attached it to the boiler. The piece of plank *A* is nailed to the rafters overhead, and to this is hinged the stick *D*, about 4 feet long. The angle valve *B* has the thread filed off the stem and the wheel removed, the upper part of the stem bearing on the stick *D*, while *C* is a whistle made from a piece of pipe. The tee hung at the end of the stick serves as a weight for varying the pressure at which the whistle will blow. All the fittings and pipe are ½ inch.

With this device installed, when the pressure reaches the safety limit the whistle blows and the engineer hurries to the boiler room and checks the fire.

James E. Noble.

Toronto, Can.

Pumping Problem

Several engineers were discussing various subjects when one submitted the following problem. A 6x12-inch duplex power pump delivering a total of 235 gallons per minute or 5.68 gallons per revolution (both cylinders) is driven by a 12-horsepower steam engine. The discharge pipe, which is 4 inches in diameter, runs along the ground for 200 feet, then rises vertically 200 feet and discharges into a tank. The water is drawn from four open wells. The first one is 100 feet from the pump with the water level 10 feet below the surface of the ground; the second is 110 feet from the pump with the water level 12 feet below the surface; the third is 120 feet from the pump with the water level 18 feet below the surface, and the fourth is 150 feet from the pump with the water level 22 feet below the surface. How much water will be delivered to the tank per minute, and how much water will be drawn from each well? The main suction is 5 inches in diameter and the down pipes at the wells, all of which terminate in the main suction, are 2½ inches in diameter. Several entirely different answers were given, so it was decided to submit the question to some of the readers of Power who are hydraulic experts.

James Ellethorn.

Toronto, Can.

A power-transmission cable is to b laid across Narragansett bay at the mout of the Providence river, between Conimicut point and Nayatt point. The distance across is about a mile and a third The line will transmit 3000 kilowatts a 11,000 volts. The cable will consist o three conductors, each with a cross-section of 100,000 circular mils.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

License Laws

A correspondent in a recent number of Power, in speaking of license laws, gives it as his opinion that they are useless or worse. It is doubtful if the men in the States or cities which have license laws would agree with him or could be induced to urge their cancelation. On the contrary, the tendency is to make the laws still stricter.

That wages are no better in license districts than elsewhere is, in a measure correct. For instance, here in Ohio the average wages for operating engineers are lower, I believe, than in any other State. I think that this is the result of faults on the part of both the employer and the employee. The employer will not pay more when he can hire for less and the employee, often, is too eager to get the position to hold out for or demand a salary consistent with the work and responsibilities.

Without doubt, license laws have resulted in a better class of engineers, but I do not think that anyone expects the laws to curtail the supply of good operatives, because the opportunity is open to everyone to study and no law or combination of men or laws can prevent a man from obtaining a license if he is really determined to do so. I have been examined by four different sets of examiners in different places and always have had fair play shown me. I have always found that if the applicant is capable and conducts himself properly, he will come out all right, and there will be no need of any appeal to the executive board of examiners.

The charge that engineers are pushing the license laws for selfish motives is, with very few exceptions, without foundation. However, I do not think that any of us are doing it from philanthropic motives. We are simply advocating these laws because we are in the best position to realize their necessity and the dangers resulting from the employment of incapable engineers. We do this for the same reason that doctors of medicine advocate and push the passage of health laws and urge the maintenance of health boards to see that the laws are obeyed. The doctors have nothing to gain but the general public has. Yet, I do not think that the doctors ever look on themselves as human benefactors on that account. They take the attitude which they do because they are in the best position to realize the necessity of such laws to safeguard the public. We engineers are for stricter and more universal license and inspection laws for similar reasons.

As I said before, license laws, so far, have not helped to raise wages, and whatever improvements have been gained in this line have come from the employers recognizing the fact that capable men, who were satisfied with their conditions, are worth more than incapable men whether satisfied or not. I belive that all future improvements will come from educating the employer to see things in this same way.

The trouble all along has come from the fact that most of the large power-plant owners have developed from small plant owners, and the plants have developed from the little dinky sawmill or stave-mill plant to the huge power or electric plant of today. The man who owned the little mill and hired his engineer for $1.75 or $2 per day has now developed into the power-plant owner who has thousands of dollars invested in his plant. But, while he would be greatly offended if anyone were to compare his magnificent plant of today with the little dinky mill of fifteen or twenty years ago, he does not hesitate to compare the engineers of the present plant who must be skilled mechanics, versed in a dozen different trades, and have a first-class technical education besides innumerable other accomplishments, with the old roustabout who ran the sawmill engine and oiled the machinery and possibly kept an old pulley with a wire through it handy to hang on the safety valve when an extra big log came up; and expect them to work for about the same wages.

However, this class of employer is passing and a new class is taking his place, a class of employer that recognizes more and more the responsibilities and requirements of a good engineer and is willing to judge and pay him on the basis of worth.

I need scarcely to emphasize the fact that engineers can help matters along by making good, and by not being afraid to put it up to the owners if they are not getting what they or the positions are worth. Do not look for license laws, or engineers' societies or unions to do it all for you; they are all useful and help to that end, but the long end of the lever is with you.

The present systems of license and inspection laws are very unsatisfactory. State or city laws will always be like sending a boy to do a man's work. There is too much politics in the present systems and a lack of uniformity which can never be corrected. The only way to do business is to do it right in the first place. Get Federal license and inspection laws and settle the matter once for all. Put them under the civil-service department, on a par with the marine license and inspection laws and we will have something worth while.

Allan A. Blanchard.

Oak Harbor, O.

Filing Catalogs

In a recent issue of Power, I saw a description of a system of catalog filing and I wish to offer a description of the system that I have been using for several years with good results. I have about 600 catalogs in the filing case.

I have a large case which is divided into four sections. Each section is divided in half vertically with a movable shelf in each half, thus dividing the sections in four parts and making it easy to take care of catalogs of various sizes. The sections themselves are 36 inches long, 27 inches high and 12 inches deep, and are fitted with two sliding doors so that one-half vertical section can be reached at a time.

On each catalog I place a small gummed label in the upper left-hand corner. The catalogs are numbered consecutively from one up. In cases where I have more than one catalog from the same firm, descriptive of different materials or machinery, I give each the same number and add a letter, as 23A, 23B. These gummed labels make it easy to change the number of a catalog without marring the catalog.

Labels are put on the outside edge of the shelves and on them are put the numbers of the catalog, those on the shelf at the top of the label and those under the shelf at the bottom of the label, as,

1 — 50
201 — 250.

These labels are always in plain sight. I further simplify the system by placing only catalogs numbered in the odd hundreds in upper half section and even hundreds in the lower half section, making the movable shelves the dividing line.

None of the hundreds is completely filled out as I wished to leave room for expansion and as the catalogs are not of uniform size.

The index consists of a card-index case divided in two halves. One side is used for the manufacturers' and sellers' names and the other side for the articles. Cross indexing is accomplished by indexing under a manufacturer's name all of the articles described in his catalog or catalogs and under the names of the articles all the manufacturers and sellers of that article from whom I have catalogs.

By this system I am able to change numbers and to keep my catalogs up to date with very little trouble, for being a card-index system it is easy to destroy old cards and typewrite new ones.

M. V. Bailliere.

Norwalk, O.

Blowoff Connections

The articles which have appeared occasionally in Power concerning blowoff connections have interested me greatly and I would like to contribute my mite on the subject.

Fig. 1 shows the blowoff and drip returns, in the rear of a pair of 72-inch return-tube boilers. The blowoff valve *A* is a 2-inch angle valve, installed so that the pressure comes under the disk. This valve is supplemented by the 2-inch gate valve *B* which is never closed unless valve *A* is out of order.

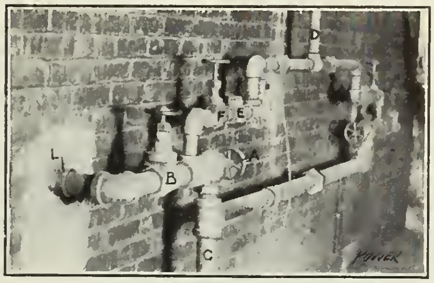


Fig. 1. Blowoff Connections

The pipe *C* connects the blowoffs from the two boilers to the main blowoff line. Between the valves *A* and *B* a 2x1¼-inch tee receives the drips from the heaters, which are returned to the boilers by means of either a return trap or a receiver and pump. The returns come through the 1¼-inch pipe *D*, pass through the check valve *E* and the globe valve *F* into the blowoff pipe. The blowoff pipe comes from the bottom of the boiler, as shown in Fig. 2. It is protected by the pipe sleeves *H*, which in turn are protected by 1½ inches of plastic asbestos *J*. In this connection I might add that I protect the handhole plate, saddle and nut with a generous wad of asbestos, as indicated at *K*.

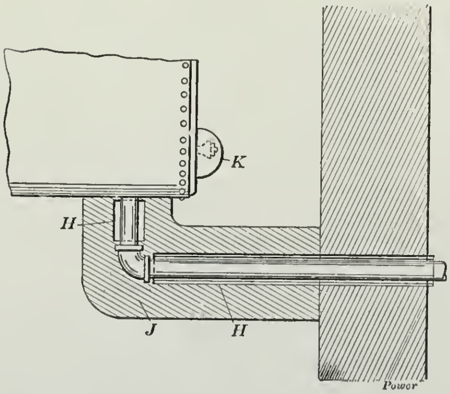


Fig. 2. Protection for Blowoff and Handhole Plate

The pipe sleeve around the blowoff is shown in Fig. 1 at *L*.

I put a new disk in the angle valve *A* once a year and have never had any trouble with it between times. However, if a disk should crack, or the valve should leak from any cause, or there should be any trouble with the 1¼-inch return line, the gate valve *B* would be closed, and the trouble remedied as easily as though the boiler was out of service.

E. H. Roberts.

Norwalk, Conn.

Engineers' Wages

In the September 13 and October 4 issues there appear articles under the heading of "Engineers' Wages" in which the authors discuss the advantages and disadvantages of unionism to some extent and at the same time unjustly criticize one of the best engineers' organizations in the country; namely, the National Association of Stationary Engineers.

I do not intend to fill up space here with a long list of items that would favor either the union or the nonunion man, but I do wish to stand up for the good name of the National Association of Stationary Engineers.

The gentleman who wrote the article printed in the September 13 issue says that the National Association of Stationary Engineers was organized for a good purpose, but as it now is, anybody may join. You are entirely mistaken, Mr. Writer. Anybody cannot join us. We want and have only engineers whose characters are good. As for men who know nothing having the floor to speak while the informed men listen, will you kindly tell us how a person is going to make known his thoughts unless he speaks them. All of us cannot be so well educated in the broad field of engineering that we are above listening to an engineer's argument, although it may not be one that would do justice to a professor. There are thousands of times when a few words spoken by an engineer, who may not be at the head of the list but who is willing and determined to get there, will pave the way for a discussion by which more than himself are benefited.

The author of the letter printed in the October 4 issue casts a few reflections upon our order by saying that it shows the fallacy of standing on one's own merits and letting the employer fix the scale of wages. If this man will stop and think of the great engineers of today who have charge of large plants, or, going farther, the engineers who once had plants but are now superintendents, he will find that they are men who leaned on their own merits for salary and advancement; and a large number of these men are members of the National Association of Stationary Engineers.

The purpose of our organization is educational advancement, and the purpose is carried out to a high degree. To either the union or the nonunion man I say that I can see no reason for his holding malice against us.

G. B. Longstreet.

Somerville, Mass.

The letter in the October 4 issue of Power, again touching on "Engineers' Wages," is certainly a "breeze." I am very sorry that there should be one man living in this enlightened day so ignorant of self-preservation and with such a swelled head as is there intimated. I am afraid that something disastrous would come to pass if we all were like that. However, we did not have any calamities in the California town where I worked years ago before there were such good rules for self-preservation. My critic discourses most ably on the conditions responsible for the improvement of the machinist and others. Who should know more about the conditions and ability of the present-day mechanic than the man who has had from 300 to 800 under his charge at various times in more than twenty States, and has superintended the building of some of the biggest engines, many of which were to fall into the hands of the poor $45 man who, when instructed how to care for the machine, had brains enough to know that things sent out right from the shop should not be tampered with?

We cannot judge the future by the present, as the writer in the October 4 number would lead us to think; we can only judge by the past. Unionism has cost many a poor man his home and caused much loss of life and property. I am not telling "pipe" stories here, nor did I in my last letter. My assertions are based on broad experience, not confined to one little State.

I favor social and educational organization. Schooled men are easily handled; they will listen, they can reason. But, alas, for the man who thinks that the ability to say "I have my card" is qualification enough for any position.

I claim to be a free-born American; I want to preserve my right to think free, act free and to stand alone as an individual; I want to be as good outside of a union as in one, or *vice versa*. Let the States furnish intelligent boards of examiners to ascertain if a man may be an engineer, carpenter or a machinist, as the case may be, and there will be more disqualified "card" holders than the proud charter member ever can count.

C. R. McGahey.

Lynchburg, Va.

In the October 4 issue of Power, Mr. Heyrodt has a letter criticizing one in the September 13 issue. Some of the remarks which he makes certainly seem unfair to me.

In the beginning, Mr. Heyrodt calls another man's article "rot," which I consider an unfair remark to make about any article which appears in this magazine. The publishers of this paper are very considerate in allowing us space in which to air our ideas and opinions and I know that they would not print anything that they considered rot.

Mr. Heyrodt tries to tell us that there are no such beings as "stopper and starters." He is very much mistaken—all the cities and towns are full of them; if he will stop to look around, he will undoubtedly find a few in his own town.

Mr. Heyrodt's remarks about unions and associations show that he has the two confused. The dictionary definitions of the two are pretty much the same, but there is a vast difference between the principles of a union and those of an association. For instance, he cannot name one union which does not indorse strikes while the National Association of Stationary Engineers does not recognize strikes at all.

L. W. Chadwick.

Bridgewater, Mass.

Installing Globe Valves

I most heartily agree with A. A. Blanchard when he states in the October 4 issue that the pressure should always be under the disk of a globe valve. But, why does he make an exception of globe valves on a feed-water line? I claim that a globe valve used for boiler-feed purposes should have the under side of the disk toward the pump, as better regulation of feed water is thus obtained, and in the event of the disk dropping off the stem, a shutdown will not occur as water may still be pumped into the boiler. This could not be done if the valve were placed with the under side of the disk toward the boiler.

Mr. Blanchard also claims that a globe valve has no business on a feed-water line, and that a gate valve is better. This would be all right where the feed pump was feeding one boiler only. Then the regulating would be done at the pump throttle. But, where a battery of several boilers are fed by one pump a globe valve on the feed pipe is much better than a gate valve because it is more convenient to regulate and also because a gate valve should never be left partly open but should either be wide open or tightly shut. It left partly open it will chatter or the seat will be cut. My practice is to place a gate valve on the feed-water pipe near the boiler, and a globe valve between it and the main feed-water line, for regulating the supply to the boiler. Of course, the gate valve must be closed at regular intervals or the seat will become scaled.

Thomas Henry.

West Toronto, Can.

Solid vs. Hollow Crank Pin

I noticed in the October 11 issue that Mr. Kavanagh states that a shaft or crank pin is stronger when hollow than if made solid. As I understand his statement he would use the same size of pin in either case.

I think Mr. Kavanagh is a little mistaken. If he increases the diameter of the pin so that it will have the same weight of metal as the solid one he will have a stronger pin but with the same diameter the hollow pin is weaker.

H. C. Stiff.

Chicago, Ill.

Putting in Sleeve in Pump Cylinder

The article by R. A. Cultra in the October 11 issue on "Pump Packing Kinks" reminds me of a case of the same kind which I once handled.

Mr. Cultra's method is the best to employ in putting in a new *solid* sleeve in a pump. If he had undertaken to drive the sleeve into place he very likely would have done more or less damage to the sleeve in the operation and would not have had as good a job either.

The case which I have mentioned was that of a vacuum pump used in connection with vacuum pans in the manufacture of glue. The sleeve was worn through at the bottom and the rod was cut and worn ¼ of an inch when I took charge of the plant. I at once ordered a new rod and a new sleeve. As the pump was in daily use, I took the old sleeve out and centered the rod and packed the plunger with common hydraulic packing, running the pump with the packing next to the cylinder casting. After this operation I got 25 inches of vacuum whereas I could get no better than 14 inches before.

I secured a sleeve made in two sections. The main section being cut and rolled to fit about seven-eighths of the circumference of the cylinder and the other, or wedge section, cut and rolled to fit the balance. This section is slightly smaller at one end than at the other to permit of a quick, neat, easy job in driving the sleeve into place with no danger of damaging either the sleeve or the pump.

I first took the cylinder head off and then the plunger out. I then oiled the cylinder on the inside to allow easy slipping of the sleeve into place. Next, I took a heavy piece of electric wire and coiled it around the main section of the sleeve four times and fastened the ends together. After filing the rough edge off the sleeve I put a wrench between the wire coils, two coils on each side of the wrench, in the space left for the wedge and twisted up on the wire until the sleeve came together enough to enable me to slip it into place with very little trouble. With a little good mechanical judgment the wedge section was driven into place just as easily. I took a block of wood first and placed it on the end of the wedge, using a hammer to drive with, then, when it was flush with the outside of the cylinder, I took a brass bar and completed the job. It took just one hour and a half to finish the job and get the pump going. It is now getting 27½ inches of vacuum and holding it. This was four months ago and the pump is still doing 27½-inch vacuum work.

R. P. Wilcox.

Chicago, Ill.

Bearing Lubrication

Referring to the letter of Mr. Kavanagh in the issue of October 11 regarding my comment on his article on "Bearing Lubrication" in the issue of July 19, I reiterate my former statement that a small tube in the crank pin, as illustrated in Fig. 1 of Mr. Kavanagh's article is not a good or in any way a modern method of lubrication. Especially so in the case where an engine has a long run of a number of hours, during which there is no opportunity of replenishing the tube. Suppose, for instance, through the admission of a little dust or foreign matter such a pin should warm up. If a good flow of oil could be turned on, in nine cases out of ten it would remove the trouble; but, not so in the case of the small tube. Will Mr. Kavanagh name some uptodate engine builders who are using this small-tube method as the sole means of lubrication for the crank pin?

Regarding the hollow-pin theory, I agree with Mr. Kavanagh that it is modern practice to make shafts, etc., of large diameter and hollow for the purpose of reducing the weight. But the amount that the crank pin of an engine of ordinary size, say, from 50 to 500 horsepower, is bored out is usually equal to about a ⅜-inch pipe in diameter, which is very small compared with the size of the pin and certainly would not be of much use as a reservoir for the lubricating medium.

Regarding the matter of soft-metal boxes, I must refer to the question in the latter part of my former letter, which Mr. Kavanagh neatly sidesteps, as to why do the majority of the large builders of the present time use babbitt-lined boxes and bearings in preference to hard bearings if they are not practical. In reference to the introduction of hard bearings in his plant and their success, while this may be so, why are large generators and motors fitted with cast-iron shells lined with soft metal in their bearings if they are not practical?

Regarding the knowledge of an engineer being limited, if Mr. Kavanagh is thus alluding to my 20 years' experience or to my statement that the engine in my charge has not been rebabbitted during that time, I say that I did not state that the 20 years had been spent in one plant. It has been spent in several plants, enough to learn, anyway, that while hard boxes may be successful in some cases they do not fit one and all and it is safe to say it will be many years before they eliminate the soft-metal box if they are ever able to do so.

Charles H. Taylor.

Bridgeport, Conn.

Dissatisfaction and Jealousy

In the October 11 issue, A. C. Terlene writes of his experience with a chief engineer who was also, as would appear from the description given, chief martinet. According to Mr. Terlene, this chief engineer treated him with less courtesy than he did the "niggers."

The editorial under discussion took up the subject of dissatisfaction and jealousy from an entirely different viewpoint. It contemplated, if I can interpret English, cases where this condition of mind was allowed to develop without cause. It did not contemplate cases where there was *cause* for dissatisfaction. Any self-respecting man should expect to be treated *fairly*. There are times when it seems that the best man gets the "worst of it," but the situation is usually such that a man of sense can see the reason why. There are times when the chief, or any other superior, must depend on someone that is reliable. He then selects the man that in his opinion meets the requirements. He can do this in such a manner that the assistant will know *why* the job is put up to him, and if he is the right sort, he will be glad that he is trusted to such an extent by his superior as is indicated by his selection for the work in hand.

I believe in discipline in the power plant. It is as essential here as in any other line of work, if not more so. A spirit of good fellowship ought to exist among power-plant employees. Unfortunately some men will not be treated well. When the chief shows a kindly disposition some take it for granted that he is "easy," and proceed accordingly. Right here tact, and sometimes the "can," is required. But with the right sort of men it is possible to have discipline without becoming a crank or martinet. In fact, as a rule, the latter type of man generally gets what is coming to him in the way of poor service from the men he must depend upon largely for results. You can drive a man when you are present, but you cannot get your rest and be on the job twenty-four hours a day, and when you are not present the men are "handing it to you."

My practice has been to eliminate as rapidly as possible men who cannot be treated decently. But I try to make every man *feel* that he is going to get a square deal. I have operated power plants with all sorts of help, even to convicts in a State prison, and I have tried out the plan and know that it wins. I had charge of the power plant of a State prison in which nearly all the help consisted of convicts.

I treated these men kindly, and never had to send one up for punishment. On more than one occasion these men in the boiler room worked as faithfully on hard pulls when we were short on boilers as ever men worked for salaries. This made me believe that after all men are simply men, and what will win in one case will almost certainly win in another. We all like the fellow that treats us nicely. If we do a good "stunt" we like to have a little credit, and it comes so cheap, I often wonder why it is that chief engineers do not dispense a little of it on proper occasions.

I can only express the hope that while praising, Mr. Terlene has not imitated the example of the chief referred to in his letter.

William Westerfield.

Lincoln, Neb.

I read with much interest A. C. Terlene's letter in Power for October 11. He claims that the chief engineer under whom he worked continually found fault with his work, and he then tries to justify this action by the fact that the chief was his friend. But, at the same time, he says that he knew all the time that he was being imposed upon. For a man to show his friendship by imposition seems to me to be a very poor way of showing appreciation of efficient service.

I have been constantly in charge of power plants for the past fifteen years and I have never found it necessary to find fault with my help to secure good service. But, I have found just the contrary. I have found that good treatment with good men always brings forth its true value in efficient service. I would not keep a man that I had continually to find fault with, nor would I expect a man to stay with me when I was finding fault with him without cause.

Some engineers do find fault with their help without cause, and I suppose they do this to show their authority. But I think that about all it shows is their lack of good sense and gentlemanly principles.

H. R. Rockwell.

Mt. Vernon, Ill.

The Canton Boiler Explosion

In the October 11 issue, Mr. Jeter says that waterhammer may be caused in a boiler by steam of higher pressure being let into the boiler. I think that he is incorrect in this. I would like to know what proof he can offer.

Daniel Olson.

New York City.

Forced Draft and Boiler Economy

The subject of forced draft as handled by C. E. Roehl in the October 25 issue presents in several instances strong indorsement of forced-draft usage. But, unfortunately, Mr. Roehl presents his subject without specifying equipment.

He states, "Artificial draft is used for obtaining capacity above the normal with ordinary fuel, or for securing commercially satisfactory capacity with low-grade fuel." I would rebut this statement by saying that forced draft as studied and produced by the company with which I am connected has been used with the exclusive idea of generating steam with the lowest possible cost without regard to fuel or amount of capacity desired.

Again, Mr. Roehl says, in effect, that when the draft used is excessive the coal is blown from the grates and a good part of it is kept dancing in the furnace. This is not true as grate bars specially designed for forced draft will give this effect on any small-size anthracite coal burned, provided sufficient draft is admitted to properly burn the fuel. The statement would be right if applied to the ordinary natural-draft grate commonly but improperly used with forced draft.

Another point, is the statement that "It is true that maintenance charges increase." This is not true of a properly designed forced-draft equipment.

Further, he states, "It is practically impossible to avoid spots where for a time the combustion rate becomes extremely high." This statement is not true if properly designed grate bars are used, and in the term "properly designed" I put a strong protest against the usage and application of forced draft by many people to the ordinary grate bars built for natural-draft purposes.

I think that Mr. Roehl's article on the subject in a general way is excellent and I thoroughly agree with the conclusions and deductions which he puts forward as the common evidences and findings where forced draft is secured by the installation of a blower without other changes being made in the boiler furnace, but where properly designed grate bars, blowers and bridgewalls, and a means of adjusting and governing the blowers are installed as a part of the forced-draft system the specific faults cited by Mr. Roehl will be eliminated.

Charles H. Parson.

New York City.

The Problem of Smoke Abatement*

By D. T. Randall

In localities where it is necessary to use bituminous coal high in volatile matter smoke can be prevented only by the installation of especially designed furnaces; even then, unless utmost skill is used, dense smoke will be produced during the periods of firing. Properly designed automatic stokers are superior to hand-fired furnaces for economy and smokeless operation.

Nearly every community that has attempted to prevent smoke has first made the mistake of trying to do so by merely passing ordinances making it unlawful to produce smoke. Experience has shown that such measures alone are not effective and that results can be accomplished only by recognizing the difficulties and overcoming them along engineering lines.

In general a plant which produces smoke may be improved in two ways: by changing the fuel or by changing the conditions under which the fuel is burned. Many have met the difficulty by burning anthracite coal, but as it is not practicable for everyone to use anthracite coal, the use of bituminous coal presents an important and difficult problem which can be solved only after careful investigation as to the nature of the fuels available and the most successful methods of burning them under varying conditions.

Practically all coal burned in New England is obtained from Pennsylvania, Maryland or West Virginia, and the problem is not essentially different in this territory from that in many of the Eastern cities which burn large quantities of bituminous coal.

Most of our boiler plants were originally built to burn anthracite coal, and are not properly equipped to burn the higher-volatile, bituminous coals. Some investigations have been made by Dr. Horace C. Porter, of the Bureau of Mines, to determine the relation of volatile matter to smoke, and it appears from his experiments that coals vary considerably in the character of their volatile matter, and that the percentage of volatile matter, as determined by the proximate analysis, is not necessarily a correct measure of the difficulty which may be expected in preventing smoke. Experience in burning different kinds of coal has demonstrated that some of the coals which are high in volatile matter, as shown by the proximate analysis, are less difficult to burn without smoke than others which are lower in volatile matter. Further studies have shown that in such cases the volatile matter is not all combustible but is composed in part of inert matter such as carbon dioxide and other combinations which do not burn, and that the real volatile combustible may be much less than appears from the ordinary coal analysis. When coals of the same general character are considered, however, the volatile matter is a very fair measure of the difficulty which may be experienced in burning them without smoke. The difference between a coal containing 16 per cent. volatile matter and another one containing 21 or 22 per cent. volatile matter is usually noticeable when a hand-fired furnace is used. A coal containing more than 25 per cent. volatile matter is difficult to burn without smoke.

Comparison of coals should be made on the basis of the volatile matter free from ash and moisture, and an allowance should also be made for variations in the percentage of volatile matter, which may be due to the determinations being made in different laboratories. Some recent experiments by the Geological Survey have shown that as much as 2 per cent. difference may appear in the same coal, due to variations in laboratory conditions such as the kind of gas used, or whether the platinum crucibles are polished or dull when used.

If the coal contains large quantities of ash the fires must be cleaned more frequently, and in addition it is more difficult to secure an even distribution of air through the bed of fuel and ash. If the ash has a low fusing point and melts and sticks to the grates, air will be shut out over portions of the grate, and the difficulty of preventing smoke will increase.

An increase in the moisture in the coal usually increases the difficulty of burning it without smoke. If a coal cokes in the fire and if the draft is such as to require the frequent breaking up of the fuel bed, the coal may give off more smoke at such times than when it is fired.

The size of the coal is also of considerable importance in connection with its combustion, for the reason that the coals which are coarse must be carried at a greater depth on the grates in order to prevent an excess of air, and the finer coals must be fired very carefully in order that they may not clog the air openings through the fuel bed and thus prevent a sufficient amount of air from reaching the gases.

Now that there are numerous byproduct coke ovens being operated for the production of both gas and coke, it may be possible in many cases to utilize the coke and coke breeze with good results. The coke contains but little volatile matter and gives a high efficiency when burned under boilers.

It is advisable for each power-plant owner to make a study of the coals which are available for his plant and to choose the ones which are readily available at reasonable prices, when the heating value is considered, and to determine whether the furnaces now in his plant are adapted to burning such fuel with economy. In choosing a coal for use in a given plant the following points should be considered:

The amount and character of the volatile matter in the fuel.

The amount of ash and its tendency to clinker.

The moisture in the coal.

The coking and caking qualities of the coal when heated.

The size of the coal.

The amount of coal to be burned in a given furnace.

The kind of furnace—hand-fired or automatic.

The available draft and its regulation.

The character of the load—steady or variable.

The ability of the firemen.

The combustion of solid fuels which are low in volatile matter is comparatively simple as compared with the combustion of bituminous coals. When fuel such as coke is burned on a plain grate, air is admitted through the fuel bed and the oxygen unites with the lower layers of carbon to form carbon dioxide, which, on passing up through the bed, is reduced to carbon monoxide, CO. Provided the air is admitted uniformly through the grate, the CO will be burned within a short distance above the fuel bed. The burning of bituminous coal, however, is more complicated for the reason that when charging, the volatile gases begin to distil off as soon as the coal is heated to the required temperature. These gases are similar in character to unpurified illuminating gas and contain tar and heavy hydrocarbons. They will produce smoke unless burned under very favorable conditions. If air could be admitted in sufficient quantities and in such a manner as to thoroughly mix with these gases as they are distilled from the coal, they could be completely burned without smoke, provided the temperature of the furnace were sufficient to ignite them. The fixed carbon remains on the grate and burns in a manner similar to coke or other fuels containing a low percentage of volatile matter.

It is evident that the design of a furnace for burning bituminous coal must be different from one for coke or anthracite coal. It has been found to be practically impossible in burning many coals in a hand-fired furnace to secure a proper mixture of the air and volatile gases and to burn them before they reach the cooling surface of the boiler, where the flame is extinguished and the gases pass on up the stack unconsumed, and small particles of carbon are deposited on the boiler as soot or pass up the stack, forming black smoke. The most successful designs of furnaces for burning bituminous coals provide for the admission of air to mingle with the liberated gases before they leave the furnace. Walls or arches are often provided, which cause the gases to whirl and mix, and long or high combustion chambers to allow time for the combustion to take place.

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*Abstract of paper delivered on November 11 before joint meeting of Boston engineers.

If it is found that an equipment is not suitable even when the most successful methods of firing are adopted, it will then, in most cases, be a good business policy to invest in an improved type of furnace which will give practically complete combustion. There are many cases in which the equipment is old and it would not be advisable to install new and expensive furnaces under an old boiler, or it may be that the power plant is in a rented building. In such cases the solution must be effected by choosing a coal which is best suited to the furnace already installed. By mixing bituminous coal with an equal weight of anthracite screenings, quite satisfactory results may be obtained and without much difficulty from smoke. To burn such a mixture usually requires a little stronger draft than for bituminous coal alone.

With a fuel which is practically all fixed carbon, such as coke or anthracite coal, nearly all of the combustible matter is burned on or just above the fuel bed and for this reason it has been found satisfactory to build furnaces for anthracite coal with the grate very close to the boiler. With fuels containing higher quantities of volatile matter, the combustible matter is burned partly on the grate and partly in the space above and beyond the grate, this distance depending upon the character of the fuel, the rapidity with which the gas is driven from the fuel and the space which is provided for the combustion. If the gas is driven from the coal so rapidly that it is impossible to supply enough air to thoroughly mix with it so that it will burn before it reaches the colder surfaces of the boiler, the flame will be extinguished and much of the gas will escape to the stack without burning.

In designing a combustion chamber, a large capacity is not necessarily a solution of the problem; it must be of such a shape that the gases will pass at a fairly uniform rate over its entire cross-section. A study of a great many plants seems to indicate that the distance through which the gases travel before they reach the heating surface is of more importance than the area of the passage through which they travel, for the reason that they have a tendency to travel in lines the shortest distance from the grate to the heating surface. In this connection it has been noted that the common, horizontal return-tubular boiler has given very good results in comparison with other types of boilers, evidently because of the fact that the travel of the gases is over a long path before they reach the tubular heating surface. Whenever a device is provided which tends to give a more intimate mixture of the air with the gases, it is possible to shorten the combustion chamber accordingly.

For each kind of coal and each furnace, there is apparently a range of capacity through which it is possible to operate without a serious production of smoke. At higher rates the efficiency decreases and black smoke is produced owing to a lack of furnace capacity to supply air and mix it with the gases.

In firing a furnace by hand, relatively large quantities of fuel must be placed on the fuel bed at one time; this tends to retard the flow of air while the heat in the furnace tends to drive off the gases rapidly. Usually enough air cannot be supplied at this time and there is smoke after each firing. There are a few coals, however, which may be fired by hand without producing much smoke, provided the rate of combustion is low and the firing is done carefully.

The handfired furnaces, depending upon brick baffles, arches and piers are better than the plain furnace, and they generally decrease the smoke after the brickwork has been heated. Their capacity to absorb heat, however, tends to counterbalance the gain from better combustion if a large amount of brickwork is used and if the period of operation is short. A brick arch over the entire grate surface increases the rate of combustion as it reflects the heat back onto the fuel bed, which assists in driving off the moisture and the volatile matter. This action may cause more volatile matter to escape than can be burned in the combustion chamber. In many cases more successful results are obtained by omitting the arch above the grate and providing arches, piers and checkerwork at the bridgewall and a short distance behind it. Provisions for admitting extra air for a short period after each firing will often reduce the smoke to one-half the amount produced when only the regular supply is furnished through the grates and fuel bed.

In general, furnaces which depend upon designs as outlined, may be operated by skilled firemen with certain coals at a good economy and with smoke only at short intervals after firing or cleaning, providing the rate of combustion is low, but they cannot be depended upon to give good results as regards smoke if the load is variable or the firemen are incompetent or careless.

Hand-fired furnaces properly equipped with steam jets and air admission may be operated with most coals so as to give but little black smoke except when cleaning the fires. The jets must direct the air in the required direction to secure a rapid mixture of the gases in the furnace. This often requires some experimenting. Steam jets require a large amount of steam to secure satisfactory results. In connection with large combustion chambers or well designed brick furnaces, fewer jets may be used or the pressure may be reduced to save steam. Often the inventors of patented devices of this kind meet the objection to the cost of operating steam jets by claiming that their system is so designed as to convert the steam into its original elements—oxygen and hydrogen—and that these are burned with a resulting economy which is very high. From a theoretical standpoint these arguments are not correct, although without doubt the use of steam jets improves the combustion, and if the methods of firing are improved at the time the apparatus is installed, the resulting economy may fully or partially offset the loss due to the use of steam.

With a hand-fired furnace and a horizontal combustion chamber, it is found advantageous to admit air freely over the fire for a short time after firing; but with boilers and furnaces having the heating surface directly above the grate, this method cannot be used, as the air simply passes to the heating surface without mixing with the gases. In such furnaces it is necessary to accomplish the mixing by means of steam jets, unless sufficient air can be admitted uniformly through the grates.

A study of the tests conducted at the Government fuel-testing plant shows that with hand-fired furnaces the best results were obtained when the firing was done most frequently and with the smallest charges of coal. In general, coals which smoke badly give efficiencies from 3 to 6 per cent. lower than coals which burn with but little smoke.

The difficulty in burning coals at high rates of combustion with good results and without smoke has led to the design of automatic stokers which are intended to supply coal in small quantities continuously or at regular intervals and to introduce air in finely divided streams at the point where the gases are liberated from the coal. The coal is fed in such a way as to be subjected to the heat gradually, and at a comparatively low temperature the gases are driven off and then mixed with air and completely burned in a combustion chamber before they reach the cooling surface of the boiler tubes. Many of these have been on the market for a considerable period of time and have been improved until they may be depended upon under normal conditions to burn almost any kind of bituminous coal successfully.

As a rule these automatic stokers must be installed with special settings and combustion chambers, and it has been found that a stoker which is successful under certain conditions may not be successful with other types of boilers and other fuels, unless the problem has been given careful attention and the proper proportions worked out.

The chain-grate stoker is successful in burning coals high in ash and volatile matter. With proper handling they are capable of burning such coals at high rates of combustion without smoke. These stokers, however, are not adapted to burning low-volatile coking coals such as are commonly used in New England.

There are several stokers on the market that feed the coal from hoppers onto inclined grates, which are kept in motion by a driving mechanism. Some of these are so designed as to grind through a large portion of the ash which accumulates at the bottom of the grate. Whether this type of stoker is successful or not depends not only upon its durability and resistance to the action of the heat but also upon its ability to feed the coal uniformly, heat it gradually, and introduce a supply of air which will mix with and burn the gases before they have traveled far from the point at which they were liberated.

It is only within the past few years that the manufacturers of such equipment have given serious consideration to the smoke problem and have recognized that the stokers must be set in especially designed furnaces, with regard to the kind of coal, the kind of boiler and the kind of service for which they are intended. It is a mistake to install these or any other stokers with the belief that because they are called automatic, they do not require as much or even a higher degree of skill than does a hand-fired plant, if good results are to be secured.

There are two or three makes of underfeed stokers on the market. If they are so designed as to be automatic in regard to the supply of coal and air, they may be operated with very good efficiency; and when reasonable care is given to the operation, they will burn the high-volatile coals with but little, if any, smoke except when they are being cleaned. As the air is supplied to these stokers under pressure and is forced through the fuel bed, it is possible to secure an intimate mixture and to burn the gases within a short distance of the fuel bed. For this reason the combustion space is usually less than with other types of stokers. Owing to competition, companies selling this type of apparatus have in some cases made the mistake of installing a stoker too small for the purpose, and the results have been unsatisfactory.

The problem of smoke abatement in power plants is not solved when competitive bids have been secured from stoker manufacturers and an equipment contracted for, even though smokeless operation is guaranteed. The type of stoker selected should be one that will handle the cheapest coals, when both quality and price are considered; it should be of such design as to provide sufficient grate surface and an ample combustion chamber; it should have as few working parts as possible exposed to the heat of the furnace; and it should feed the coal continuously or in small quantities at frequent intervals in order that the fuel and air supply may be properly adjusted with respect to each other.

TABLE 1.

Time.	Draft back of boiler.	Thickness of fire inches.	Gas Analysis.			Coal Fired.
			CO$_2$	O$_2$	CO	
4:03	0.10	18	....	...	...	Fourteen shovelsful.
4:07	....	18	13.6	1.5	2.3	
4:20	....	18	....	...	...	Fifteen shovelsful.
4:25	0.11	17	12.5	1.1	3.5	
4:37	....	18	....	...	...	Fourteen shovelsful.
4:42	....	18	....	...	...	Raked fire.
4:45	0.21	18	13.7	1.0	6.7	
5:05	....	18	....	...	...	Fifteen shovelsful.
5:15	0.25	18	18.2	0.6	1.0	
5:33	....	18	....	...	...	Sixteen shovelsful.
5:35	0.20	18	15.3	1.0	3.6	
5:55	....	..	....	...	...	Seventeen shovelsful.
5:58	0.08	19	12.0	0.3	4.7	

TABLE 2. ACTION OF DAMPER REGULATOR.

Time.	Draft at outlet from boiler.	Remarks.
P.M.		
6:25	0.06	Damper closed.
6:26	0.35	Damper half closed.
6:27	0.07	Damper closed.
6:30	0.42	Damper open.
6:32	0.46	Damper open.

It is always necessary to take into consideration the draft and the methods of regulating it in both hand- and stoker-fired plants. The best results are usually obtained when the draft is low for the reason that it is less difficult to maintain an even fuel bed, and because there is less leakage of air through holes in the setting. A great many chimneys are smoky because the draft is insufficient to supply air in the correct proportion at critical periods of operation. Damper regulators are often the cause of smoke, and it often happens that firemen charge large quantities of coal into the furnace when the dampers are closed and there is no air admitted to burn the gases.

In many plants difficulty is experienced in preventing smoke at times when the load changes rapidly, or when the boilers are heavily overloaded. Often the furnaces smoke when boilers are cut out and banked, and they are also liable to smoke when they are being brought from a banked condition into service. It is difficult to start a fresh fire with bituminous coal under a cold boiler without producing smoke. This problem is being met at the present time in some boiler plants, in foundry cupolas and other industrial furnaces, by using a bed of coke, ignited by means of a gas or oil flame from a portable burner installed for the purpose. This will build up a fuel bed and bring the furnace walls up to a high temperature so that fresh coal may be fired without difficulty.

To illustrate the variations in conditions under which coal is being burned, the following notes on boiler plants are submitted. The observations, the results of which are given in Table 1, were made at an electric-light plant equipped with Babcock & Wilcox boilers, rated at 350 horsepower each. These boilers were set to give about 42 inches between the grate and the tubes:

This analysis shows that when several shovelsful of coal were fired at one time, large amounts of carbon monoxide and unburned gas escaped from the furnace and that there was also a considerable loss after long intervals when the damper was nearly closed. The stack smoked almost continuously during this period and the smoke was more than 60 per cent. black for periods of four or five minutes after each firing.

TABLE 3. GAS SAMPLES AT POINT WHERE FLUE ENTERS BOILER ROOM.

Time.	CO$_2$	O$_2$	CO	N$_2$	Draft, inches of water.		Flue temperature degrees Fahrenheit.
					Over fire.	In flue.	
Jets on							
1:00	12.0	8.0	0	80.0	0.30	0.40	470
1:20	11.0	7.9	0	81.1	0.35	0.35	480
1:40	11.8	7.7	0	80.5	0.30	0.30	470
2:00	10.2	9.3	0	80.5	0.30	0.40	490
2:20	11.2	8.3	0	80.5	0.30	0.35	490
Jets off.	12.6	6.2	0.3	80.4	....	....	...
Jets off.	12.6	6.6	0.6	80.2	....	....	...
Jets off.	11.9	7.4	0.9	79.8	0.25	0.30	490
Jets off.	13.5	5.4	0.6	80.5	....	....	...

These readings indicate the frequency of changes in the draft due to the action of the damper regulator. Such changes are unfavorable to good combustion. A damper regulator in perfect working order should not permit such fluctuations in draft.

The analysis of the coal burned at this plant was:

Moisture	2.69 per cent.
Volatile	16.30 per cent.
Fixed carbon	73.99 per cent.
Ash	6.92 per cent.
Total	100.00 per cent.
Sulphur	0.61 per cent.
B.t.u.	14,207

These results indicate that the conditions under which the coal was burned were very unfavorable, that the fire was too thick, that too much coal was fired at one time, and that the draft was not properly regulated. The combustion chamber was small and the gases passed directly from the fuel bed to the boiler tubes.

The readings given in Table 3 were taken on a 225-horsepower vertical fire-tube boiler of the Manning type, having a grate surface of 41½ square feet, and provided with two firing doors and eight steam jets, the distance from the grates to the heating surface being 6 feet. This furnace was such that it could not be operated at full capacity without smoke unless steam jets were used. The coal burned was of the following composition:

Moisture	1.87 per cent.
Volatile	18.18 per cent.
Fixed carbon	73.55 per cent.
Ash	6.40 per cent.
Total	100.00 per cent.
Sulphur	0.63 per cent.
B.t.u.	14,538

Each steam jet was located about 3½ feet above the grate and drew in a quantity through a pipe surrounding the steam pipe, provision being made for the regulation of this air supply. The fire was maintained level at a thickness of about 12 inches, and coal was charged in quantities of eight or nine shovelsful at a time through one door of the furnace. At intervals of eight or ten minutes, depending upon the load; the other door was charged with a similar quantity of coal. The rate of combustion was about 20 pounds of coal per square foot of grate.

With the steam jets on there was practically no smoke issuing from the top of the stack, but without the jets there was considerable unburned gas and smoke of varying density.

The observations in Table 4 were taken in a boiler plant burning coal of the following composition:

Moisture	3.38 per cent.
Volatile	20.32 per cent.
Fixed carbon	66.94 per cent.
Ash	9.36 per cent.
Total	100.00 per cent.
Sulphur	2.22 per cent.
B.t.u.	13,528

TABLE 4.

STIRLING BOILER.

Time.	Minutes.		Position of damper	Draft, inches of water.		Flue Temp., Degrees Fahrenheit.	Gas analysis, percentage by volume.				Excess air, per cent.
	After firing.	Damper had been closed or opened.		Over fire.	In flue.		CO_2	O	CO	N	
10:40	4	1	Closed	0.04	0.05	480	12.7	0.9	7.0	79.4	5
10:55	2	9	Open	0.03	0.04	455	13.4	0.5	4.9	81.2	3
11:10	10	24	Open	0.11	0.16	445	7.1	12.1	0.8	80.0	136
1:05	2	5	Closed	0.03	0.04	445	16.2	0.6	0.8	82.4	3
1:20	6	11	Open	0.17	0.26	455	5.5	14.9	0.0	79.6	248
1:35	2	1	Closed	0.03	0.04	420	13.0	1.0	4.6	81.4	5
1:50	8	3	Open	0.09	0.12	420	8.8	9.9	1.9	79.4	90
2:05	5	18	Open	0.08	0.11	410	10.1	8.3	1.4	80.2	65
Average.	....	....	....	0.07	0.10	440	10.8	6.0	2.7	80.5	69
HORIZONTAL RETURN-TUBULAR BOILER.											
2:30	5	43	Open	0.16	0.23	455	11.5	8.5	0.0	80.0	68
2:45	0.05	58	Open	0.09	0.13	450	13.4	7.0	0.0	79.6	50
3:00	0.05	73	Open	0.15	0.21	475	13.3	5.5	0.0	81.2	35
3:15*	2	88	Open	0.13	0.19	500	15.0	3.8	0.4	80.8	22
3:30	2	103	Open	0.15	0.22	490	12.6	7.2	0.0	80.2	52
3:45	12	9	Closed	0.14	0.20	450	11.6	8.8	0.0	79.6	73
4:00	7	24	Closed	0.13	0.17	460	12.1	8.0	0.0	79.9	62
Average.	....	....	....	0.14	0.19	470	12.8	7.0	0.1	80.1	52

*Sliced fire 3:14½.

	Stirling Boiler.	Horizontal Return-tubular Boiler.
Average thickness of fire, inches	11	9
Average weight of coal fired each time, pounds	65	55
Average interval between firings, minutes	9	7
Percentage of time damper was open	60	80
Pounds of coal burned per square foot of grate surface per hour	10.3	14.5

The plant was equipped with a 150-horsepower horizontal return-tubular boiler having 33 square feet of grate area, and a Stirling boiler rated at 225 horsepower, having a grate area of 48 square feet. The draft was regulated by means of a damper regulator operating on individual dampers at each boiler. During the time the observations were taken on the Stirling boiler the load was light and the damper was closed most of the time. The stack smoked badly during this period. While the observations were being taken on the horizontal return-tubular boiler the greater part of the load was on this boiler; a higher rate of combustion was maintained and economical operating conditions resulted. It is probable that if this boiler had been operating alone, the amount of smoke would have been greatly reduced.

It will be noted that the damper on the horizontal return-tubular boiler did not close sufficiently to reduce the draft. The method of firing in this plant was very poor; there was too much coal fired at one time and the intervals between firings varied from 2 to 18 minutes. The fireman paid no attention to the position of the dampers and coal was charged in large quantities when the damper was closed.

These examples serve to emphasize the importance of favorable kinds of coal, careful firing and large combustion chambers. It will be observed that with careful firing results were obtained which compare favorably with plants in which automatic stokers are being operated. The best results were obtained with plants burning low-volatile, low-ash coals, and with practically constant loads, usually below the rated capacity of the boilers. It has been demonstrated that plain hand-fired furnaces may, under favorable circumstances, be operated without smoke but that under ordinary conditions they are inefficient and smoky.

During the investigations of smokeless plants by the United States Geologica Survey, it was noted that very few were being fired by hand without smoke unless steam jets were used for at least a minute or two after each firing. Some plants were doing fairly well without steam jets but they were in many cases burning large-sized coal and admitting an excess of air, or they were operating at a low capacity. Some of the better-operated plants were not smokeless but the stack smoked only for one or two minutes afte each firing, beginning with a density o about 60 per cent. black and diminishing to about 20 per cent., at the end of hal a minute. Such plants are not usuall subject to fines. General deductions may be drawn as follows:

Plain hand-fired boilers may be op erated with but little smoke with low volatile coals, or, in some cases, wit other coals if the rate of combustion is low.

Hand-fired furnaces with brick arches are more easily operated without smoke than the plain furnaces.

Almost any hand-fired furnace may be operated by means of steam jets so as to produce but little smoke.

While the preceding may be accomplished by a skilled fireman, such plants cannot be depended upon for smokeless results at all times. Experience has shown that a change of fuel or a change of firemen may cause them to become smoky plants, due to a difference in methods or rate of firing.

Stokers without suitable combustion chambers or when improperly operated may be expected to produce a great deal of smoke.

Experience has demonstrated that the best stokers properly installed are superior to hand-fired furnaces for economy and smokeless operation.

Boiler Explosion at Union City, Ind.

By Charles F. Read

A 54-inch return-tubular boiler working under a steam pressure of 55 pounds per square inch exploded with much damage to property but no loss of life. The boiler was twenty-nine years old.

A boiler recently exploded at the Witham & Brown Lumber Company's plant at Union City, Ind., wrecking a large part of the plant but fortunately killing no one, although there were several injured and many had miraculous escapes.

The boiler was of the return-tubular type, 54 inches in diameter and 16 feet long, and had been in service for 29 years. The heads were ½ inch thick and the shell had been $\frac{5}{16}$ inch thick when new but had wasted to ¼ inch. The girth seams were single-riveted and the longitudinal seams double-riveted lap joints. The dome was single riveted to the shell and the mud drum was connected to the bottom of the shell by a 4-inch cast-iron pipe.

In December, 1908, the boiler had been inspected by a local boilermaker who allowed a pressure of 60 pounds per square inch when used for heating drying houses, and 40 pounds per square inch when supplying steam to a small slide-valve engine. This engine, however, was run only occasionly, as a gas engine was used to carry the usual load of the plant.

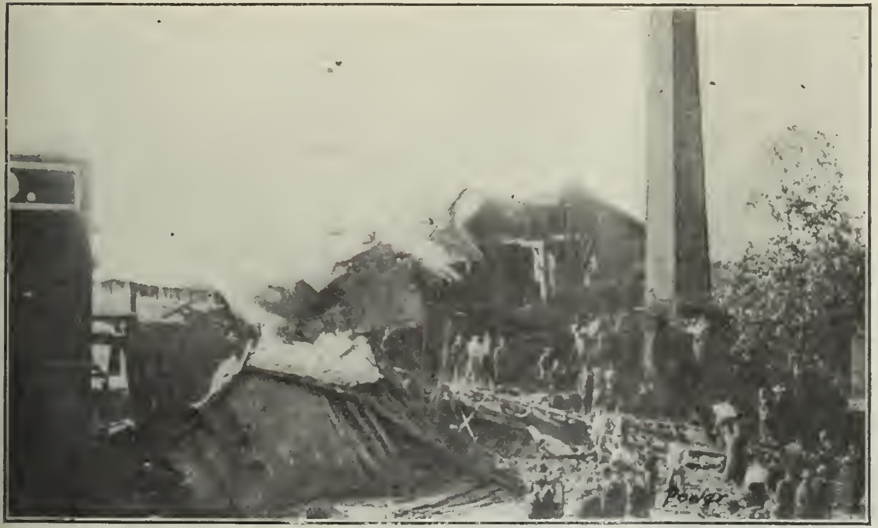

Fig. 1. Wrecked Building and Chimney

At the time of the explosion the working pressure on the boiler was 55 pounds, the engineer having glanced at the steam gage a minute before and walked into the frame room on an errand, thereby saving his life. Without the slightest warning the boiler gave way, wrecking the building and rending itself into many pieces which landed from one to two hundred feet distant. A curious feature is that a 40-foot brick chimney standing within 6 feet of the boiler was apparently uninjured. Fig. 1 shows the wrecked building and the brick chimney.

Everything was torn from the heads and the front head was blown about 150 feet, landing in a field with the water column and glass unbroken. The pipe leading to the mud drum was broken in two, one-half attached to the boiler and the other half remaining on the ground

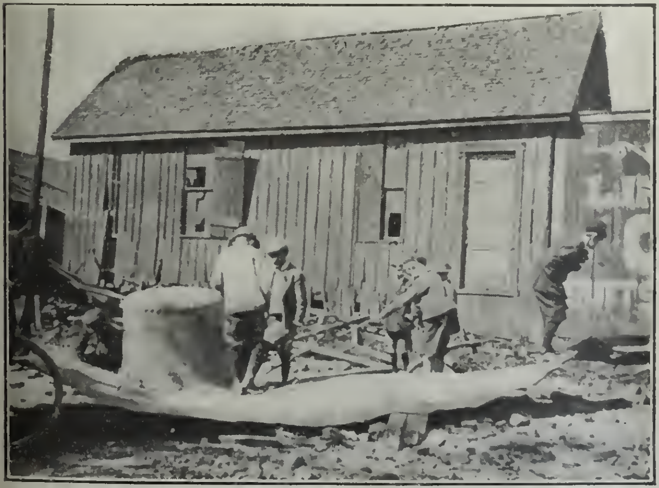

Fig. 2. Boiler Dome and Top Sheet

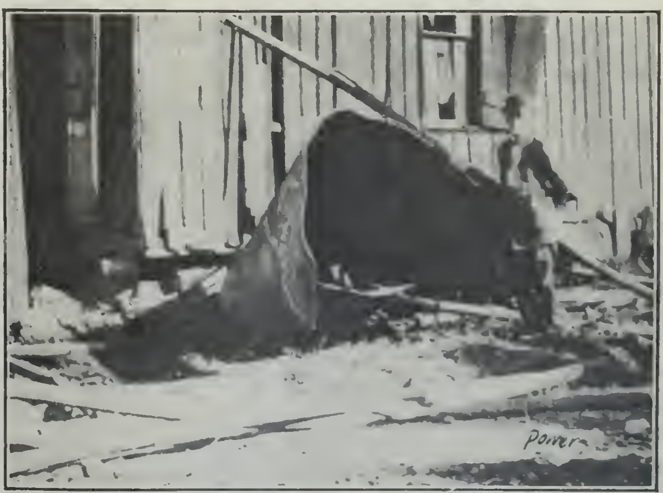

Fig. 3. Head and Part of Shell Blown 150 Feet

with the drum. An examination of the circumstances showed no signs of low water and the boiler was apparently in good condition except that the sheets were badly crystallized in places. The seams appear to have been no weaker than the sheet, as the rupture in some places was parallel and a few inches from a seam, while in others, such as that shown in Fig. 2, it crossed the seam. In twenty-nine years some serious defect must surely have developed.

A remarkable circumstance connected with the explosion was that, in spite of the fact that there were twenty men in close proximity to the boiler, none were killed and only one was seriously injured, although many had to crawl from under the wreckage after the explosion. Several narrow escapes are related; besides the engineer having just left the room, the boiler-room attendant had stepped to the side door to throw on a belt when the explosion occurred.

The Cost of Heat, Light and Power

By E. F. Tweedy

A discussion of Mr. Ripley's article which appeared in the October 25 issue. Evidence is offered to show that, at least in one instance, his article is extremely partial to the isolated plant.

An article with the above title appeared in Power and The Engineer under the date of October 25. A footnote stated that the article was "From an address to be delivered before the Blue Room Engineering Society of Greater New York," which is understood to be a fraternal organization composed for the most part of operating engineers. The entire subject is presented in such a manner as to invite criticism from any person who is possessed of accurate knowledge bearing upon the cost of heat, light and power in city buildings.

I will make an analysis of certain of the figures given in the article. As almost any one of the buildings considered will serve equally well, I will confine myself to the first one presented, the Terminal building located at 103 Park avenue, New York City. The cost of the electrical plant is given as $12,000. Some of the items which go to make up this total cost are presented. The cost of the engines is given as $4750, and that of the dynamos, $4000, making a total of $8750 for three generating units of 75-kilowatt rating each. If this total is taken from $12,000, the sum of $3250—representing the cost of the boiler, flue and chimney capacity required for the generating plant *in excess* of that required for a heating plant, the cost of high-pressure and exhaust-steam piping and plant auxiliaries together with the cost of the plant switchboard and wiring—is an absurdly inadequate amount.

If a boiler plant had been designed for heating purposes only, in connection with the use of purchased electrical energy, two boilers of 100-horsepower capacity each (one acting as a reserve) would have been amply sufficient. As the present boiler equipment comprises two units of 175-horsepower capacity each, or a total capacity of 350 horsepower, it is evident that 150 horsepower is directly chargeable to the generating plant. With the several omitted items properly charged to the generating plant, the cost would be as follows:

Cost of additional boiler, stack and flue capacity required by the generating plant	$ 2,000
High-pressure and exhaust-steam piping and plant auxiliaries	4,000
Plant switchboard and wiring	2,000
Engines (as given)	4,750
Dynamos (as given)	4,000
	$ 16,750

This total agrees very closely with the figure of $17,000 which was given by Mr. Ripley in a paper which he read before last year's convention of building managers as representing the cost of the electric plant in the same building, and included "the cost of one boiler and such part of the steam work and electrical work as would not have been necessary if no plant had been installed."

With interest at 5 per cent. and with depreciation taken at 7 per cent.—a very conservative figure, when it is realized that the life of a generating plant is almost invariably terminated by economic considerations long before a condition of physical decrepitude is reached—the yearly fixed charges (with no allowance for taxes or insurance—either fire or employer's liability) amount to $2010 instead of $1200 as given in the article under consideration.

Coming to the item of labor, it would appear that the salary of the "chief engineer and building manager" had been overlooked on the plant side of the balance sheet, although it was conveniently remembered when estimating the cost "if electricity had been purchased." If we refer to an article by the same author which appeared in a small monthly publication devoted to the interests of a certain supervision company, we find a statement of operating costs of this same plant in which the labor item is given in more detail. The following is extracted from the article in question:

The principal items in the pay roll are:

Assistant engineer	$80
Night engineer	80
Fireman	60
Chief engineer*	25

It thus appears that the yearly pay roll at that time, when the output of the plant was somewhat less than it is at present, was $2640 a year, exclusive of the salary of the chief engineer. Adding his salary of $1500 a year, the total yearly cost of labor becomes practically $4200, or about $1000 a year more than the figure of $3217 which is now given as the total yearly plant wages. The amount of labor, moreover, is inadequate for a plant of this size and for a building of this character. A night fireman, at least, should be added, so that the night engineer could be relieved of the work of attending to the boilers and left free to properly clean and make any necessary repairs and adjustments upon the engines and generators.

In the article from which the above extract is taken, it is clearly stated that the engineer and building manager received only $1200 a year when the Edison service was in use, although in "the estimated cost, if electricity had been purchased" this item appears as $1500.

The cost of fuel for heating this building, with electricity purchased, is estimated in the paper at $2000 a year, wit the use of No. 1 buckwheat coal, costin $3.35 per ton. In other words, it is estimated that this building would require for heating purposes alone, approximatel 600 tons of No. 1 buckwheat coal pe year. If a building of this size, and hav ing the wall and glass exposures of thi one, should require a yearly consumptio of buckwheat coal—or even of "screenings"—in excess of 400 tons for heatin alone, it would be evident that somethin was at fault with the plant management It is worth while to note in this connec tion that the author of the article ha seen fit to burden the operation of th heating plant with a fuel costing $3.3 per ton as compared with a fuel costin $2.22 per ton for the generating plant Of data showing the yearly consumptio of coal for heating in twenty office an loft buildings in New York City as com pared with the volumes of these build ings, the figures relating to the larges building, with a gross volume of 1,800,00 cubic feet, or nearly one-third larger tha that of the Terminal building and with all sides exposed, show that the yearly consumption of coal for heating is only 450 tons.

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*This latter item comes about in the following manner: The superintendent of the building when the Edison service was used was receiving $100 per month. When the isolated plant was installed his salary was increased $25 per month and he now performs the double duty of superintendent and chief engineer; therefore, it is obvious that the e pense for chief engineer is only $25 per mont for the isolated plant. The $100 in payme for his service as superintendent of buildin assisting in renting the offices, attending t complaints of tenants and otherwise directin the scrubbing force, elevator service, etc.

The cost of water for the boilers chargeable to the plant should be $200 instead of $100 as given in the article. The combined cost of repairs, supplies, removal of ashes, etc., is given as $851. It is very evident that the present yearly cost of repairs does not fairly represent what the average yearly repairs will be during the life of the plant. This plant is now comparatively new, having been in operation for some two or three years only.

In his paper before last year's building managers' convention, the author stated that the average monthly repairs in this same plant amounted to $10; while in the article above referred to, the total yearly cost of repairs upon the plant was given as $98.

After making the various corrections outlined above, the yearly operating costs, with and without a generating plant, are found to be as follows:

GENERATING PLANT.

Coal	$ 3,098
Wages	4,140
Water for boilers	200
Repairs, oil, ash removal (in excess of heating plant) and miscellaneous (not including lamps)	851
Total	$ 8,289
Less sale of electricity	400
	$ 7,889
Add 12% of plant cost	2,010
Net cost	$ 9,899

PURCHASED ELECTRICITY.

Coal 400 tons @ $2.22	$ 900
Fireman 8 months	400
Engineer and building manager	1,200
Electricity 156,340 kw.-hours @ 5 cents per kw.-hour	7,817
Care of elevators, heat and plumbing and wiring repairs (estimated)	300
Oil and grease	50
Total	$ 10,667
Less sale of electricity	400
	$ 10,267

The above estimates, in their corrected forms, show an apparent saving of about $360 a year by the operation of a private generating plant—quite a different matter from the $3600 which is claimed in the article and paper referred to. As a matter of fact, if this plant were properly designed and operated, the yearly cost of its operation would be considerably in excess of the cost with electricity purchased, as some of the following considerations should make clear.

No storage battery has been provided in connection with this plant, although the need for one is fully as great as in the other buildings mentioned in the paper where batteries are provided. The effect of the elevators upon the lighting service is very noticeable to anyone who has occasion to visit this building. With a storage battery of proper capacity installed, the additional fixed charges, plus the cost of battery renewals, would amount to at least $1000 a year, and this amount should be added to the plant operating cost as given above.

A building of this character should be provided with what is known as a "break-down" connection to provide against the disability of the entire plant by a bursting steam main or engine flywheel, or due to a failure in the water supply. The generating plants of several prominent buildings in New York City have been entirely crippled from one or another of the foregoing causes within the past few years, and means should be provided for operating at least one elevator and a comparatively small number of lights in the event of an emergency of this kind.

As mentioned above, the amount of plant labor is inadequate at the present time and a night fireman should be added to the present staff, which would increase the labor item by some $700 a year. The owner of electrical generating apparatus —or of productive machinery of any kind which requires skilled attention for its economical operation—can effect no saving of a permanent nature through the employment of labor which is either underpaid or insufficient in amount. In the operation of an electrical generating plant it is particularly true that "cheap labor is dear at any price."

No allowance has been made for the items of taxes and insurance upon the plant valuation. Whether employer's liability insurance is carried or not, the employment of labor in connection with a generating plant involves a risk which is a distinct liability and which should be capitalized as such. Since the recent amendment of the employers' liability law by the legislature of the State of New York, the rates for liability insurance for electric-power plant operatives have been materially increased. This question is now one of considerable importance to the owner of electrical generating machinery.

While it would be difficult to state the exact money value of the space which the generating plant occupies, it should be evident, at least to those who have a knowledge of building rentals, that this space—located within a building such as the one under consideration—is possessed of some earning value, the amount of which should be directly charged against the operation of the plant.

If such items as are given above are included—items which are generally belittled or entirely ignored by those who are interested in the installation and operation of private generating plants, but which are nevertheless real, it will be found that the net yearly cost of operating the generating plant is increased to an amount which is considerably in excess of what the yearly cost would be with electricity purchased.

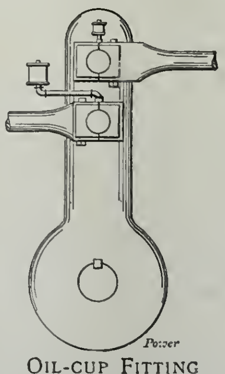
It is believed that this single analysis will show, with sufficient clearness, the tendency of the article as a whole. I have confined myself to a "dollars-and-cents" comparison solely, and have not considered in any way the many advantages which invariably accompany the use of purchased electricity—advantages which are now very generally recognized by the great majority of building owners and building managers. If this latter statement requires any confirmation, it is to be found in the fact that the owners of over 96 per cent. of all the buildings erected in New York City during the year 1909 found it to their advantage to adopt the central-station service for their supply of electric light and power.

Convenient Oiling Device

More than one oil-can spout has been jammed and battered when the engineer or oiler attempted to lubricate some particular bearing or rod brasses located in restricted places.

The illustration shows how an engineer placed an oiling device on the strap end of a rocker-arm connection which was located immediately below the strap end of a second rod attached to the same arm.

A short ⅛-inch nipple was screwed into the strap of the lower rod and an elbow screwed onto it. A straight piece of ⅛-inch pipe was screwed into the elbow and extended horizontally out past the end of the top strap. An elbow supporting an oil cup was screwed onto the horizontal pipe. Now there is no danger of getting the spout of the oil can caught and the brasses have a constant supply of oil.



Oil-cup Fitting

Old Hal Mossback's wife persented him with twins tother day. Hal sed thet he'd allus perferred a cross-compound t' either a twin er a tandem, but gessed he'd have t' stand fer jist what he'd got.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Stack Temperature Costs

Two plants burn 100 tons of coal each per day. In one plant the temperature of the waste gases is 450 degrees and in the other the temperature is 200 degrees. What is the saving in tons of coal per day in the second plant over the first?

C. C.

In the first plant each pound of air is raised

$$450 - 200 = 250$$

degrees Fahrenheit over that in the other plant.

The heat required to raise one pound of air one degree Fahrenheit is 0.2375 B.t.u. Therefore, each pound of air would require

$$250 \times 0.2375 = 59.375 \textit{ B.t.u.}$$

For good combustion 18 pounds of air are allowed per pound of coal. Hence 100 tons of coal would require

$$200{,}000 \times 18 = 3{,}600{,}000 \textit{ pounds}$$

of air and this raised 250 degrees would require

$$3{,}600{,}000 \times 59.375 = 213{,}750{,}000 \textit{ B t u.}$$

Assuming that one pound of coal contained 14,000 B.t.u., this would represent

$$213{,}750{,}000 \div 14{,}000 = 15{,}268 \textit{ pounds}$$

of coal. which is equivalent to

$$15{,}268 \div 2000 = 7.634 \textit{ tons.}$$

This takes into consideration only the heat required to raise the air through 250 degrees. If the coal contains 10 per cent. of ash then 90 per cent. of the coal must go up the chimney as gas, and as the weight of this gas is approximately 1/19 of the total weight going up the chimney 5.26 per cent. should be added to the 7.634 tons required to heat the air.

$$7.634 \times 1.526 = 11.64 \textit{ tons}$$

more coal per day are required in one plant than in the other, other things being equal.

Steam Pressure for Given Horsepower

What boiler pressure will be required to develop 20 horsepower in a slide-valve engine, 12x14 inches, running 250 revolutions per minute, cutting off at three-quarters stroke?

P. G. H.

The horsepower of an engine is the product of the mean effective pressure, the piston area and the number of feet it travels per minute divided by 33,000. In this case, as the horsepower is given, the mean effective pressure which will develop one horsepower must be found. Take the formula

$$\frac{P\,A\,S}{33{,}000} = \textit{Horsepower}$$

in which

P = Mean effective pressure per square inch of piston area;
A = Area of piston in square inches;
S = Number of feet of piston travel per minute.

Substituting the known factors it reads

$$\frac{P \times 113.10 \times 583.3}{33{,}000} = 20$$

which transposed becomes

$$P = \frac{20 \times 33{,}000}{113.10 \times 583.33} = 10 \textit{ pounds}$$

the mean effective pressure required per square inch of piston area to develop 20 horsepower in a 12x14-inch engine running 250 revolutions per minute.

The mean effective pressure being known the initial may be found by transposing the formula for finding the mean pressure for a given ratio of expansion. which is

$$mp = p \times \left(\frac{1 + \textit{hyp. log. of } R}{R}\right)$$

in which

$m.p.$ = Mean effective pressure plus 14.7;
p = Absolute initial pressure;
R = Ratio of expansion.

Substituting,

$$24.7 = \frac{p \times 1.2852}{1.33}$$

$$p = 25.6$$

$$25.6 - 14.7 = 10.9 \textit{ pounds,}$$

the initial pressure in the cylinder necessary to develop 20 horsepower under the conditions named. If the initial pressure in the cylinder is 80 per cent. of the boiler pressure, the boiler pressure per square inch required is

$$\frac{26.5}{80} - 14.7 = 17.3 \textit{ pounds}$$

Starting Single Acting Engine

Will a two-cylinder, single-acting, 100-horsepower engine, with cranks set at 90 degrees, always start, without reference to the position of the cranks?

S. S. E.

No, for during a portion of each revolution the pistons of both cylinders are moving toward the head ends of the cylinders with the exhaust valves open. It is plain that if the engine comes to rest within this portion of the revolution it cannot be started by simply admitting steam to the valve chest. It will start only when the steam valve is open and the exhaust closed on one of the cylinders.

Demagnetizing a Watch

How can I demagnetize a watch?

D. A. W.

Suspend it at the end of a twisted string near a strong magnet and as it revolves with the untwisting of the string move it slowly from the vicinity of the magnet. The operation should be repeated several times. Or it may be wrapped in moisture-proof paper and buried in the earth for a few days.

Brake and Indicated Horsepower

What is the difference between the indicated and the brake horsepower of an engine?

B. I. H.

The indicated horsepower of an engine is the work done in the cylinder. The brake horsepower is the work delivered at the rim of the flywheel. It is the work done in the cylinder minus the friction of the engine, and is sometimes called the net horsepower.

Vibrating Steam Pipe

The steam main leading from the boiler room to the engine, 60 feet away, vibrates badly. It seems to keep step with the dashpots of the engine. What is the cause and what is the remedy?

V. S. P.

The vibration is caused by the stopping of the flow of a long column of steam every time the valves on the engine close, the momentum of the moving steam being sufficient to start the pipe in motion.

It may be stopped by securely anchoring the pipe to something which will hold it rigid; by throttling at either end of the pipe so that the flow of steam through the pipe will be constant instead of intermittent and by a receiver or drum of about four times the volumetric capacity of the engine cylinder placed as near the throttle valve as possible.

Reasons for Giving Lead

Why is lead given when setting engine valves?

R. G. L.

Lead is given for the purpose of getting the valve open sufficiently to supply full initial pressure on the piston at the beginning of the stroke.

Power and The Engineer

Editorial

Issued Weekly by the
Hill Publishing Company
John A. Hill, Pres. and Treas. Rob't McKean, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of Power solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "Powpub," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 we printed and circulated 1,875,000 copies of Power.

Our circulation for October, 1910, was 132,000.

November 1..	33,000
November 8.....	33,000
November 15........ .	33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Verdict on Brooklyn Boiler Explosion

At the coroner's inquest into the cause of the deaths resulting from the recent boiler explosion at the works of the American Manufacturing Company at Greenpoint, Brooklyn, the jury arrived at a verdict holding the American Manufacturing Company guilty of negligence in operating a boiler which was in an unsafe condition.

In several recent cases of this kind the juries have either placed the blame upon someone killed in the accident or have whitewashed the whole affair. Therefore, when we find an exception to this rule—an instance in which the jury has endeavored faithfully to discharge its duty—we hesitate to offer any criticism. However, in the present case much of the evidence was extremely vague and many important points were either passed over lightly or not brought up at all.

The inspector for the insurance company testified that he had made a careful inspection, both internal and external, a year previous to the explosion and had found the boiler to be in a safe condition, his inspection having consisted of the hammer test in addition to looking over every part of the boiler. He claimed that the sound from the striking of the hammer would have revealed any dangerous condition of the head in the section at which the rupture occurred—*about two inches from the flange.* Furthermore, the fact was brought out that there was only three inches clearance between the bottom of the head and the base

It was shown and the fact emphasized, that the owners had been aware of pitting going on within the boiler and had taken steps to arrest this action; but from an inspection of the bottom head after the explosion it was plainly evident that this pitting was not responsible for the failure, which was due to external corrosion.

A point which would have been of utmost importance in fixing the responsibility but which was not mentioned, was the fact that a hole had been drilled and apparently plugged (the plug was probably blown out during the explosion) at the center of the bottom head. The question naturally arises—when and by whom was this hole drilled? Its presence certainly afforded a means for determining the thickness of the metal at the center of the head, which, although not as thin as at the section of rupture, nevertheless indicated the dangerous condition of the head.

The experts were unwilling to say whether, in their opinions, the plate had corroded to such an extent within the period between the last inspection and the explosion.

The only evidence which appeared to connect the American Manufacturing Company directly with the cause of the disaster was their knowledge of the pitting action going on *within* the boiler and the fact that the chief engineer had been within the boiler five weeks previous to the explosion and had reported everything all right. On the other hand, the company seems to have complied with the requirements of the law, insofar as having the boiler inspected and subjected to a hydrostatic test.

It is unfortunate that in cases of this kind the coroner is usually handicapped through unfamiliarity with the subject in hand, and as a result much important evidence never reaches the jury.

The Fireman's Task

It is beginning to be recognized that the position of fireman in the power plant is one of utmost importance. It does not matter whether the plant is of good or bad design. It does not matter whether the piping is well covered and free from leaks or bare and leaking at every joint. It does not matter whether the engines are of the best design and carry economical loads or are "steam eaters," with loads which range from friction to the limits of possible power. But there is one factor in the cost of the steam and the power derived from it that is more important than all of the others. This is the fireman.

If a change is made from simple to compound engines a saving of twenty per cent. may be effected, but by careless firing and neglect of proper handling of the feed water twenty-five to forty per cent. of the heat value of the fuel may be wasted.

If an efficient economizer is bought, it is for the purpose of extracting a few of the escaping heat units from the gases as they go up the chimney. But if an inefficient fireman puts so much air through the fire that the temperature of the gases is so reduced that the economizer has nothing to do, then this state is worse than before the economizer was installed.

Throughout the entire list of power-plant apparatus not an auxiliary nor an appliance can be named the beneficial effects of which cannot be nullified by an ignorant or indifferent fireman. Wherever possible, machinery is being installed with the intention of rendering human skill unnecessary and the myriad forms in which the automatic stoker appears is only one of the ways in which this trend is shown. But the stoker with the others falls far short of its possibilities in the hands of the man who will not, or cannot, be a fireman in all that the name implies.

To be this he must know how to burn coal in such a way as to get from it the last available heat unit. He must know how to so handle the boilers that the greatest possible percentage of the heat from the fuel will be transmitted to the water. He must know how to treat feed-waters with their varying scale making content. He must understand the principles of furnace construction and operation. He must not only know all these things but must be able to use this knowledge. Such a man is a fireman and can go into any boiler room and exercise a greater influence on the efficiency of the plant than any man, however well equipped in any other department.

That such men are available and are factors of importance is being realized by power-plant managers everywhere, and the man who knows and can produce results in the boiler room is coming into his own.

Wages and Worth

Notwithstanding the agitation relative to the conservation of the national resources and all the advice handed out gratis to users of the world's natural products, there are those who still persist in putting men to work in the boiler room who are offered little encouragement to give much consideration to conservation.

What fireman working for a dollar and a half per day is going to lie awake nights devising ways and means of reducing the operating expenses of the boiler room? And, furthermore, how many employers are moved to substantially recognize the fireman in a financial way when he does cut down the coal bill?

Plant owners naturally maintain that the fireman is paid to look after their interests, but if this is so, it would appear that a low valuation is placed upon the interests.

As a matter of fact a fireman is looked upon by many plant owners as a man with plenty of muscle, capable of shoveling ten or more tons of coal per day and of handling the heaviest slice bar. The question as to whether the man can handle boiler furnaces with the intelligence necessary to produce the best results with the coal available is seldom considered.

For instance, the chief engineer of a certain power plant observed that one of his firemen was more capable and showed more interest in his work than any of the others. He, therefore, recommended that the company increase his wages, which was done. After a time the other firemen learned of the increase and went in a body to the chief engineer to know why one of their number was receiving more pay. The reply was that the fireman was worth more than any of the others.

These men then appealed to the manager for an increase in wage equal to their more capable companion, and the manager, being a just but misguided man, granted the increase, saying that one fireman doing the same kind of work was worth as much to the company as another.

It is safe to say that the poorer firemen were not overpaid even after their wages were raised; but if their less effective services were worth the higher price the more intelligent and effective service of the better man were worth still more. If such service is to obtain no recognition where is the encouragement to render it except in the fact that one owes it to himself and to his career to do the best he can in any position, and is the better man and probably the more successful man, in the long run, for so doing.

Of two men doing the same kind of work one is not always worth as much to the firm as the other. It is not the amount of work a man does that counts; it is the way he accomplishes what he sets out to do. Firing ten or twelve tons of coal may be a big day's work, but the man who fires that amount of coal into a furnace and keeps the steam pressure right on the dot is not worth as much to the company as the man who takes things easier and keeps the steam pressure on the dot by burning but eight tons of coal a day. He does not work as hard as the other man, but he does get better results, and this is what should count with the man who pays the bills.

Engineers are handicapped in much the same way. There seems to be an undervaluation of the real worth of good engineers. The salaries of those who are capable of holding good positions are based largely on what it is necessary to pay low-grade men. Sixteen dollars per week appears to be a common wage for putting in from ten to twelve hours each day, doing repair work on Sundays and holidays, besides working overtime when it is necessary.

The employer of the unskilled engineer will take the ground that he pays his man sixteen dollars per week, and that he is expected to keep the plant running. If he has to work overtime, Sundays and holidays, that is his fault. Apparently the plant owner does not take the trouble to ascertain how much these hours of extra repair labor are costing him.

An efficient engineer will have but little overtime work because he will see that repairs are made as soon as the defects are discovered, so that he always has the machinery in the plant in first-class condition. The owner does not stop to make a comparison of the two classes of workmen. He does not stop to think that his engineer is worth more to him than the other would be. The capable engineer may not work as hard as the other, but he is worth more to any plant owner because he is capable and saves expenses in operation and repairs.

An engineer or fireman should receive the amount of wage he is worth, based on his capabilities, and not on what a low-grade man in the same field would get.

The Institute of Operating Engineers

The provisional committee of this organization is getting out a prospectus containing a suggestion for a constitution and directions for the formation and conduct of local branches. The hope is that operating engineers in various cities who are anxious to obtain an engineering education by following the courses laid out by the Institute and to secure its certificate or degree will associate themselves for that purpose and commence work under the rules here suggested. When a sufficient number of such branches have been formed and have had some experience with the work, a convention can be held with representatives of such associations from all over the country and the constitution formally adopted in its suggested form or revised as indicated by experience.

Copies of the prospectus may be had by addressing M. L. Rice, secretary, Institute of Operating Engineers, Engineering Societies building, 29 West Thirty-ninth street, New York City.

Guarding the Coal Pile

In Europe it is not uncommon to see a huge pile of coal held up by a retaining wall built of the larger lumps or of briquets. The upper edges of this wall are neatly whitewashed, a fact which we laid at first to the universal striving for artistic effect. It developed, however, that the whitewash has a more utilitarian purpose. It is impossible to take a piece of coal out of the pile anywhere within reach without breaking that white wall or line, and petty thieving is discouraged by the tell-tale black gap which it leaves.

What have you learned today?

A fire-room job is dirty work, but it earns clean money.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

The Parson Distributor

The standard practice in automatic stoking in America has been to introduce the fuel in a single coking line at some part of the furnace and to advance it through the furnace as it burned, rejecting the ash at another point or points. For years there have been in common use abroad stokers which throw the coal continuously or at short intervals over the fire bed, preserving a fire upon an ordinary flat grate just as though it were hand fired but with a constant influx of fuel in proportion to the demand and without the disturbance of conditions favorable to combustion and the chilling of the furnace due to opening the firing doors. The firing doors remain intact and accessible, nevertheless, and may be used for barring, slicing and cleaning, and even for stoking if the automatic stoker gives out. Furthermore, any grate adapted to the fuel may be used and this may be of the shaking and dumping variety if desired.

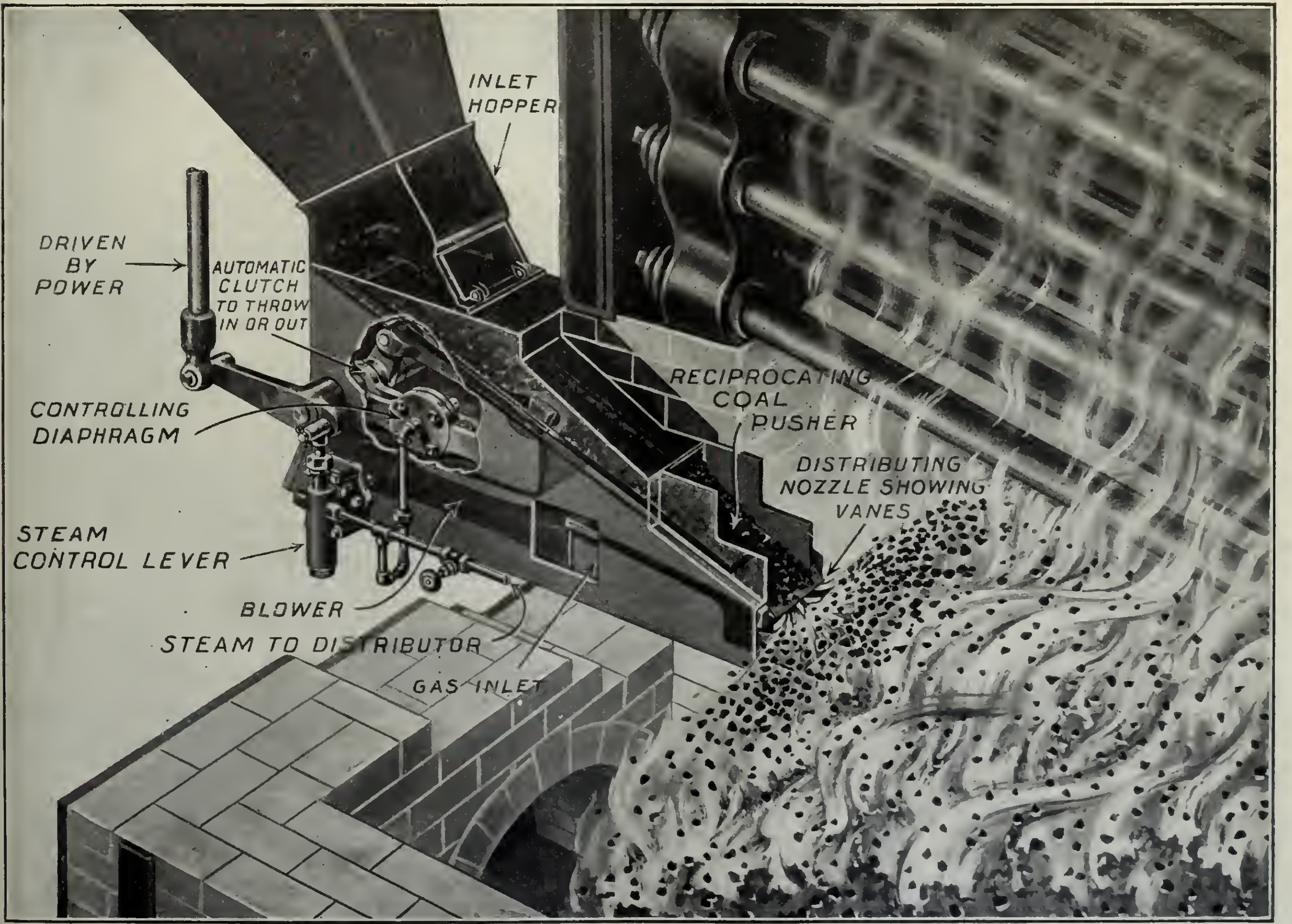


Fig. 1. The Parson Distributor in Action

In all of these stokers hitherto described in Power, the fuel has been projected into the furnace by rotating or vibrating blades or reciprocating plungers. In that under review it is blown in by an intermittent jet of air and steam. Referring to the engraving, the coal settles down from the hopper upon the reciprocating sliding plate which carries it forward and spills it over the edges of the projecting spout. Simultaneously a valve is opened admitting steam into an air blast which is frayed at the point of admission to the furnace by the divergent guides shown so that as the coal falls into it it is scattered in all directions over the fire. The opening of the steam valve is furthermore controlled so as to make one blast of sufficient intensity to deliver the coal to the far end of the furnace, the next to the middle distance and the next to the front end. As the intermittent injection of so much cold air as would be necessary to effect this would disturb combustion conditions and unduly cool the furnace, the major portion of the blast is drawn from the furnace itself through the gas duct shown. The portions of the apparatus which project into the furnace are protected to a much greater extent than the engraving would imply, by firebrick. There is no mechanism exposed to the heat and no surfaces subject to erosion by impact with the fuel. The blower can be removed and replaced in a few moments while the furnace is in operation and as the stoker is entirely free from the fire doors the operation of the boiler would suffer no interruption in case of its disablement.

These stokers, as manufactured by the Parson Manufacturing Company, 149 Broadway, New York, will handle some 2500 pounds of coal per hour and distribute it over a grate surface 10 feet in depth and 9 feet wide. For wider furnaces or higher rates of combustion, multiples may be used. They are adapted to burning all grades of bituminous and anthracite coals, coke and refuse fuels, which they will handle satisfactorily in sizes up to one-inch cubes; and the apparatus uses, its makers say, not over one-third of a horsepower, including the steam for the jets when running at its full capacity.

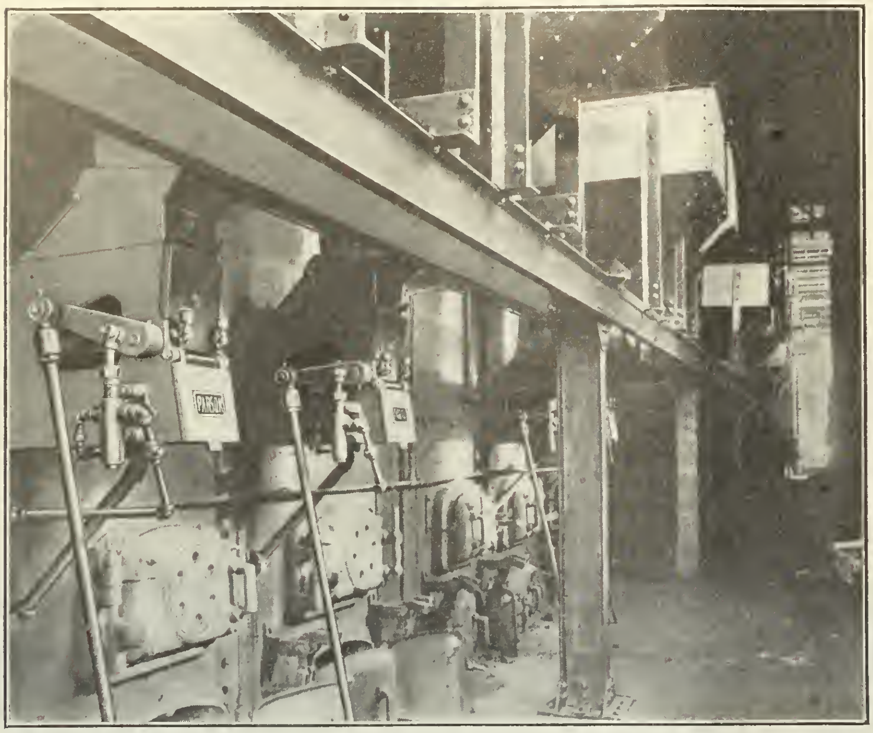


Fig. 2. An Installation of Parson Coal Distributors

Garrett Combination Feed Water Heater

This combination exhaust muffler, oil separator, feed-water heater and purifier consists of a shell made either of steel or cast iron, as may be required. When steel is used, cast-iron rings are riveted on each end, to which cast-iron covers are bolted, as shown in the illustration. These covers can be readily removed and either end opened for cleaning or repairs. The shell is divided into two parts by a partition, giving about two-thirds of the space for extracting the oil from the steam and one-third for filtering and heating the water; the structure is horizontal, requiring but little head room and is supported on cast-iron cradles having pipe extension so that the hight can be varied to suit conditions.

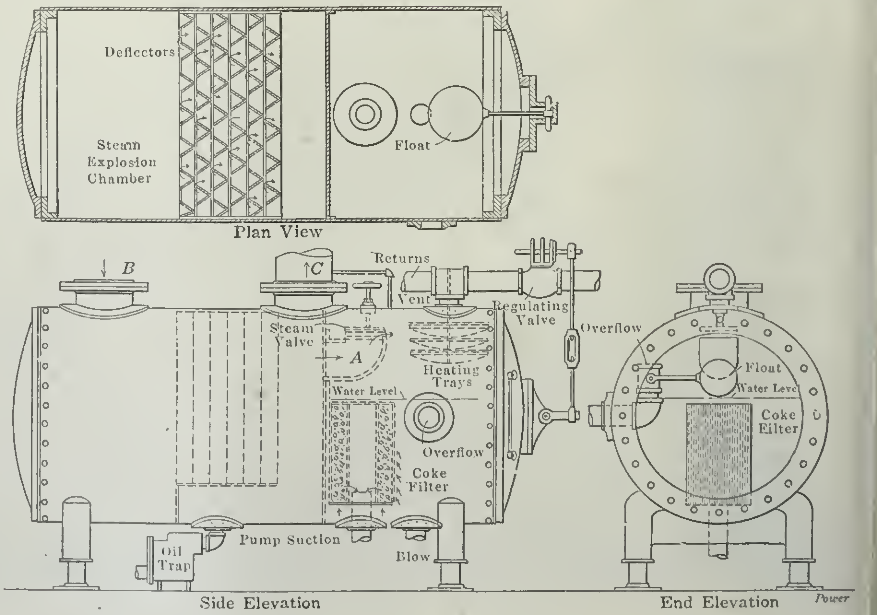
A large area is provided for all separation as it is necessary to reduce the velocity of the incoming steam. The flow of the steam is turned at right angles, bringing it in contact with a number of baffle plates consisting of V-shaped gutters placed to break joints, as shown. As the oil cannot escape with the exhaust steam it flows down the gutters and is gathered in a separate oil trap.

All of the exhaust steam does not pass through the heaters, only such portion as the water can absorb, the balance going to the heating system in a very dry condition. On the outlet an elbow bypass valve *A* is provided which can be opened or closed by a wheel located on the top of the shell. This simple feature reduces the cost of installation and dispenses with the complication of piping and an expensive bypass valve. A balance valve is provided on the make-up water line and is operated by a copper-ball float to maintain a true water level. An overflow valve with a copper float is also provided, as shown in the end view, to dispose of any excess water.

The water filter consists of a cast-iron cylinder perforated on the side and bottom. It is filled with hard coke and provided with a removable cover. This device filters the make-up water.

The heating trays consist of a series of cast-iron perforated saucers and flat disks, suspended on a hook arrangement. A disk is placed between each pair of perforated saucers for the purpose of returning the water to the periphery of the saucer below it in order to secure a better distribution and maintain a shower effect as it falls through the steam. Returns from the heating system flow over the heating trays to be reheated.

The exhaust steam enters at *B* and flows into the expansion chamber, where it turns at right angles and comes in contact with the V-shaped baffle plates. The entrained oil and moisture flows downward out of the path of the steam to the oil trap. Sufficient steam to heat the water to the highest possible temperature flows through the bypass valve, the balance going to the outlet *C*, to the heating system, or wherever it may be required. The return flows over the heating trays with the make-up water and through the coke filter to the feed pump.



Combination Exhaust Muffler, Oil Separator, Feed-water Heater and Purifier

This filter is manufactured by W. D. Garrett & Co., 136 Liberty street, New York City.

Evening Technical Courses

The Education Department of the City of New York is giving free instruction in the evening trade schools in many branches in which the readers of Power are interested, including mechanical drawing, machine design, applied electricity, steam engineering, electric wiring and installation, industrial chemistry, applied physics, plumbing, machine-shop work, mathematics, etc. The school in the borough of Manhattan is the Stuyvesant at Fifteenth and Sixteenth streets, near First avenue; in Brooklyn, on Seventh avenue between Fourth and Ninth streets, and in Long Island City on Wilbur avenue and Academy street. Timothy Healey, president of the International Brotherhood of Stationary Firemen, has taken an active interest in impressing upon these courses the practical character which they have assumed and has encouraged many members of his own society and others to profit by the advantages offered.

SOCIETY NOTES

On December 1 to 3 the Practical Refrigerating Engineers' Association will hold its first annual convention at Shreveport, La. The constitution and bylaws of the new association will be submitted at this meeting. The object of the organization is educational and it should receive the earnest support of refrigerating engineers in the South.

PERSONAL

Byron E. Parks & Son, mechanical engineers, formerly of Detroit, Mich., announce the opening of offices at 541 Michigan Trust building, Grand Rapids, Mich., where they will continue their business as consulting mechanical engineers. Special attention will be given to the design and superintendence of construction of power plants, power-transmission systems, heating and ventilating systems, drying of materials, apprisals and tests with view of betterment of economies, etc.

Kenneth Seaver has been appointed chief engineer of the Harbison-Walker Refractories Company, Pittsburg, Penn.

NEW PUBLICATIONS

The Steam Engine Indicator. By F. R. Low. McGraw-Hill Book Company, New York and London. Cloth; 169 pages, 9x12 inches; illustrated. Price, $1.50.

This is the third edition, revised and enlarged, of a book which is addressed to the man who has an indicator and who wants to be able to work it correctly and to interpret its revelations aright. The first chapter tells what to look out for in selecting an indicator and how to take care of it. Reducing motions are then explained with such a fundamental and yet simple consideration of principles as will lead the user aright. There follows a chapter with detailed directions for the use and application of the instrument and one upon the diagram which is then picked apart and considered in separate chapters, each devoted to one of its lines or points.

Succeeding chapters treat of the measurement of and deductions from the diagram, Computing the Horsepower, Steam Consumption, etc., Diagrams from Compound Engines, Errors in the Diagram, concluding with a chapter upon the Measurement of Clearance.

A Pathfinder. By Dr. Edward Goodrich Acheson. Published by the Press Scrap Book, New York. One hundred and forty-three pages, 9x12 inches.

It is not often that a successful inventor takes time from the development and promulgation of his creations to put the story of his life upon record. In the volume before us, Dr. Edward Goodrich Acheson tells not only of his discovery of carborundum siloxicon, his synthetical production of graphite and its emulsion with oil and water as oil and aquadag, but gives a circumstantial account of his early struggles and experiences in the coal and oilfields, with Edison at Menlo Park, introducing the incandescent light into Europe, etc., with many references to his associations with well known pioneers in these fields. Detailed accounts of the formation of the Carborundum Company, the Acheson Graphite Company, the Acheson Oildag Company and other organizations with which the author is identified are given, together with lists of the clubs and societies to which he belongs, medals and honors conferred upon him, etc. The volume is an elegant example of the bookmakers' art, being printed upon heavy Old Stratford antique, hard, deckle-edged paper with a gilt top edge and limp ooze leather overhanging covers, presumably for private circulation.

Heat Engines. By John R. Allen and Joseph A. Bursley, 1910. McGraw-Hill Book Company, New York and London; 9x12; 288 pages; cloth; illustrated. Price, $3.

When a successful professor consents to present to the public in book form the notes which he has used in his classes, there ought to be a chance for the practical man, the home student, to get access to some of the college lore, especially if the professor is enough of an author to teach an unseen class.

The book under review is a formal presentation of the notes which have been used for a number of years at the University of Michigan, where Mr. Allen is professor and Mr. Bursley assistant professor of mechanical engineering. It is written as a textbook but its freedom from higher mathematics, except in the chapter upon Thermodynamics, and the generous number of practical examples render it an excellent book for self study.

The first chapter defines Heat and describes its properties. The second chapter treats of Elementary Thermodynamics. Somebody is going to write this story sometime in a way that the man with a limited mathematical outfit can understand it. There is then a chapter upon the Properties of Steam, teaching what the steam table means and how to use it. Succeeding chapters treat of Calorimeters and Mechanical Mixtures, Combustion and Fuels, Boilers, Boiler Auxiliaries, Steam Engines, Types and Details of Steam Engines, Testing of Steam Engines, Valve Gears, Governors, Compound Engines, Condensers and Air Pumps, Gas Engines, Details of Gas Engine Construction, Economy of Heat Engines. The chapter on turbines is introduced by the overworked pictures of the turbines of Hero and Branca, and several others of the engravings have a familiar look.

"The Volatile Matter of Coal" is the title of the first bulletin to be issued by the new Federal Bureau of Mines. The authors, Horace C. Porter and F. K. Ovitz, conducted their investigations at the Pittsburg station while it was under the Technologic Branch of the Geological Survey, the work being a continuation of the fuel investigations begun several years ago at the Louisiana Purchase Exposition, St. Louis, Mo. The results obtained at that plant showed that the work of determining the fuel values of the coals and lignites in the United States with a view to increasing efficiency in their utilization would be incomplete if it did not include systematic physical and chemical researches into the processes of combustion. Hence in their later investigations the authors carried on such researches, concentrating attention on those lines of inquiry which promised results of greatest economic importance. This bulletin is therefore a report on an investigation of the volatile matter in several typical coals—its composition and amount at different temperatures of volatilization. Quoted directly, the authors say:

The investigation has already shown that the volatile content of different coals differs greatly in character. The volatile matter of the younger coals found in the West includes a large proportion of carbon dioxide, carbon monoxide and water, and a correspondingly small proportion of hydrocarbons and tarry vapors. The older bituminous coals of the Appalachian region yield volatile matter containing large amounts of tarry vapors and hydrocarbons, difficult to burn completely without considerable excess of air and a high temperature. Coal of the Western type, moreover, gives up its volatile matter more easily at moderate and low temperatures than that of the other type. The volatile matter produced at medium and low temperatures is rich in higher hydrocarbons of the methane type, such as ethane and propane, which contain a larger portion of carbon than is present in methane.

These facts help to explain the difficulty of burning Pittsburg coal, for example, without smoke, the low efficiency usually obtained in burning high-volatile Western coals, the advantage of a preheated auxiliary air supply introduced over a fuel bed, and the advantage of a furnace and boiler setting adapted to the type of fuel used. They bear directly also on the question of steaming "capacity" of coal for locomotives, the designing and operation of gas producers for high-volatile fuels and the operation of coke ovens and gas retorts.

The results show further that certain bituminous coals of the interior and Rocky Mountain provinces give promise of good yields of byproducts of coking, notably ammonia and high-candlepower gas, comparing favorably in these respects with the high-grade coking coals of the Eastern province.

They show also that inert, noncombustible material is present in the volatile products of different kinds of coal to an extent ranging from 1 to 15 per cent. of the coal.

The bulletin will be of interest to fuel engineers and those engaged in the suppression of smoke. The bulletin may be obtained by applying to the director of the Bureau of Mines, Washington, D. C.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

APPARATUS FOR PRODUCING MOTIVE POWER. Hudson Maxim, New York, N. Y. 974,166.

WATER MOTOR. Louisa M. Wilber, Buffalo, N. Y. 974,221.

WAVE MOTOR. Theodore La Brum, San Diego, Cal. 974,282.

PRESSURE-FLUID ENGINE. Henry H. Mercer, Claremont, N. H., assignor to Sullivan Machinery Company, Claremont, N. H., a Corporation of Maine. 974,413.

EXPLOSIVE ENGINE. William J. Wright, Franklin, Penn. 974,455.

ELASTIC-FLUID TURBINE. Charles A. Backstrom, Philadelphia, Penn., assignor to International Power Generator Company, a Corporation of Delaware. 974,457.

ROTARY ENGINE. Malcolm L. Harris, Chicago, Ill. 974,485.

FLUID-PRESSURE MOTOR. Daniel Shaw Waugh, Denver, Colo., assignor to the Denver Rock Drill and Machinery Company, Denver, Colo. 974,543.

STEAM ENGINE. Bert C. Ball, Portland, Ore. 974,546.

INTERNAL COMBUSTION ENGINE. John W. Akerman, Newburyport, Mass. 974,544.

ROTARY TURBINE ENGINE. Joseph Blaeker, New York, N. Y. 974,634.

CURRENT MOTOR. Henry Allen, Joseph, Idaho. 974,735.

BOILERS, FURNACES AND GAS PRODUCERS

FURNACE. Eber C. Carter, Carrollton, Ill. 974.120.

SMOKE-CONSUMING FURNACE. George E. Wells, Dayton, Ohio. 974,322.

OIL BURNER. Rufus J. Wheeler and Henry R. Neil, Bakersfield, Cal. 974,324.

BURNER FOR COMBUSTION UNDER HIGH PRESSURE. Charles A. Backstrom, Philadelphia, Penn., assignor to International Power Generator Company, a Corporation of Delaware. 974,624.

OIL BURNER. Adolphus Henry Forstner, Portland, Ore. 974,652.

POWER PLANT AUXILIARIES AND APPLIANCES

TURBINE NOZZLE. Thomas Bell, Clydebank, Scotland, assignor to General Electric Company, a Corporation of New York. 974,108.

TURBINE GOVERNOR. Frederick W. Bentley, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 974,113.

VALVE. Henry F. Quickel, Harrisburg, Penn. 974,185.

REGULATOR FOR STEAM-GENERATING APPARATUS. Otto F. Persson, Lynn, Mass., assignor, by mesne assignments, to General Electric Company, a Corporation of New York. 974,181.

ROD PACKING. Thomas Smith, San Luis Potosi, Mexico. 974,197.

LUBRICATOR. Austin D. Williams, Roseville, Cal. 974,222.

ENGINE STOP. Frank Henneböhle, Chicago, Ill. 974,266.

BOILER-WATER GLASS SHIELD. George H. Sargent and Arthur G. Hollingshead, Chicago, Ill. 974,299.

VALVE FOR GAS ENGINES. Baxter M. Aslakson, Salem, Ohio. 974,337.

CENTRIFUGAL PUMP AND COMPRESSOR. Harry F. Benson, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 974,345.

PUMP, INJECTOR OR THE LIKE. Gerhard Joan Otto Doris Dirkers, Lonneke, near Hengelo, Netherlands. 974,366.

AUTOMATIC VALVE. John B. Perkins, Revere, Mass. 974,424.

ENGINE VALVE. Clifton Reeves, Trenton, N. J. 974,432.

TURBINE VANE AND WHEEL. Duston Kemble, Lakewood, Ohio. 974,496.

GLOBE VALVE. Arthur L. Kerbaugh, Allentown, Penn. 974,498.

PIPE UNION. Charles J. Luther, Elizabeth, N. J., assignor of one-half to Joseph C. Davis, Maplewood, N. J. 974,504.

PIPE JOINT. Claude Revaillot, Nice, France. 974,517.

GAGE COCK. Samuel Paxton, Washington, D. C. 974,515.

GASOLENE OR OIL FILTER. George T. Reynolds, Providence, R. I. 974,519.

STARTER FOR HYDROCARBON ENGINES. John B. Runner, Indianapolis, Ind., assignor to Leonard M. Quill and Louis A. Browne, Indianapolis, Ind. 974,525.

GASOLENE OR OIL FILTER. George T. Reynolds, Providence, R. I. 974,519.

PISTON VALVE. David F. Stayman, New York, N. Y. 974,534.

GAS-LIBERATING HEADER FOR STEAM CONDENSING COILS. William F. Paradise, Leavenworth, Kan. 974,590.

CONDENSER. Garhard Reimers, Milltown, N. J., assignor of one-half to Frederick A. Sondheimer, New York, N. Y. 974,598.

ENGINE GOVERNOR ATTACHMENT. William Hilton Giltner, Paris, Ky. 974,665.

PUMP DRIVING GEAR. James W. Myer, San Francisco, Cal. 974,694.

CRANK SHAFT AND BEARING THEREFOR. Emil A. Nelson, Detroit, Mich., assignor to Rotary Valve Motor Company, Detroit, Mich., a Corporation of Michigan. 974,697.

DIRECT-ACTING PUMP. Philip Francis Oddie, Wimbledon, near London, England. 974,698.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC-MOTOR CONTROL. Henry D. James, Pittsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 974,152.

CIRCUIT INTERRUPTER. Carl B. Auel, Wilkinsburg, Penn., and John R. Spurrler, Manchester, England, assignors to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 974,109.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power
and The Engineer

Vol. 32 NEW YORK, NOVEMBER 22, 1910 No. 47

A NUMBER of race horses are lined up before the barrier. One of them—and only one—will be the winner. All are as fit as their trainers possibly can make them.

The barrier is snapped up, the horses lunge into motion and the cry, "They're off!" announces that the race is on. The distance to be run is one and one-quarter miles.

The favorite, a famous "miler," takes the lead—his characteristic performance—and holds it until the seven-eighths pole is reached. Here, his right to the lead is challenged by two other horses. At the mile, all three are on even terms and each jockey "goes to the whip." Under the punishment of the lash one of the tired contenders swerves out and the cry goes up, "He's yellow!" The other contender gamely increases his speed and passes the favorite which, having run his distance, the mile, after a weak effort, quits in despair when the other horse passes him. A "long chance" has won.

The performances of these three animals are characteristic of three classes of men.

The favorite, which stopped at the end of his "distance," typifies that man who can do just so much and no more. In other words, the routine man. So long as he has a definite amount of work to do which requires only so much effort or initiative, this man gets along grandly. He does his "bit" willingly and well and is ready to go home when the whistle blows.

But, let an emergency arise when special effort is required or initiative action needed and this man thinks that he is "licked" before he even tries. And, like the horse, he "quits in despair" although, based on "past performance" and "form," he should win. Instead of saying "I will," he says "I can't." Having first convinced himself that he can't, it is small wonder that what feeble effort he does make is not crowned with success.

The horse which swerved to avoid the whip last symbolizes the human being whom we variously characterize as a "quitter," "sidestepper" or "fourflusher." Such a man deliberately tries to avoid his duty—to step from under, so to speak. He does not bother even to ask himself whether or not he should try to make good; he simply schemes to avoid being put to the test.

The winner represents the man who never admits that he is beaten, who battles on just as long as there is a single chance or scrap of hope left. Life teems with examples of success won by men who were badly handicapped.

Perseverance and bulldog grit explain their "luck."

The path to success is up a steep hill, laborious to climb. In places, the footholds are dangerously insecure. He who hesitates or is careless to make sure each step, often finds, all too late, that he is swiftly sliding back the distance which he had with great difficulty attained.

The adage, "All things come to him who waits," has long since been revised to, "All things come to him who *works* while he waits."

The Metropolitan Plant at Reading

By A. D. Blake

> *A central turbine station of 13,500 kilowatts capacity furnishing light and power to the city of Reading and the entire county of Berks. The equipment is uptodate in every respect and provision has been made to double the capacity of the plant.*

The Metropolitan Electric Company, of Reading, Penn., besides furnishing electricity for the lighting, street railways and commercial power of that city, also supplies power, through high-voltage transmission and substations, to a number of constituent companies which serve practically the whole of Berks county.

Formerly the main generating station for this system was located at Seventh street in the heart of the city and consisted of four buildings which had been added from time to time. The total output of the plant was about 7000 horsepower, the equipment including units varying in capacity from 800 to 60 kilowatts, and in the first building there was a line shaft driving twenty Edison bipolar machines. With the enlargement of the system and the consequent increased demand for power the need of a new central station became imperative.

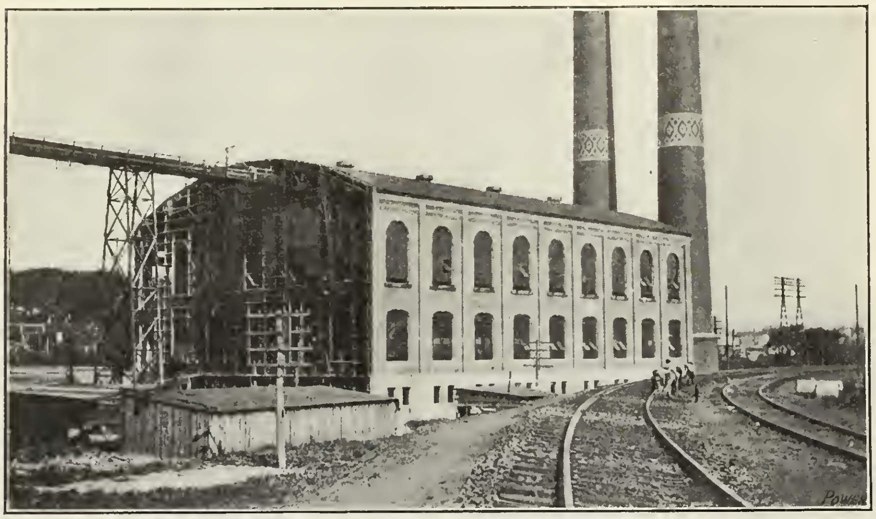


Fig. 1. Exterior View of Plant

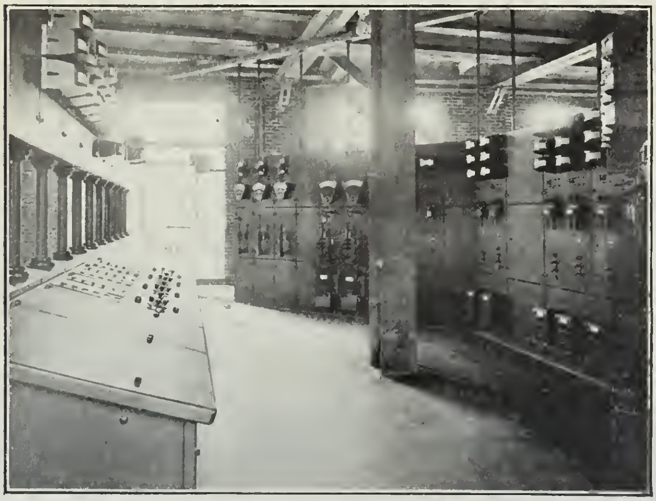


Fig. 2. Switchboards

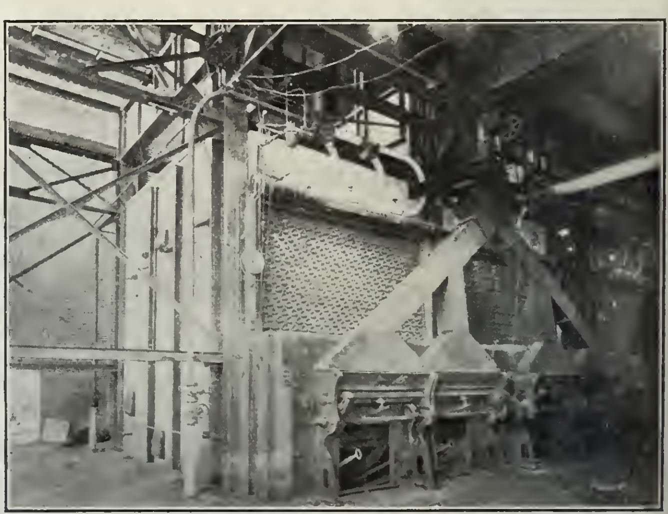


Fig. 3. Fronts of Boilers

The new power house, an exterior view of which is shown in Fig. 1, is located on the outskirts of the city, a few hundred feet from the Schuylkill river, and at a point adjacent to two railroads; hence, the problem of coal delivery and circulating-water supply has been greatly simplified. The building is of steel framing with self-supporting brick walls resting on concrete basement walls, and has a curved roof consisting of supporting trusses carrying a 3-inch reinforced-concrete slab which is covered with three-ply roofing felt. Owing to the irregular geological formation the foundations rest partly upon rock, partly upon solid ground and partly upon piling. The building has been laid out with a view to future extension, the intention being to ultimately double its present size; hence all bays have been built alike and when finally completed the structure will be entirely symmetrical with another pair of chimneys at the far end. Owing to the prospect of an extension having to be made in the near future a sheet-metal wall, stamped to represent brick, has been placed at the extension end of the building. This is bolted to the framework in sections and can be removed and reerected as often as desired.

Referring to the general plan, shown in Fig. 4, it will be seen that the main floor of the building proper is taken up by the boiler room and turbine room, separated by a dividing wall. The auxiliaries are located in the basement under the turbine room and the switching apparatus and substation are located in a wing structure; the latter expedient being made necessary by the shape of the available property. The basement floor is at a level which places it above all ordinary high-water marks of the river, but to protect the auxiliaries from any unusual spring freshets, the walls have been waterproofed to a hight of 8 feet from the floor.

Boiler Room and Coal-handling Outfit

There are sixteen 625-horsepower Edge Moor water-tube boilers set two in a battery and arranged in two rows. These furnish saturated steam to the turbines at 200 pounds gage. Two rectangular steel flues, one under each row, conduct the gases to two 242-foot radial brick stacks which are located outside the building.

Spurs from the railroads run over a track hopper which dumps direct to a crusher, from which the coal is carried on a 24-inch inclined belt conveyer to the power-house bunker. This bunker, which is of reinforced-concrete construction, has a capacity for 3900 tons. From the bunker the coal is delivered through chutes to the Wetzel automatic stokers with which the boilers are equipped. Each battery of boilers is supplied by one chute having a Y-connection and furnished with a weighing device. Fig. 3 is the front view of one of the boilers.

Ash hoppers are suspended from the floor beams under the boilers, and from these hoppers the ashes are dumped into motor cars (see Fig. 3) which convey them to a dump.

Main Generating Units

These consist of three 4500-kilowatt Allis-Chalmers turbo-generators running at 1800 revolutions per minute and delivering three-phase, 60-cycle, current at 13,200 volts. Each turbine has its individual oiling system consisting of a pump operated direct from the turbine and a supplementary steam-driven pump for use when starting up. Excitation current is furnished by two 100-kilowatt, 125-volt, horizontal units, each consisting of a General Electric Company generator driven by a Curtis turbine. A general view of the turbine room as seen from the switch gallery is shown in Fig. 6.

Auxiliaries

Each turbine exhausts into a Tomlinson barometric condenser located just back of the boilers. As will be seen by referring to Fig. 4 each condenser has two heads and two injection-water pipes. This arrangement was resorted to in order to avoid providing a greater clearance between the backs of the boilers and the wall, which would have been necessary with one large head. Each condenser is provided with its own circulating pump, which is of the centrifugal type driven by a 175-horsepower Kerr turbine, and by means of a main supply pipe and cutoff valves one pump may be made to supply another condenser if necessary. The circulating water is taken directly from the Schuylkill river through a 6x6-foot rectangular concrete flume running under the pump-room floor. On top of this is another flume of the same size into which the hotwells discharge through the openings in the foundation wall. The hotwells are placed so as not to be affected by any floods and this necessitated raising the exhaust-pipe connections above the condenser heads to the barometric distance above the possible high-water level.

The boiler feed water is taken from a dam in a creek at a point about a mile distant from the power house. The difference in head between these two points is so small that in order to make use of gravity it was necessary to avoid, as much as possible the loss in head due to pipe friction; this meant an unusually large supply pipe which would have entailed a great expense. Therefore, the pipe line was constructed of old boiler shells joined together by concrete. Three five-stage, turbine-driven Jeansville centrifugal pumps, each capable of delivering 700 gallons of water per minute at 260 pounds pressure, supply the boilers with feed water.

Advantage is taken of the fact that the Schuylkill river water is slightly acid, by providing connections whereby the boiler feed may be taken periodically from the circulating-water supply, thus

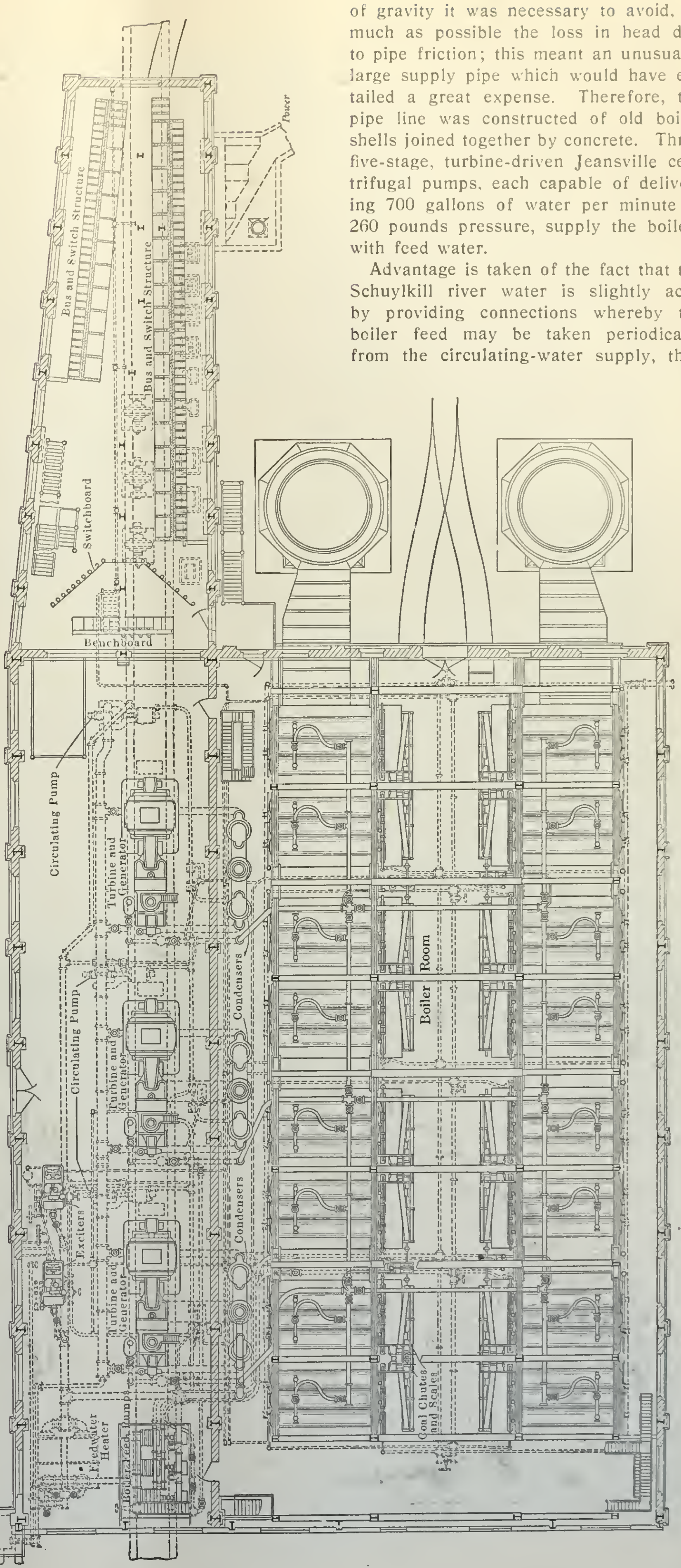


Fig. 4. General Plan of Plant

neutralizing the action of the regular supply. So far, this has been found to work admirably in preventing scale.

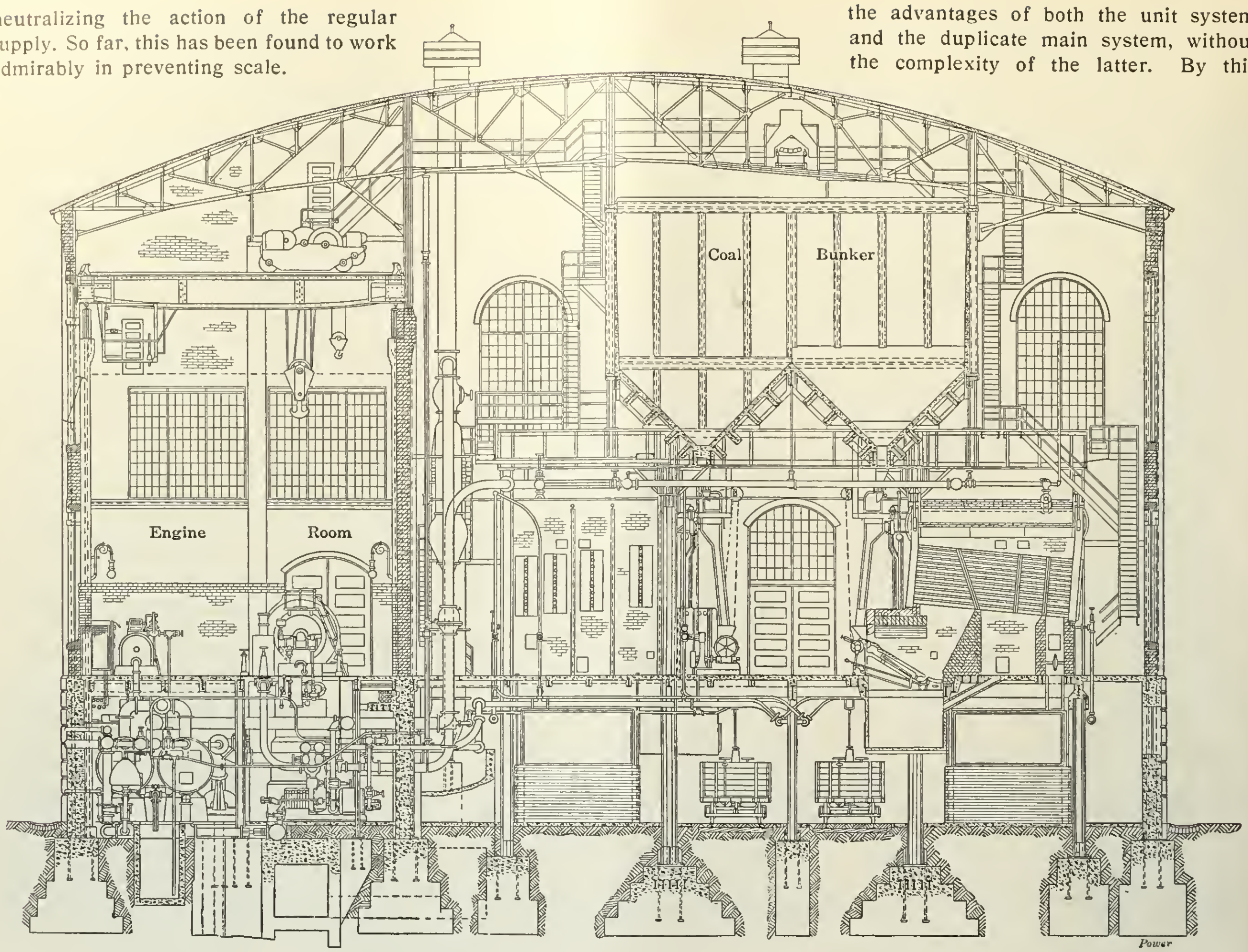


Fig. 5. Elevation through Engine and Boiler Rooms

The feed water is heated by two Cochrane heaters which utilize the exhaust steam from the auxiliaries.

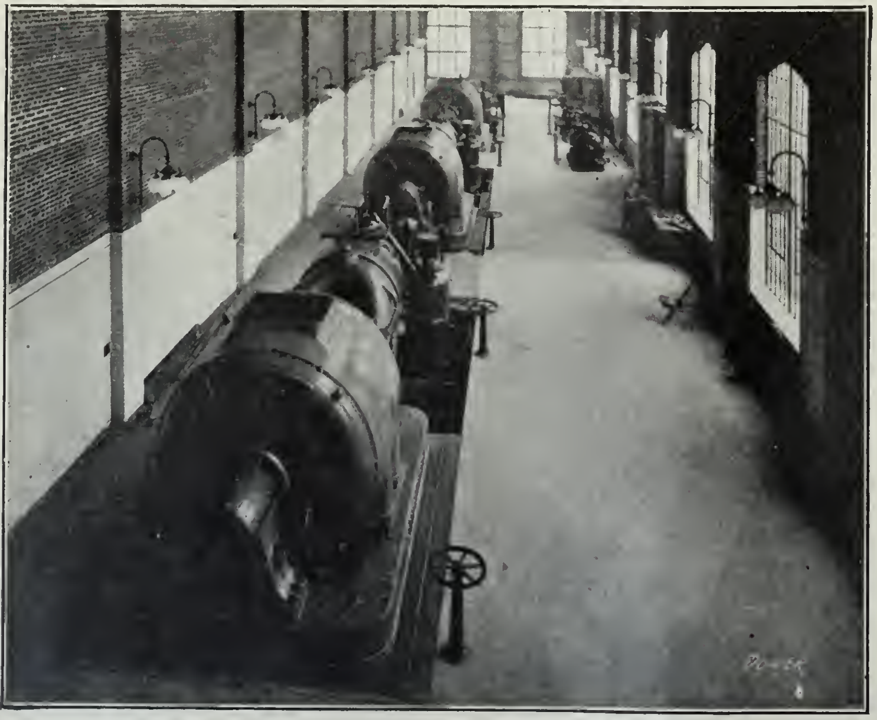


Fig. 6. View of Turbine Room

Piping

The main steam piping from the boilers to the turbines is laid out so that it has the advantages of both the unit system and the duplicate main system, without the complexity of the latter. By this arrangement it is always possible to have enough boilers connected to any one turbine, regardless of what boilers are out of commission. All expansion is taken up by long-radius bends and no piping is visible in the turbine room except the risers through the floor to the machines.

The feed lines from the pumps to the boilers consist of duplicate mains running the length of the boiler-room basement. Each boiler has a front connection to one of these mains and a rear connection to the other, thus insuring the feeding of any boiler should one of the mains or branch lines become disabled.

Electrical Equipment

As previously stated, the electrical equipment, except the generators and exciters, is housed in the wing structure. On the first floor there are two 500-kilowatt Westinghouse railway rotaries, each supplied from a three-phase transformer, with provision made for a third rotary and transformer. Two small transformers supply current for the plant lighting and power.

On the second floor is located the high-tension switch and bus structure which is constructed of concrete throughout, excepting the floor slabs, and contains, in addition to the regular service and spare switches, two sets of buses. The disconnecting switches are arranged so that either of these buses may be thrown out of circuit for repairs without interrupting the service.

On this floor is also located the feeder switchboard, containing the recording instruments and the remote control for the high-tension switches, and the bench board for the generator and exciter control. These are shown in Fig. 2. They face one another and are placed so that the switchboard operator may have an unobstructed view of the generator room at all times.

The lightning arresters, which are of the tank type, made by the General Electric Company, are located on the third floor.

The plant at present supplies six substations, in addition to a portable substation consisting of a 500-kilowatt rotary and transformer mounted on a car which may be run anywhere over the lines to handle an extra load or to temporarily take the place of any other substation that might be out of service for repairs or other reasons.

The largest of these substations is the old generating station at Seventh street. The greater part of the old equipment of this plant has been either sold or scrapped and the buildings are being renovated to make them more nearly fireproof. When the alterations have been effected this will be the main distributing station for handling the city load. This being on the opposite side of the river from the new power house, the high-tension cables are carried under the river in armored submarine cables and through underground ducts to the substation. The high-tension lines going to the other substations are carried partly on towers and partly on poles.

The entire plant was designed by and constructed under the supervision of Walter J. Jones, consulting engineer with offices in New York City.

Engineer's Problem in Selecting Coal

By George H. Bayne

In selecting coal, one is not concerned so much with the actual heat contents as with the ability of a certain coal to produce the greatest evaporation under the particular operating conditions. With this in view, directions are given for making standardization tests to determine the comparative evaporative efficiency of the various coals under consideration.

Rapidly increasing competition and the high cost of labor have made it necessary for manufacturers throughout this country to look most carefully into their operating expenses and cost of production. Having reduced to a minimum their expenses in the finishing rooms, sales departments and executive offices, and having installed elaborate systems for estimating and recording the cost of their output, they have now awakened to the fact that while they are giving strict and intelligent consideration to every phase of the economical manufacture of their product, they have neglected to consider seriously one of the most essential and fundamental points in their first cost; namely, the input of their coal and the economic distribution of their power as delivered to the factory from the boiler room. It is remarkable that in many large manufacturing concerns in this country, where every attention is given to systematizing and regulating the departments, very little, if any, consideration is given to the boiler room, in which may be wasted the most essential of our natural resources—coal—the economical consumption of which is one of the most vital questions in the generation of power. The first cost of the fuel, its adaptability to the furnace and economic firing by the fireman, in a great many instances, make or break competition and profits.

In the production of steam the first cost to the manufacturer arises; here his initial loss begins and here 50 per cent. of his expense is invested. Therefore, the question that confronts the wide-awake president or general manager of uptodate commercial industries is, how, under their present conditions, are they to increase the efficiency and at the same time reduce the cost of their power production. The answer to this question can be ascertained only by careful tests of the boiler plant under actual operating conditions; such tests to be conducted by a competent engineer, who will determine, practically and scientifically, what can be expected of the plant in conjunction with the different grades of fuel available in the market, also the most economical methods of firing and the cost of producing power under the existing conditions. When this is done, the manufacturer is confronted with the bare facts of having used so many pounds of coal and having produced so much measurable work. It then becomes a matter of simple arithmetic for him to establish a definite cost for his power production. This test, followed up by systematic daily reports from the boiler room, will give the operator such an understanding of the relative value of horsepower that it will enable him to locate the cause instantly, when excessive rates become apparent and the power department has passed the economical point. It then becomes the duty of every operating engineer to furnish such data as may be required for this purpose, and to familiarize himself with the practice and methods involved, wherein he can find out for himself just what his boiler room is doing. The engineer who produces the most power for the least money is invariably the man who is giving this question the most consideration.

In advancing a few suggestions on this topic the writer will confine the discussion to the testing of coal and boilers for establishing the rate for a boiler horsepower, and will endeavor to lay down a few essential rules whereby this rate can be ascertained without going into a long theoretical treatment.

The majority of manufacturing concerns have installed certain boilers and appliances which to the mind of the constructing engineer, at the time the plant was built, were best suited to the purposes for which they were designed. Owing to the cost of installation and so forth, it is, in most cases, up to the operating engineer to make the best of the installation as he finds it, although it might often prove more economical in the end for the concern by which he is employed to completely remodel the plant. Unfortunately, this is usually out of the question, owing to the necessary investment of extra capital; however, every plant has its gait and this gait can be found and the maximum point of economy determined. Whether the plant is hand fired, or mechanical stokers are used, or whether the steam producer is a horizontal return-tubular boiler or a water-tube boiler, etc., the engineer is given charge of the plant and is expected to produce the best possible results under the existing circumstances. This may or may not be true economy, but it is nevertheless a glaring fact which faces 80 per cent. of the stationary engineers. Again, it must be borne in mind that the purchasing of the fuel, in ninety-nine cases out of a hundred, is done by a purchasing agent whose business it is to buy coal as cheaply as possible, regardless of quality. It is usually not the fuel that gives the most heat per ton, but the coal that gives the most bulk per dollar that is turned over to the engineer for steam-producing purposes. This point often works a hardship on the engineer and fireman and is one of the most vital questions of boiler-room economy, over which, unfortunately, the average engineer has no control. The writer thoroughly believes in the old adage that the proof of the pudding is in the eating, and that the man who controls the burning of the coal, and is responsible for it, should have a voice in determining which fuel, at the market price, is best adapted to his needs.

With these facts in mind assume, for example, a boiler room of three to five boilers, of either the water-tube or horizontal return-tubular type, hand fired, properly installed, equipped with the necessary auxiliaries, and in charge of a competent and wide-awake stationary engineer, a man of common sense, ability and experience in boiler-room practice. These are the first essentials of boiler-room economy. The next question to take up is the purchasing of fuel. This, as before mentioned, is generally done by the purchasing agent of the firm. To be able to produce the best results, this man should be in close touch with the engineer and they together should pass upon the choice of the fuel supply. The former is thoroughly in touch with the market conditions and with the different grades of fuel at his disposal; the latter, on the other hand, from his experience in burning various grades of coal, can supply the practical information as to which fuel is the best to be used under the particular conditions; but he can do this only after testing the several grades which are at the disposal of the purchasing agent.

The question of buying fuel on the heat-unit basis, or the B.t.u. specifications, has been so widely discussed and exploited of late that it is hardly necessary to go into the matter in detail; that the B.t.u. specifications have good points and that they also have limitations is conceded by all. Specifications of this nature, when in the hands of experts, are usually satisfactory to both consumer and seller, but otherwise the result is apt to be unsatisfactory to both parties. The wise purchasing agent, having classified the several grades of coal available in his territory, as far as prices are concerned, calls in the services of a competent engineering chemist, who analyzes the fuel and advises him as to its constituents. This, then, establishes a comparative standard of analysis for each of the grades of fuel and gives the buyer an insight into the quality of the commodity he is buying, besides establishing a basis upon which he can place his contract with the coal company supplying the fuel and thereby compel the shipper to furnish, during the life of that contract, a uniform quality of coal within reasonable limits. This is as far as the purchasing agent can go for the present and is only one-half the problem in the placing of his fuel contract. With this information in hand the purchasing agent orders sample cars of the several grades under consideration and turns them over to the operating engineer for practical tests, to determine which coal will work to the best advantage under his particular conditions. With the results of the tests, together with his knowledge of the fuel, the purchasing agent is in a position to place his contract to the best advantage and with precision, putting the responsibility upon the engineer to produce results.

This preliminary test by the engineer for the benefit of the purchasing agent is generally conducted in the following manner: One boiler in the line is designated for test purposes and is put in thorough repair; the boiler tubes are blown out and the combustion chambers and furnace are cleaned; the brickwork and settings are pointed up and made as tight as possible; the blowoff cocks are blanked off; the feed-water line is bypassed, so that this boiler can be fed separately, which is usually done by attaching the bypass to the emergency feed pump, with which all plants should be equipped, and by placing a meter in the feed line between the pump and boiler. A Worthington duplex piston meter or a Venturi meter should be used for this purpose, or the water may be weighed in barrels placed on scales. The latter method, although the most accurate, usually requires the attendance of an extra man and is frequently impracticable. It has been the writer's experience that when meter readings are taken with either of the above instruments, which should be calibrated before and after the test, that the results are sufficiently accurate and have the advantage of eliminating, to a large extent, the personal equation of the attendant. A thermometer for taking the feed-water temperature should be placed in the feed line as near as possible to the entrance of this line into the boiler. A pyrometer should be placed in the breeching between the damper and the boiler in order to take the temperature of the escaping gases. A hand-draft gage of the "U" type should be used to ascertain the draft over the fire, at the stack, and in the combustion chamber in the case of horizontal return-tubular boilers; if a water-tube boiler is being tested the draft is taken in the different passes and at the stack. A platform scales should be placed in the fire room and the coal under test is weighed upon them before delivering to the fireman. With this equipment in place, the engineer in charge is ready to proceed with the test and must bear in mind the following points:

1. That his object is to ascertain the commercial efficiency of the coal in question under the particular conditions, and that he is making a fuel test and not an elaborate boiler test with all its refinements.

2. That the value of this coal to him is just in proportion to the number of pounds of coal that he has to use to produce a boiler horsepower over a fixed period of time, and is in proportion to the number of pounds of water evaporated by the boiler over this period when reduced to an evaporation from and at 212 degrees Fahrenheit.

3. That to accomplish the best results he should place the boiler and furnace in charge of an experienced fireman and should personally supervise the taking of readings and the general method of firing throughout the test.

4. That he must keep a record of the initial and final readings of the meter (if one is used), the average pressure maintained by steam gage, the average temperature of the feed water, the average temperature of the escaping gases (by the pyrometer), the number of pounds of coal fired and the weight and nature of the ash and clinker taken from the furnace during the cleanings.

5. That this test should cover the usual factory day, either eight, ten or twenty-four hours, as the case may be.

6. That careful observations should be made at frequent intervals to see that there are no holes in the fire and that the fireman is doing his work properly.

The simplest method of starting a test is to have the fire completely cleaned out and a new fire started, and take note of the hight of the water in the gage; for a matter of convenience, a string may be tied around the gage showing the hight of the water column. This done, the test is ready to start, the feed water is turned on through the meter and the fireman begins to use the weighed coal, and continues to do so during the entire test. Care should be taken to keep the feed water as near as possible to the hight of the string on the gage. To close the test the water is brought back to the starting point on the gage and the fire is allowed to die down and, as soon as practicable, is cleaned. The total amount of refuse from the firebox and ashpit is removed and weighed on the scales, care being taken that the fireman is not permitted to wet down the resultant refuse, as is common practice, before it is weighed.

When these instructions have been carefully adhered to, the engineer has at his command all the necessary information to calculate the results. This process of testing is repeated for each grade of coal that is under consideration by the purchasing agent and each test should be conducted by the same man and fired by the same fireman. The results of these tests, when reduced to an evaporation from and at 212 degrees Fahrenheit and referred to the different costs per ton, will give the engineer and the purchasing agent a definite comparison of the commercial value to them of the respective fuels.

Installing Refrigerating System

By F. E. Matthews

Kind of pipe and fittings to use, how joints are made and leaks in the system detected and repaired.

A complete refrigerating system embraces a power plant with the usual details in addition to the several members included in the refrigerating system proper. The installation of the compressor requires little more care than that of the engines and boilers of a refrigerating or any other plant. It is scarcely necessary to state, however, that every detail connected with the installation of the piping of ammonia, or other refrigerable gases as a working medium should be executed with the greatest care. Only such materials as have been found by responsible builders to be well adapted to their respective purposes, should be employed. For ammonia, good full weight wrought-iron pipe is to be recommended for the expansion or low-pressure side, and extra-heavy pipe and ammonia fittings of approved design for piping the compression side.

In damp places, where the low-pressure gas headers are liable to rust abnormally, these also should be of extra-heavy pipe.

There is always a favorable opportunity for corrosion where pipes pass through cooler walls, it usually being practically impossible to make the insulation air tight at this point. If such parts are protected by a thick coat of asphaltic paint, or better, a double layer of canvas saturated with asphaltic paint or other waterproofing agents, it may save great annoyance and material expense in the years to come.

Making Joints

Ammonia pipe joints are usually made up with lead or rubber gaskets in male and female flanges, sweated on the pipes. Some builders employ a litharge and glycerin cement in making up screwed joints instead of solder, and there is little difficulty in making such joints tight if scrupulous care is exercised in seeing that only true, sharp, properly formed threads are used; that they are thoroughly cleaned, for which purpose gasolene is to be recommended; that the litharge is freshly and thoroughly mixed into a thin paste; and that the joints are made up tight.

It seems trite to suggest that gasket and flange joints should be drawn up squarely, but many a charge of ammonia has been lost through lack of attention to this detail. Rubber gaskets are particularly likely to blow out of improperly drawn up flanges months later when the rubber has had opportunity to be softened by oil. It can be most truly said that eternal vigilance in erection is the only price by which a permanently tight refrigerating system can be purchased.

When the erection of the plant is complete, and the piping thoroughly blown out to free it from dirt, scale, metallic chips and other foreign substances, an air pressure of not less than 300 pounds should be pumped on it to test for leaks.

Detecting Leaks

Leaks, such as split pipes and improperly made up gasket joints may be readily located by the sound. In fact, in a still cooler, sound is the most efficient means of detecting very small leaks even of ammonia, where the air has become so laden with the fumes to make the usual methods of testing difficult. When all of the leaks have apparently been stopped, it is advisable to pump a pressure on the piping and let it stand for ten or twelve hours. The drop in pressure, provided there is no appreciable change in temperature, will indicate the amount of leakage. As a final precaution the air may be allowed to escape and the system again charged with air into which a sufficient amount of ammonia has been fed to make any leak easily detected either by smell or by means of sulphur sticks.

Sulphur sticks used for testing for ammonia are made of pieces of white pine, or other wood which burns with little smoke, split into splinters about half the size of a lead pencil and from 6 to 8 inches long. These sticks are then dipped into molten sulphur so that about 4 inches of the ends are thoroughly coated, and after being cooled are ready for use. In testing for leaks, the sticks are ignited and held close to the suspected pipe or fitting. If there is escaping ammonia, it will, on coming in contact with the burning sulphur, produce a very noticeable white cloud. The approximate location having been found by means of the sulphur fumes, the exact position of the leak may be located by means of oil applied to the leak by means of a long-nosed oil can or soap suds applied with a brush.

Where there is a likelihood of existence of leaks in pipe or submerged condensers, or other places where escaping ammonia would not readily be detected because of its entering into solution in the cooling water or cooled brine as the case may be, it is advisable to test these liquids periodically with some reliable reagent. Where there are not too many foreign substances present, the litmus and turmeric papers are fairly reliable. A more satisfactory reagent, however, for use under the varied operating conditions of refrigerating and ice-making plants, is Nessler's solution, a few drops of which added to the suspected water or brine will show a yellow discoloration for slight traces of ammonia, increasing with the amount of ammonia present until with large quantities a reddish-brown precipitate is formed.

Repairing Leaks

Many small leaks such as occur in ammonia fittings may be stopped by the judicious use of a set of small calking tools.

Porous spots in iron and steel castings may sometimes be remedied by the judicious use of some rusting solution such as sal ammoniac or hydrochloric acid. Where the leaks are occasioned by blow holes of considerable size occurring where the application of pressure will tend to drive the substance into the porosities of the iron, some of the patented rust-joint preparations may be effective.

Troublesome leaks due to imperfect welds in the seams of pipes may be effectively repaired by first cleaning the pipe with a file and some suitable soldering solution, then applying a closely laid course of bright steel wire. The layer of wire should then be saturated with the soldering solution and the whole surface thoroughly coated with solder, special care being taken to see that it is thoroughly sweated in at the point where the leak occurs. The steel wire supplies the tensile strength, the lack of which in the solder would often allow the ammonia under pressure to lift off the solder coating. A hard solder should be employed and the steel wire should be thoroughly "tinned" to protect it from rust.

For soldering iron and steel pipes two soldering solutions should be employed, the first being simply a cleaning solution of concentrated hydrochloric acid, and the second a saturated solution of zinc chloride, commonly known among tinners as "cut acid." This second solution is prepared by dissolving metallic zinc in concentrated hydrochloric acid. Some builders add to the zinc chloride thus formed an equal amount of ammonium chloride.

Leaks, both in pipes and castings, may be repaired and separate pieces of pipe may be welded together to form continuous pipes, coils and headers, by means of improved processes of electric and oxyacetyline welding. Ammonia receivers, as well as larger shells such as are used for constructing absorbers, condensers and generators of absorption machines, are also made by this process.

Boiler Tube Brushes and Scrapers

By Warren O. Rogers

> *A stiff wire brush will clean a boiler tube if the deposit has not caked. A more effectual device is a combination of brush and scraper, the scraper loosening the soot and the brush removing it. A number of these types are described.*

Although the improvements in tube scrapers have been marked, manufacturers have also improved the tube brush, and in several instances a combination tube brush and scraper has served to good purpose. The idea of making a combination device is due to the fact that as a general thing, a scraper will not push out all of the soot it detaches from the surface of the tube. The brush following the scraper blades not only brushes the soot loosened on the tube, but also cleans out the soot that sifts past the scraper blades. With a combination scraper a tube should be cleaned thoroughly.

An early type of wire brush used for brushing out soot is shown in Fig. 1. The brush is so constructed that the sections do not allow the open spaces between them to extend the whole length of the brush. Each section has an offset at the middle, as shown, so that the open space at one end is covered by the offset at the other end. This cleans the whole surface of the tube each time the brush is pushed through.

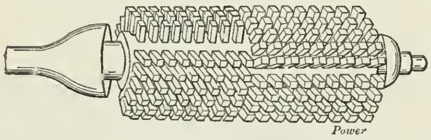


FIG. 1.

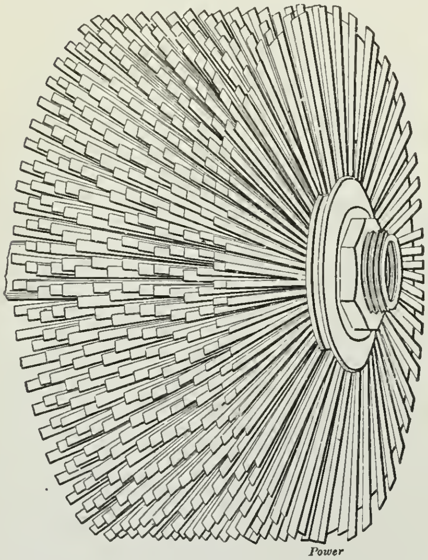


FIG. 2.

On the end of the brush is fitted a conical cap which not only protects the wire, but also allows the brush to easily slip back into the tube in case it is pushed entirely through. The brush is made of steel wire and malleable-iron castings.

Another make of tube brush is shown in Fig. 2. It is made of stiff steel wire set into a center piece, as shown. The size ranges from 2 to 20 inches in diameter, the latter size being illustrated. This brush is made by the J. W. Paxson Company.

A flue brush with an iron guard, made by the Pilley Packing and Flue Brush Manufacturing Company, and others, is shown in Fig. 3. It is made of steel and is built up on an iron casting having flanged ends, making it impossible for the brush to slip in the tube. The brush is easily brought back into the tube, in case it is pushed through at the rear end by the iron guards.

Another type of brush is shown in Fig. 4. It is rigid, and the wires are supported in iron rings which are clamped on the spindle. It is made by J. W. Paxson Company.

Fig. 5 illustrates a very durable brush. It is made of flat tempered-steel wire. The ends of the brush are protected by shields. It is made by A. W. Chesterton Company, which firm also makes a brush similar to that shown in Fig. 4.

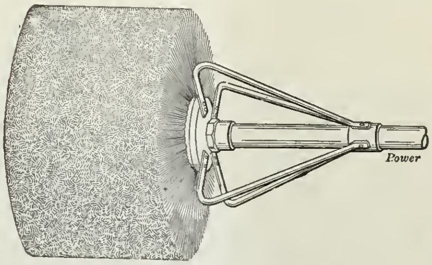


FIG. 3.

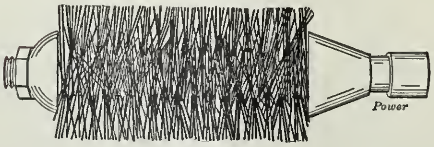


FIG. 4.

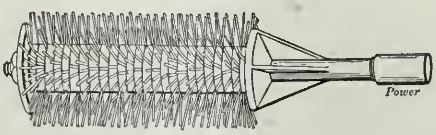


FIG. 5.

The brush shown in Fig. 6 is made by Warren F. Flint. It is constructed of fine steel wire secured to a twisted central piece. One end is filled with a pipe section having a thread for connecting a handle. Another design, made by the same company, is shown in Fig. 7. This is a hard wire brush, the ends of which are protected by caps.

Still another brush tube cleaner is shown in Fig. 8. It consists of a brush composed of strong, flexible wire, which when pushed through the tube removes the soot from off the surface and pushes it out at the end.

Fig. 9 shows a design of brush made by F. N. Browne. It is constructed of stiff steel wire and is fitted with a pipe connection for a handle attachment.

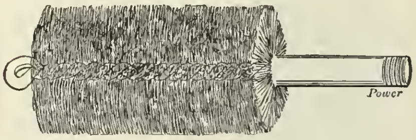


FIG. 6.

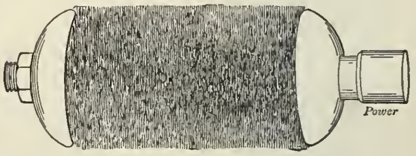


FIG. 7.

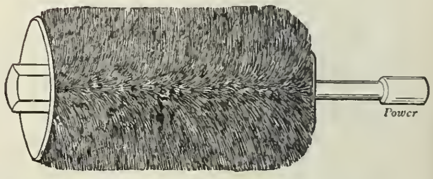


FIG. 8.

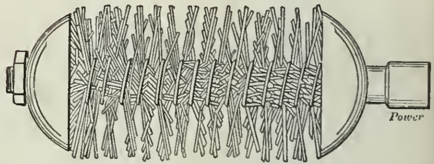


FIG. 9.

William Mulholland makes the brush shown in Fig. 10. Steel wire is used, and the brush is reinforced at one end by a cap which also serves as a receptacle for the handle.

Fig. 11 shows a design of flue brush made by several firms. It consists of steel wires secured to two wire rods, twisted as shown in the cut. The result is that the steel wire forming the brush takes a spiral form.

A combination brush and scraper is shown in Fig. 12. It is manufactured by James L. Robertson & Sons, and several other firms. The scraper section consists of scraping blades which readily conform to the opening in the tube, due to the joint construction. As the scraper edges are forced in by the tube surface the spring is compressed, and as a result the scraper blades are forced against the tube by the action of the spring which is protected from heat and dirt by a casing. The brush removes the loosened soot.

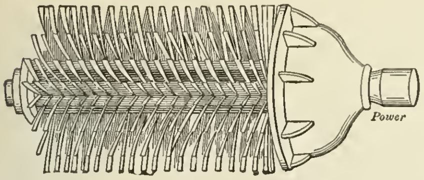


Fig. 10.

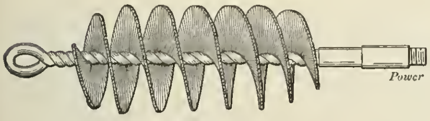


Fig. 11.

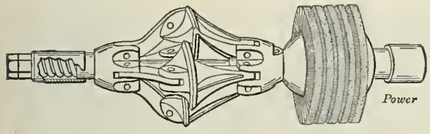


Fig. 12.

The combination flue brush and scraper shown in Fig. 13 is made by A. W. Chesterton Company. The scrapers cut the soot from the tubes, and it is then pushed out of the tubes by means of the steel brush, the tube being cleaned in one operation. The scraper knives are expanded or contracted by turning the rod to the right or left, thereby moving the adjusting nut up or down on the threaded center rod. The brush is made of tempered-steel wire.

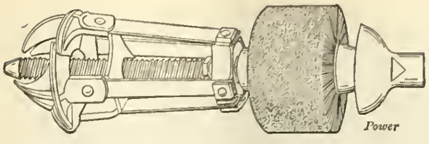


Fig. 13.

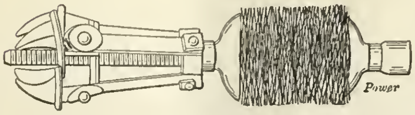


Fig. 14.

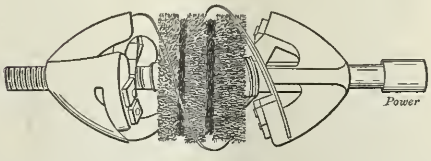


Fig. 15.

A combination brush and scraper made by the Newark Brush and Scraper Company is shown in Fig. 14. The head is equipped with a malleable-iron scraper which removes the soot from the tube, and the wire brush finishes the work.

The Gem spiral brush tube scraper, made by the Gem Manufacturing Company, is shown in Fig. 15. The scraper is self-adjusting to irregular surfaces in the tube. The spiral-spring blades are made of tempered steel. To facilitate cleaning the tube the brush is placed midway between the two heads. It can be removed and replaced when worn out.

Another design of combination brush and mechanical cleaner has the head driven forward by a spline shaft set in the hollow shaft of a rotary engine. Fig. 16 shows the type of cleaner used with fire-tube boilers. The spiral portion is composed of small cutters which cut the scale and soot, while the brush at the inner end pushes the loosened deposit out of the tube.

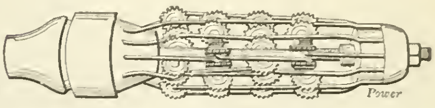


Fig. 17

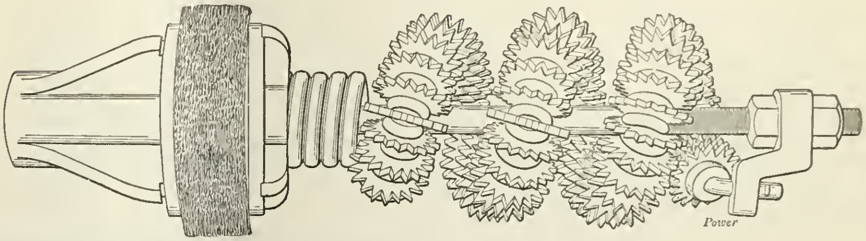


Fig. 16.

A flexible fire-tube cleaner is illustrated in Fig. 17. It is made up of a series of cutting wheels which revolve between steel-wire springs. This allows the cutting wheels to give in case any obstruction is encountered. At the rear end of the head is a brush which removes the loosened soot. The last two mentioned scrapers are made by the Poole Manufacturing Company.

The Perquisites of an Engineer

Under the above title the *American Journal of Steam and Electrical Engineering* published an editorial which is worthy of repetition and should be given the earnest attention of every engineer.

"An operating engineer is rightfully regarded as one of the chief units in an industrial establishment. It is absurd for an employer to have skilled labor and expert workmen in other parts of his plant and have incompetent men at the power-generating end. Employers generally realize this, but they do not always recognize the fact that the lives of their men and the preservation of their property are dependent in large manner upon the men in the engine room. It is argued, and very often with reason, that employers show their lack of appreciation by failing to pay their engineers commensurate salaries.

"Yet there are causes underlying many of the low salaries, not the least of these is the 'leak' that employers find in the engine room.

"This is not an ordinary leak which takes tangible shape in increased steam consumption, waste of oil and fuels, etc., but a leak from the employer's pocket just the same, which is caused by what is commonly called 'graft.'

"From being a regular calling of its own, 'grafting' has drifted into practically all professions until we find men in all walks of life losing their manhood and self-respect by accepting, and in many cases demanding, little gratuities.

"It is for such men primarily that there has recently been formed the Brotherhood of Stationary Engineers, an organization which does not aim to interfere or injure any of the existing bodies of engineers, but which has for its main tenets and principles the abolishment of all graft, big and little. Any engineer who is discovered accepting a bribe from a salesman is blacklisted by the new organization, who have what they call an honor roll, from which employers can pick men whom they are sure will be absolutely honest.

"To be sure these men will command higher salaries than the grafting class, but employers as a rule are willing to pay something extra for honesty and integrity. When an engineer belonging to this new association is in charge of a plant, his firm can have perfect confidence in his buying material for that plant which is best adapted to conditions, rather than from the salesman who can be bled for the most money.

"The name operating engineer has fallen into reproach on account of the conditions above stated. It will be the task of those men who believe that honesty in little things means honesty in greater things, to revive public confidence."

The Largest Valves in the World

Special Correspondence

> *Three 9-foot gate valves for the control of hydraulic turbines of 12,000 horsepower capacity each. The body is of cast iron and the gate of cast steel, the latter being raised and lowered by electric power, requiring three minutes for each operation.*

What it is fairly safe to assume are the three largest gate valves in existence, have recently been constructed by the Chapman Valve Manufacturing Company, of Indian Orchard, Mass., for the Ontario Power Company, of Niagara Falls, Canada. These valves are 9 feet in diameter of water way, and each is to control the water to drive a hydraulic turbine generating 12,000 horsepower. It was found when one of the valves was completed that it weighed 130,000 pounds.

From the various illustrations a good idea of the size of the valve may be had. The overall hight is 30 feet 3 inches and the width is 11 feet. The total thickness over the flanges is 6 feet 8 inches. Due to the size of the gate, a 14-inch bypass valve is provided for relieving the water pressure when opening or closing the valve. Fig. 2 shows the valve body mounted upon a flat car for shipment, and its size relative to the men and the locomotive will be appreciated. The two flat cars are required to handle the parts of one valve. In Fig. 3 is shown the cast-steel gate resting in one-half of the cast-iron body, although the bronze shoe which goes between the gate and the guide is not in place.

In actual operation the valve gate will be under a pressure of 60 pounds per square inch, and the total load carried by the gate will then be more than 550,000 pounds. The gates are of cast steel and weigh close to nine tons each. The valve bodies are of cast iron, heavily ribbed to withstand the pressure, and the valve seats are of bronze, these being held in the body by special bronze, headless screws through the ring face. The usual method of construction for the valve seat, that is, threading it into the body, was not deemed safe because of the heavy pressure around the periphery of the seat. It was feared that this pressure would cause the seat ring to collapse, and so the screws were used to give it a firm grip upon the valve body all around. Tightness between the bronze seat ring and the cast-iron body is assured by lead calking.

Guides are provided for the cast-steel gate, and brass shoes are suitably placed to take the wear between the gate and the guides. At the instant of closing, the valve gate is forced away from the guides and rests entirely against the bronze seat ring, the water pressure holding the gate tightly shut. This final seating upon the bronze ring is attained by putting the seat in on a slight angle, giving the effect of the wedge-shaped plug of an ordinary gate valve.

Raising of the valve gate is accomplished by two tobin-bronze spindles, 4½ inches diameter, with threads of 2 inches pitch. These spindles are 12 feet 3 inches long and are operated through gearing, by

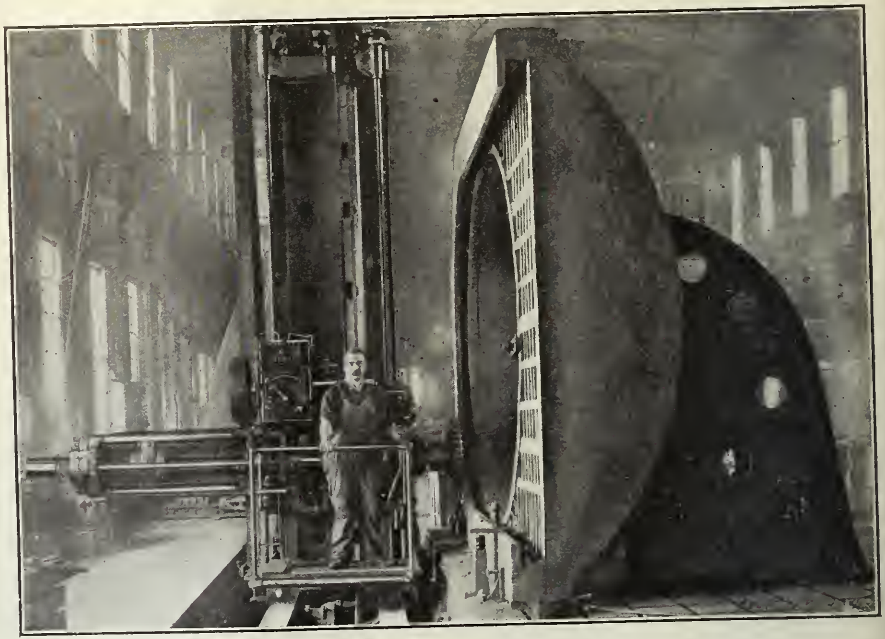


FIG. 1. HALF VALVE BODY ON SELLERS' MILL

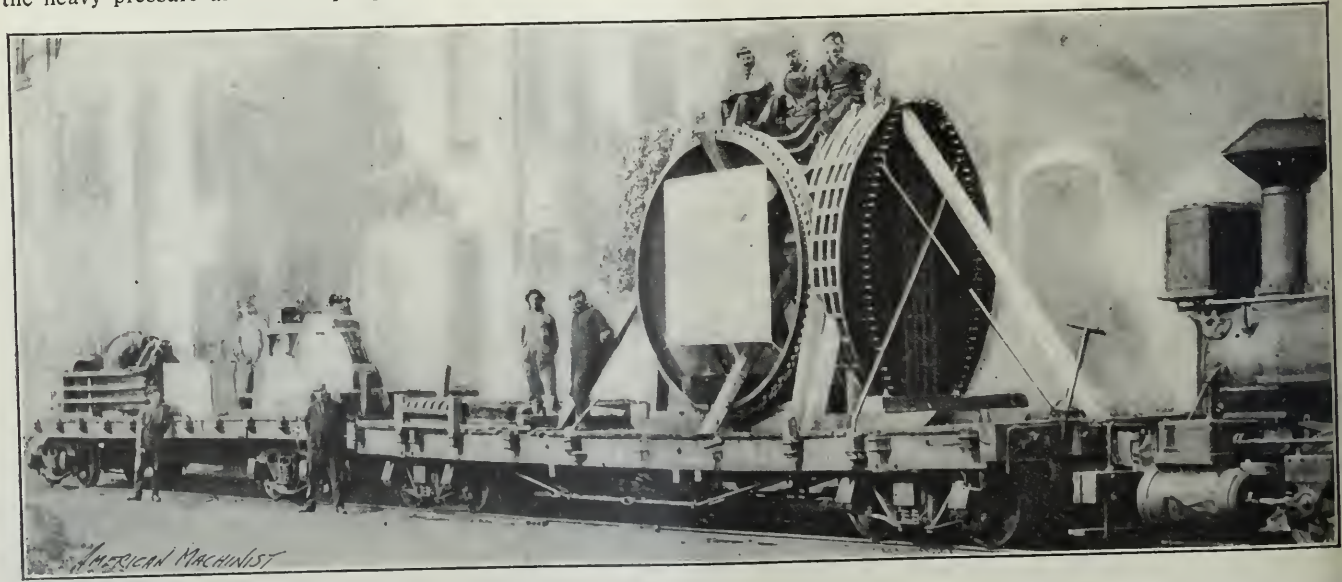


FIG. 2. COMPLETED PARTS OF ONE VALVE, REQUIRING TWO CARS FOR SHIPMENT

a 15-horsepower alternating-current motor, three minutes being required to completely open or shut. To prevent thrashing and knocking of the spindles inside the gate as the former rotate, the spindle sleeves are bored in the cast-steel gate, to an easy fit over the spindles. Automatic limit switches are provided at the top and bottom of the gate travel and arrangements are such that at the end of this travel it is impossible to restart the motor in the wrong direction. If, however, the motor is stopped with the gate part way open or shut, the motor may be restarted in either direction. A magnetic brake assures quick stopping.

As may be supposed the machining of valves of this size is of some interest. In Fig. 1 is shown the milling of the facing of one of the body halves, this work being done in the Sellers' mill. The large angle bracket at the back of the body, and bolted to the bed, keeps the body in an upright position, while the cutter is traveled over the U-shaped surface.

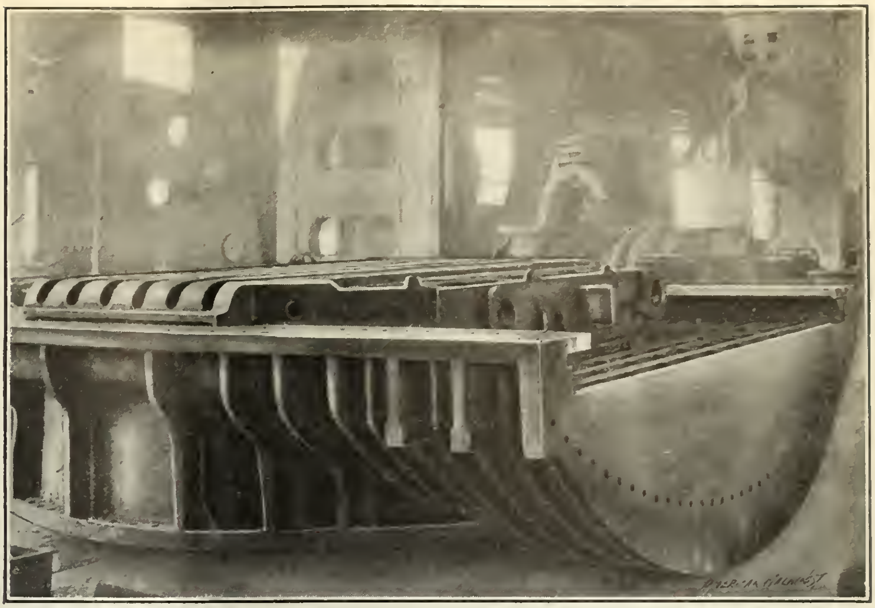


FIG. 3. GATE IN PLACE IN ONE-HALF OF BODY CASTING

Curves for Factors of Evaporation

By Julian C. Smallwood

> *A self-contained chart giving the factors of evaporization for a wide range of boiler pressures and feed-water temperatures and for qualities from 96 per cent. dry to 300 degrees of superheat. The curves are based upon the heat values given in the recently accepted steam tables of Marks and Davis.*

The accompanying chart shows the factors of evaporation for boiler pressures from 15 to 300 pounds per square inch absolute, with steam qualities from 96 per cent. dry to saturated and from saturated to 300 degrees superheat, and feed-water temperatures ranging from 32 to 300 degrees Fahrenheit. The usual formulas for finding factors of evaporation were used in plotting the curves and the heat values were taken from Marks & Davis' new steam tables.

The expression for the factor of evaporation for the three different cases of wet, dry and superheated steam may be written as follows:

For dry steam,

$$F_d = \frac{H - q_1}{970.4} = \frac{r + q_2 - q_1}{970.4}$$

For wet steam,

$$F_w = \frac{q_2 + xr - q_1}{970.4} = \frac{q_2 + r - (1 - x)r - q_1}{970.4}$$

$$= F_d - \frac{(1 - x)r}{970.4}$$

For superheated steam,

$$F_s = \frac{q_2 + r + C_p(T_2 - T_1) - q_1}{970.4} =$$

$$F_d + \frac{C_p(T_1 - T_2)}{970.4}$$

where,

F_d = Factor of evaporation for dry steam;

F_w = Factor of evaporation for wet steam;

F_s = Factor of evaporation for superheated steam;

H = Total heat in one pound of saturated steam above 32 degrees Fahrenheit, at the given pressure;

q_1 = Heat in feed water reckoned above 32 degrees Fahrenheit;

q_2 = Heat of liquid reckoned above 32 degrees Fahrenheit in one pound of steam at the given pressure;

r = Latent heat of vaporization of one pound of steam at the given pressure;

x = Quality of steam expressed in per cent.;

C_p = Mean specific heat of superheated steam at the given pressure and temperature;

T_1 = Temperature of saturated steam at the given ressure;

T_2 = Temperature of the superheated steam;

970.4 = Latent heat of vaporization of steam at 212 degrees Fahrenheit and atmospheric pressure.

HOW TO USE THE CURVES

On the "pressure curve" for the given boiler pressure find the point corresponding to the given quality of steam. From this point follow a vertical line downward and note its intersection with the horizontal line marked with the given temperature of feed water. The "factor curve" passing through this point of intersection is that of the required factor of evaporation.

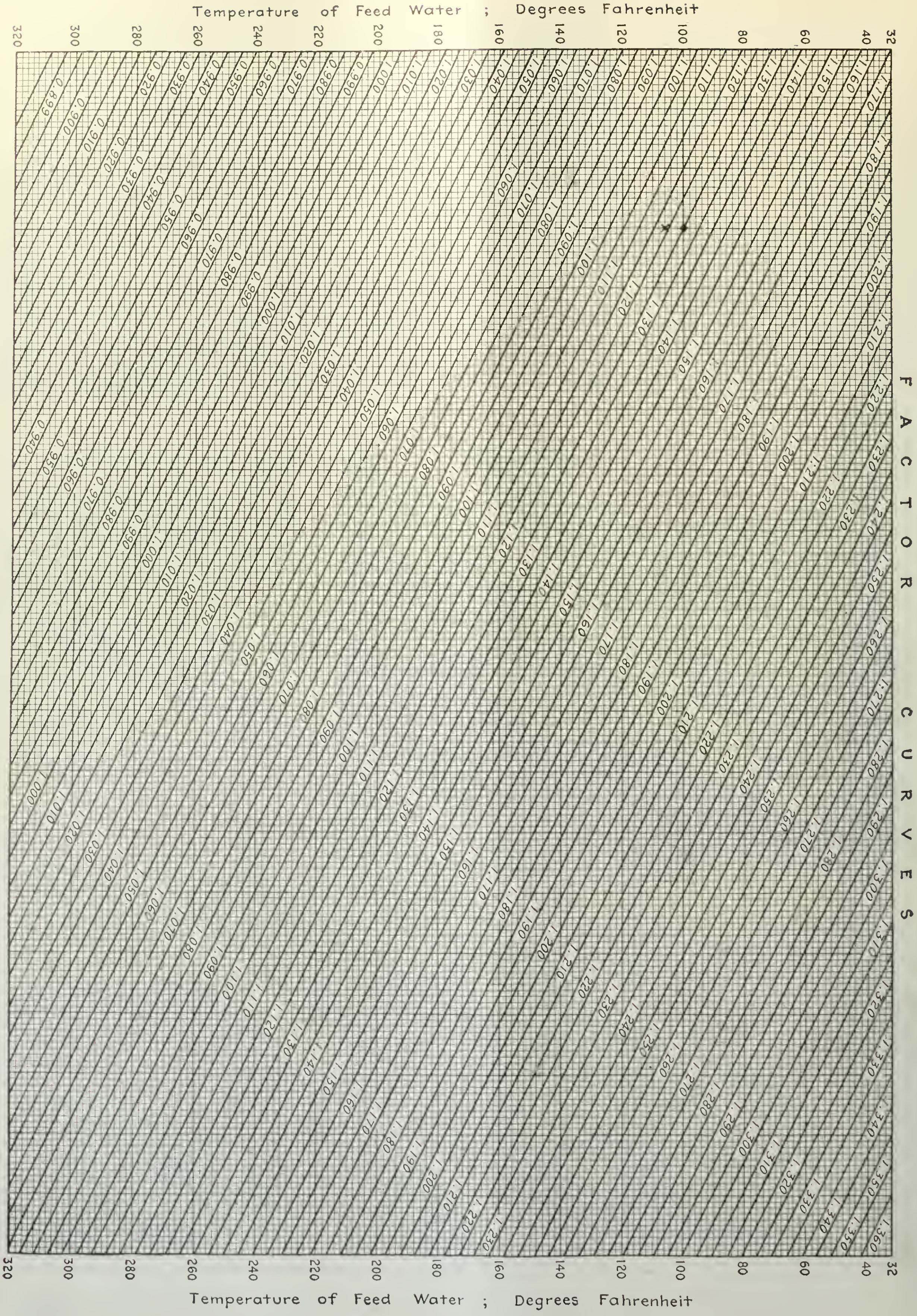
EXAMPLE

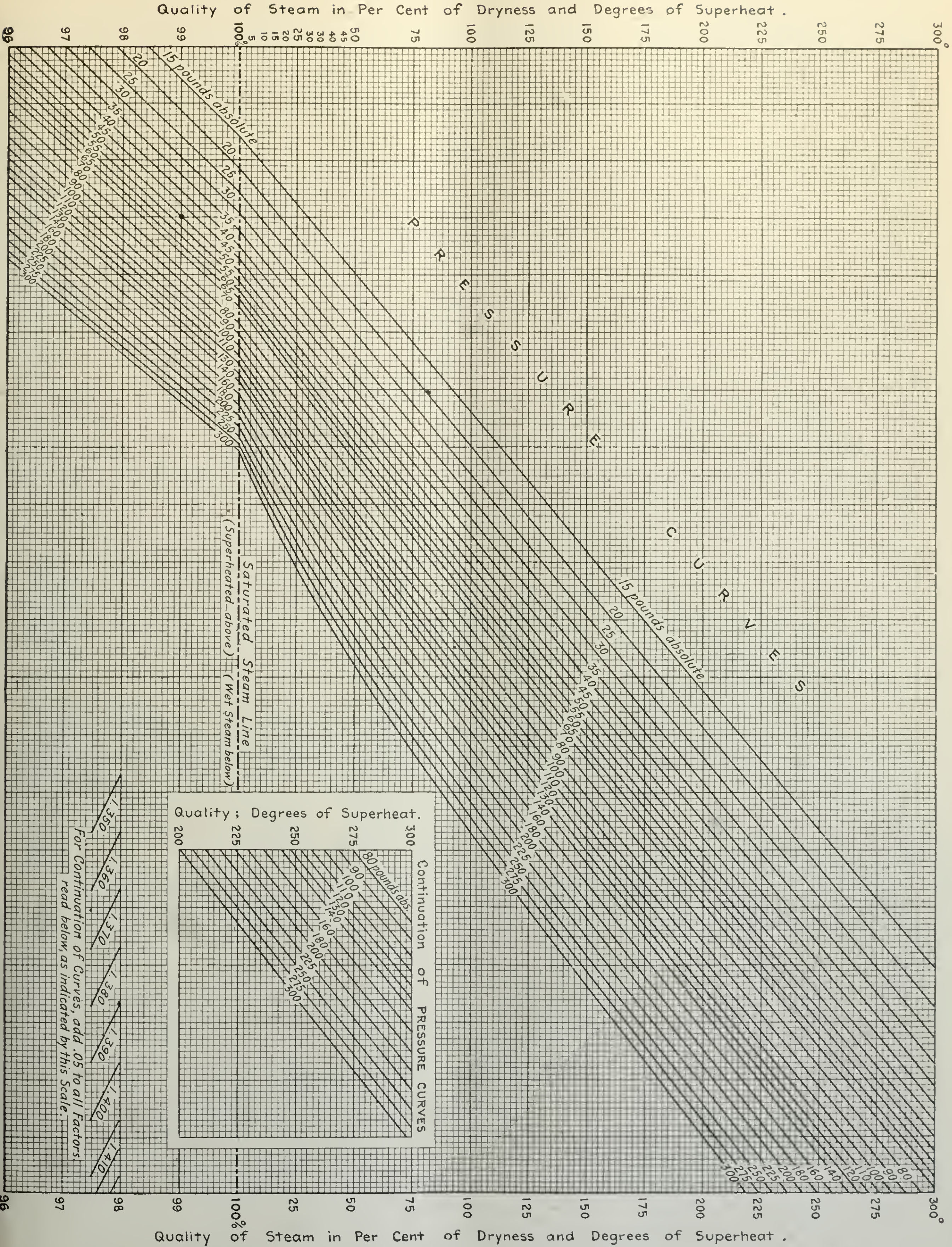
The steps to be taken in solving the following examples are indicated by the dot, square and cross on the chart.

Given a boiler pressure of 50 pounds absolute, a steam quality of 99 per cent., and the temperature of the feed water as 100 degrees Fahrenheit; what is the evaporation factor? Follow the 99 per cent. line to the right until it intersects the 50-pound pressure curve; this point is marked on the chart with a dot. From here follow down the vertical line to the line corresponding to 100 degrees feed-water temperature. Through this point (marked with a square) the factor curve 1.130 passes and this number is the required factor of evaporation.

TO INTERPOLATE

Now, suppose that the boiler pressure is 160 pounds, the steam quality 96½ per cent. and the feed-water temperature 106 degrees. Starting at the point in the steam-quality scale corresponding to 96.5, trace horizontally until the diagonal line for 160 pounds is reached. Then trace downward over to the other half of the chart until the vertical line being followed intersects with the horizontal line for 106 degrees feed-water temperature. This point is marked by the cross, and is located between the diagonal factor lines 1.12 and 1.13. Consequently, the first three digits of the required factor are 1.12. The 1.120 curve intersects the 106-degree feed-temperature line 3.7 small divisions to the left of the cross, the 0.7 being estimated; hence the required factor is 1.1237.


Temperature of Feed Water ; Degrees Fahrenheit
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
32
FACTOR CURVES
Temperature of Feed Water ; Degrees Fahrenheit



For Continuation of Curves, add .05 to all Factors, read below, as indicated by this Scale.

The Effect of Air in Jet Condensers

By Paul A. Bancel

Due to the air contained in the circulating water a larger air pump is necessary with a jet condenser than with a surface condenser. On the other hand, with a jet condenser the temperature of the mixture of air and vapor can be reduced to practically that of the entering circulating water; this is impossible with a surface condenser. The method of figuring the capacities of air pumps is given.

With both jet and surface condensers the size of the air pump and the power consumed in its operation depend largely upon the temperature at which the noncondensable gases are withdrawn. This temperature determines the weight of air in each cubic foot of the mixture of steam and air passing to the pump. If with a 28-inch vacuum the air-pump suction be at 90 degrees Fahrenheit, an air pump to remove one pound of air and steam mixed must have a volumetric capacity of about 710 cubic feet. But if the suction temperature instead of being 90 degrees were 70 degrees, the volumetric capacity of the pump need be only 320 cubic feet. This reduction in volume is explained by referring to Dalton's law of gases, which states that *a constant weight of gas at a constant temperature in a vessel of constant volume, exerts the same pressure regardless of the presence of any other gas or vapor.* In a condenser the total pressure is constant, corresponding in this case to 28 inches vacuum, or a pressure of practically one pound absolute. This total pressure is the result of the partial pressures of the steam and the air. The steam or vapor pressure is determined by the existing temperature; the lower the temperature the lower the pressure. However, since the total pressure is constant, the lower the temperature and pressure of the vapor the higher will be the air pressure, and therefore the air removed will be denser. At 28 inches vacuum and 90 degrees suction temperature, the absolute pressure of the steam in the mixture must be 1.417 inches of mercury; hence the remaining pressure, 0.583 inch of mercury, is the pressure of the air, and this determines its volume per pound.

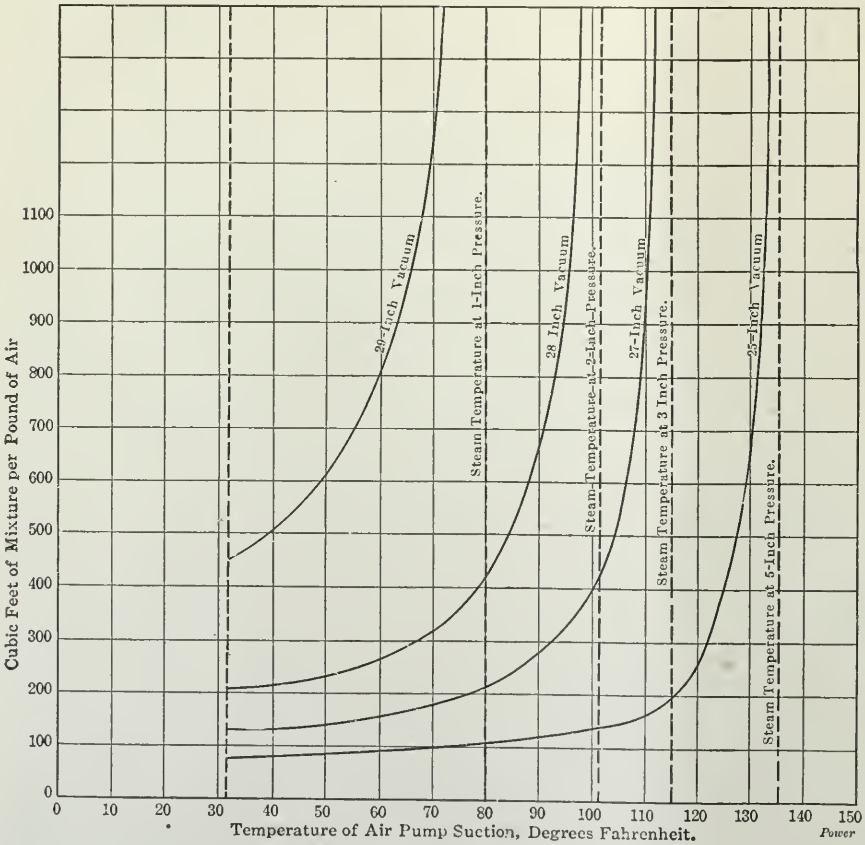


Fig. 1. Mixture per Pound of Air at Different Temperatures and Pressures

Knowing the pressure and temperature of the air (the temperature of the air is the same as that of the water vapor), its volume per pound can be determined. At 30 inches of mercury and 32 degrees Fahrenheit the volume of one pound of air is 12.4 cubic feet. According to the law of perfect gases this volume varies inversely as the absolute pressure and directly as the absolute temperature; hence the volume of one pound of air at a pressure of 0.583 inch of mercury and 90 degrees Fahrenheit temperature is

$$\frac{30}{0.583} \times \frac{460 + 90}{460 + 32} \times 12.4 = 710 \textit{ cubic feet}$$

Again, when the steam pressure existing in the mixture is that corresponding to 70 degrees Fahrenheit, which is 0.739 inch of mercury, the air pressure must be the difference between that and the total pressure.

$$2 - 0.739 = 1.261 \textit{ inches}$$

of mercury. A pound of air at a pressure of 1.261 inches and 70 degrees Fahrenheit has a volume of 320 cubic feet. The curves shown in Fig. 1 are obtained by calculations similar to the foregoing and are useful in determining the air-pump capacity for any conditions of pressure and temperature.

In a surface condenser the weight of air to be removed is considerably less than that to be removed from a jet condenser. In a surface condenser the air which must be removed is that which enters with the steam; namely, that which was in solution in the feed water and that which leaked into the piping and other parts of the system. In a jet condenser in addition to this the air which enters with the circulating water must be handled. Roughly speaking, the air pump of a jet condenser must remove twice the weight of air removed by the air pump of a surface condenser of equal capacity.

It would seem necessary, therefore, to have an air pump of twice the size. This would be true were it not for the fact that with a countercurrent jet condenser a much colder air-pump suction can be obtained than with a surface condenser. The coldest air-pump suction that can be maintained with either type is the temperature of the incoming circulating water, and with a countercurrent jet condenser this is quite possible. However, in a surface condenser, not equipped with an air cooler, there must always be a considerable difference between the temperature of the air-pump suction and that of the incoming circulating water, as the transmission of heat through the tubes depends upon this temperature difference. In practice it is seldom that the air-pump suction of a surface condenser is less than 15 or 20 degrees above the temperature of the incoming circulating water. Fig. 2 shows the effect of the temperature of the air-pump suction upon the weight of air removed by a pump of 237 cubic feet capacity. With a suction temperature of 50 degrees Fahrenheit and a condenser vacuum of 28 inches, one pound of air would be withdrawn, as shown by point *A*. With a suction temperature of 80 degrees the pump would handle only 0.56 pound of air, as shown at *B*.

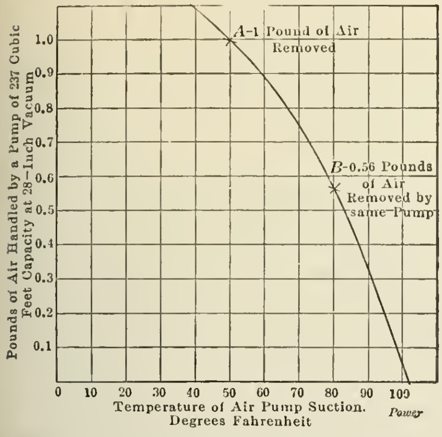


Fig. 2. Effect of Suction Temperature on Air Removed

By operating a jet condenser upon the countercurrent principle and cooling the air and vapor mixture, passing to the vacuum pump, to practically the temperature of the incoming condensing water, the great amount of air that must be removed can be handled effectively by a small pump. For reciprocating engines, vacuums of 25 or 26 inches can be readily maintained with small pumps, and vacuums of 28 to 29 inches for steam turbines can be economically maintained with vacuum pumps no larger than are necessary with surface condensers. With such a condenser the air drawn out by the pump is at its greatest density and the amount of steam going to the pump with it is almost negligible. Furthermore, the air passing to the pump, for the greater part is not heated at all; that is, it is drawn from the entering circulating water which contains a large amount of air in solution and which liberates it as soon as it enters the condenser. This air constitutes from 30 to 50 per cent. of the total air removed and, as is evident from the cross-section of a modern jet condenser shown in Fig. 3, the vacuum pump withdraws this air from the top of the condenser where the circulating water has just entered. Thus, with this condenser, the air leaking into the system and that coming in with the steam is cooled to its greatest density, and at the same time the air coming in with the circulating water is withdrawn by the pump before it is heated at all.

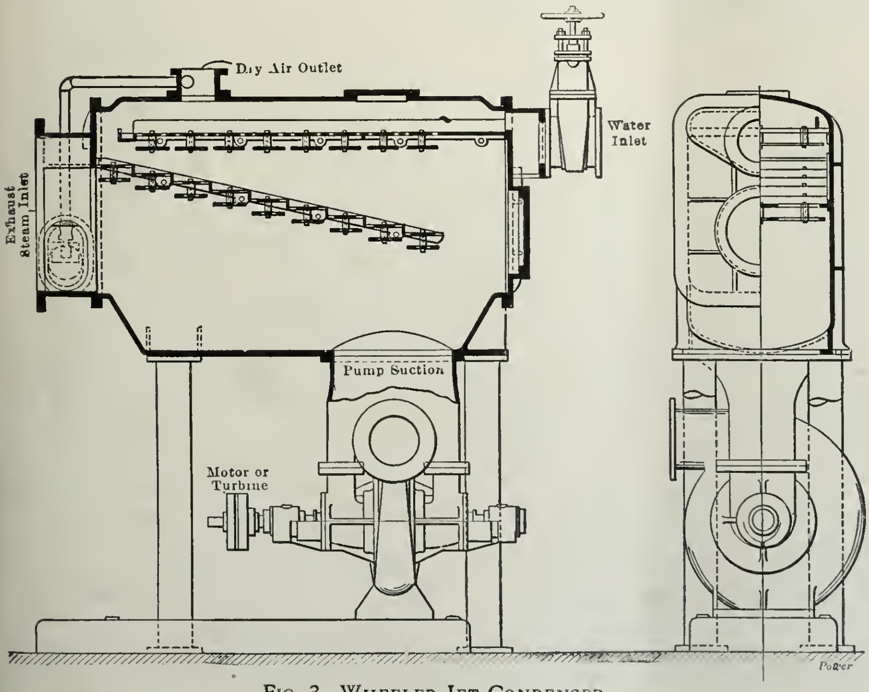


Fig. 3. Wheeler Jet Condenser

TABLE 1. TEST OF RECTANGULAR JET CONDENSER.

Time.	Corrected Vacuum Referred to 30-inch Barometer.	Steam Temperature Corresponding to Vacuum.	Temperature of Inlet Water.	Temperature of Outlet Water.	Vacuum Corresponding to Temperature of Discharge Water.
6:40	28.52	91	51.0	90.5	28.56
45	28.57	90	51.0	90.0	28.58
50	28.62	89	51.0	91.0	28.54
55	28.67	88	51.0	89.5	28.63
7:00	28.52	91	51.0	85.0	28.79
05	28.57	90	51.0	90.0	28.58
10	28.51	91	51.0	91.0	28.54
15	28.56	90	51.0	91.0	28.54
20	28.57	90	51.0	92.0	28.46
25	28.42	93	51.0	90.0	28.58
30	28.52	91	51.0	89.0	28.63
35	28.42	93	51.0	92.0	28.49
40	28.37	94	51.0	93.5	28.42
45	28.51	91	51.0	92.0	28.49
50	28.21	97	51.0	93.0	28.44
55	28.31	96	51.0	91.0	28.54
8:00	28.55	90	51.0	87.0	28.91

According to Professor Henry, carbon dioxide, oxygen, nitrogen and some other gases are dissolved in the exact ratios of the pressures under which they are exposed to the surface of the liquid, and the solubility of gases in liquids not only diminishes with diminished pressure, but also with increased temperature. From this it is evident that water will hold air in solution in an amount depending upon the temperature and the pressure. As regards the relation of air in solution to temperature, the amount of air which water can hold under a pressure of one atmosphere and at various temperatures, varies according to the curve shown in Fig. 4, which is plotted from figures given in the *Smithsonian physical tables.*

The figures shown in Table 1 are from a test of a Wheeler rectangular jet condenser at the Elmira Water, Light and Railway Company.

A pound of saturated steam at 28.57 inches vacuum has a latent heat of 1041 B.t.u., and with 85 per cent. quality the heat per pound of exhaust steam is 885 B.t.u. Dividing this by 39 degrees, the rise in temperature of the circulating water, gives 22.7 as the pounds of circulating water per pound of steam. In this particular case the turbine exhausting into the condenser was operating under an average load of 2000 kilowatts, and at 15 pounds of steam per kilowatt-hour this would be 500 pounds of steam entering the condenser per minute. Using this figure and the 22.7 pounds of circulating water per pound of steam the weight of circulating water per minute is found to be 11,350 pounds, and dividing this by 62½, the weight of 1 cubic foot of water, gives 181.6 cubic feet of water per minute entering the condenser.

Referring to Fig. 4, it will be seen that for a temperature of 51 degrees each cubic foot of water entering the condenser might contain 0.00157 pound of air. From Professor Henry's law it is evident that the weight of air liberated will be in proportion to the reduction in pressure. With a vacuum of 28.57 inches of mercury, it is found by referring to the steam tables that if the temperature of the vapor leaving the condenser is 51 degrees the vapor pressure is 0.19 pound, corresponding to 0.38 inch of mercury. Subtracting this from 1.43 leaves 1.05 as the total air pressure in inches of mercury at the exit of the condenser. This is the pressure which determines the amount of air which will be held in solution by the water.

It has been shown by reference to Fig. 4 that at atmospheric pressure and 51 degrees Fahrenheit a cubic foot of water will hold 0.00157 pound of air. Therefore, the air liberated under reduced pressure will be

$$\frac{(30 - 1.05) \times 0.00157}{30} = 0.001515 \textit{ pound}$$

of air liberated per cubic foot of water entering the condenser. This represents practically the total amount of air liberated by the cooling water, although to be more accurate it would be necessary to take into account the fact that the water in passing through the condenser is heated about 40 degrees.

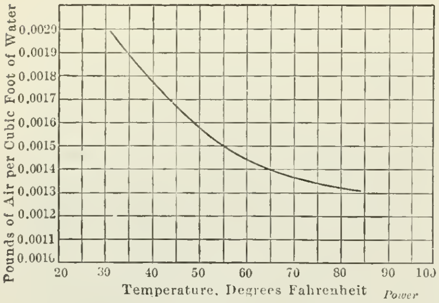


FIG. 4. CURVE SHOWING SOLUBILITY OF WATER AT VARIOUS TEMPERATURES

According to Fig. 4, its capacity for holding air in solution is lessened; hence it liberates free air which must be withdrawn by the vacuum pump. The effect of the temperature upon the air present is small in comparison to the effect of the reduction in pressure. With the pressure reduced to 1.43 inches in the condenser and the air pressure on the water reduced to 1.08 inches of mercury, the amount of air liberated per cubic foot of water is 0.001515 pound and the amount of air still remaining in solution per cubic foot of water is 0.000055 pound. The variation of this quantity with the temperature will be in proportion to the values shown in Fig. 1, so that after the water in the condenser has been heated to 90 degrees there would be further liberated from the water about 0.00001 pound of air. However, at the same time, the air pressure on the water has been reduced to a negligible quantity, since the pressure in the entrance to the condenser, namely, the pressure in the exhaust of the turbine, is due almost entirely to steam. Hence, practically all of the remaining air is liberated from the water. Therefore each cubic foot of water entering the condenser liberates about 0.001515 pound of air before it is heated and 0.000055 pound of air after it is heated.

Knowing the number of cubic feet of water per minute and the amount of air per cubic feet of water, the total air liberated from the circulating water per minute is found to be

$$181.6 \times 0.00157 = 0.285 \textit{ pound.}$$

This is the quantity of air which has to be removed per minute by the vacuum pump from the condenser in addition to the air coming in with the steam and leaking in from the various parts of the condenser and piping. Of this amount 0.276 pound is liberated from the cold entering water before it is heated and is therefore in its densest state and can be removed by the vacuum pump with the least expenditure of power and with the minimum size of air cylinder.

The installation at the Elmira Light and Railway Company consists of a rectangular jet condenser and a 10x26x12-inch Wheeler rotative dry-vacuum pump. Figuring the displacement of this air pump, its volumetric capacity per stroke is found to be 3.7 cubic feet, and with a speed of 75 revolutions per minute, the total volume would be 555 cubic feet per minute. Owing to the effect of clearance this capacity is slightly reduced. In this type of pump a rotative valve opens a special snifting port connecting the two ends of the cylinder at the end of the stroke; thus the air mixture in the clearance volume on one side of the cylinder is expanded into the closed cylinder on the other side of the piston. On the succeeding stroke, therefore, very little expansion serves to reduce the pressure of the vapor in the clearance to suction pressure, and this permits the entrance of a greater mixture of air and steam. Assuming that the effect of clearance reduces the air-pump capacity by 5 per cent., the net capacity per minute is then

$$0.95 \times 555 = 527 \textit{ cubic feet.}$$

The quantity of air contained in a cubic foot of the mixture going into the pump depends upon the temperature and pressure of the air. Unless the temperature of the air-pump suction is kept quite low the cubic feet of the mixture or the vacuum-pump displacement per pound of air removed becomes excessively large.

In the results shown in Table 1, the vacuum at time 6:45 was 28.57 inches referred to a 30-inch barometer. In the air-pump cylinder the vacuum would be somewhat higher in order to cause the flow of the gases against the resistance of the piping and the pump passages. Assume that the difference in pressure is 0.2 inch of mercury, or, in other words, that the vacuum within the pump cylinder is 28.77 inches. This is the total pressure of the mixture within the pump cylinder. The temperature of the mixture leaving the condenser is close to 51 degrees, the temperature of the incoming water; this temperature would be raised by the absorption of heat in the passage of the noncondensable gases from the condenser to the pump, and within the pump cylinder itself. Assuming that this rise in the temperature is 14 degrees the following will be the conditions:

Temperature of mixture, 65 degrees Fahrenheit.

Corresponding vapor pressure, 29.378 inches vacuum.

Air pressure, 29,378 — 28.77 = 0.608 inches vacuum.

The volume of one pound of air actually within the pump cylinder is, therefore,

$$12.4 \times \left(\frac{460 + 65}{460 + 32}\right) \times \frac{30}{0.608} = 652 \textit{ cubic feet}$$

Previous calculations showed that the volumetric capacity of this pump operating at 75 revolutions per minute was 527 cubic feet; hence the weight of air removed by the pump per minute is

$$\frac{527}{652} = 0.808 \textit{ pound}$$

Therefore, the circulating water, if saturated with air, may bring into the condenser every minute 0.285 pound of air and the air pump is capable of removing 0.808 pound of air; in other words, the air pump with a jet condenser must be designed to handle a much larger weight of air than would be encountered with a surface condenser, in a proportion of about 3 to 2. On the other hand, with a jet condenser, the temperature of air can be reduced to the temperature of the incoming water while with a surface condenser such low reduction is impossible owing to the fact that a considerable temperature difference is necessary to transmit heat from the outside to the inside of the tubes.

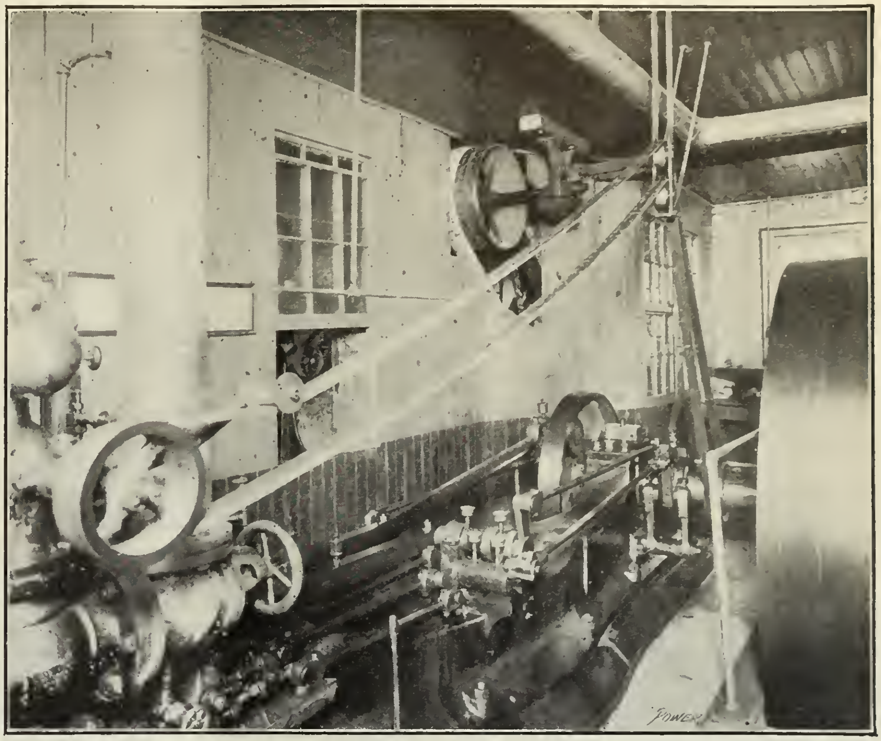
From the figures given it will be seen how essential it is that a jet condenser not only discharge its noncondensable gases at the lowest temperature, the importance of which is shown in Fig. 1, but also that it discharge its circulating water at the very highest temperature compatible with the vacuum, so that the amount of circulating water used, and therefore the amount of air brought into the condenser, is a minimum. If the previously mentioned condenser were so constructed that the water, instead of being heated to 90 degrees Fahrenheit, were heated to only 70 degrees, which is a very fair figure for the old type of jet condenser, twice as much water would have been used per pound of steam. Thus twice as much air would have been brought into the condenser every minute and the vacuum pump would necessarily have been twice the capacity in order to remove the air and maintain the vacuum.

Eccentric Movement on Old Engine

The accompanying illustration shows a peculiar design of steam engine. The cylinder is fitted with one exhaust and two steam valves, the riding-cutoff idea being used. The engine contains a name-plate on which it is stated that it was built by Samuel Secor & Co., 1868. The plate also contains the name, East River Iron Works.

The inner, or exhaust valve, has two valve stems which terminate in a yoke piece outside of the steam chest. The valve is operated by the top eccentric rod, which connects with the top rocker arm. The steam and cutoff valves are both operated by the lower eccentric rod which is connected to the lower rocker arm. The lower rocker arm is fitted with two studs on the inner side of which the valve stems of both steam and cutoff valves are connected. This rocker arm is so designed that the steam valve has the full port travel for each revolution of the eccentric but the cutoff valve has less travel as the stud on the rocker described a smaller arc than the steam-valve stud.

The eccentric strap is of peculiar design. The eccentric is fixed permanently on the engine shaft, but the strap is designed with a link in which the link block of the eccentric rod operating the steam and cutoff valves must have originally been controlled by some type of governor, although no indications remain as to the type formerly used. The exhaust eccentric rod is attached to a stud placed at the top of the link. At the bottom of the link is a boss through which a pin passes and supports the two side pieces forming the eccentric rocker arm. This arm swings on a pin which extends from the engine bed. By this arrangement, the top of the link has considerably more throw than the bottom, hence the motion conveyed to the cutoff valve is quite rapid when opening and closing the steam port.



Peculiar Eccentric and Valve-gear Movement

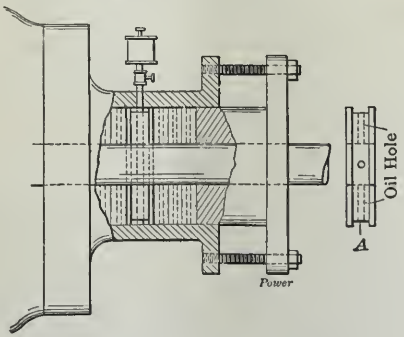
Nobody seemed to know much about this engine. Even the engineer was not familiar with the design of the exhaust valve. There are two features worthy of notice. The frame, although designed over 40 years ago, has the appearance of a modern tangye frame. The other feature is the design of a crank rod, which is of the solid-end type, fitted with a wedge and adjusting bolt. Whether this rod has been put on the engine since it was first built is not known. If not, it would appear that the builders of the engine were the pioneers of the solid-end rod.

The method of suspending the governor belt is an indication that the engine is not now run as it was designed to years ago. Now, the point of cutoff is fixed, although the cutoff valve has a detachable gear.

Lubricating Piston Packing

Here is a method of lubricating piston packing. A metal ring is made with a ¼-inch channel cut on the outside edge, as shown at *A*. Oil holes are drilled radially through the channel so that oil and graphite may find their way from the oil cup to the piston rod.

The method of applying the ring is as follows: First, two rings of regular packing are inserted in the stuffing box. Then the brass ring, which is made in two sections, is put in place, followed by as much more ring packing as is necessary to fill the stuffing box.



Device for Oiling Packing

The hole for the oil-cup connection must be drilled so as to come close to the inner edge of the groove in the metal ring when the packing is new. This will allow considerable compression of the packing rings before the groove of the ring has been forced past the oil-inlet hole in the stuffing box. The valve placed below the oil cup makes it possible to feed the oil and graphite at will.

I leaned my cheer back agin th' wall tother day an' got t' thinkin'. I'd run ingins fer nigh on t' forty years an' I'd never hed a hot box, never hed a breakdown, never cussed th' coal man when he fetched dirty coal, allus hed my salery raised ever time I'd ast fer it, hed run gas'line ingins an' never swore at 'em, an'—but jist at this pint th' dumd water glass busted an' woke me up.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Primer of Electricity

By Cecil P. Poole

Dynamo Field Excitation

(Continued)

As explained in the preceding lesson, the shunt-wound dynamo is used to a much greater extent than the series-wound machine. The reason for this is that the latter type is suitable for only one sort of service, that of supplying a circuit of devices connected in series with each other, as illustrated by the elementary diagram, Fig. 71. In such a circuit, every device must take the same current that passes through every other device in the circuit, so that the only way to make one take more *power* than another is to have it take a higher voltage.

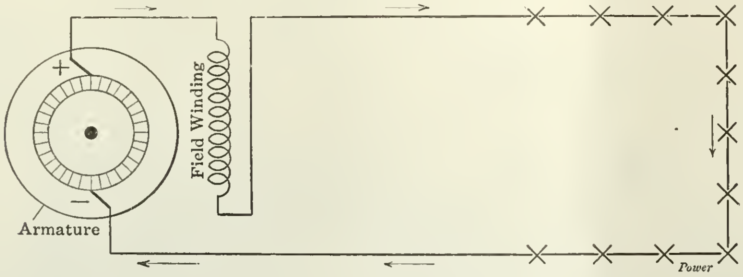


Fig. 71. Elementary Diagram of Series System

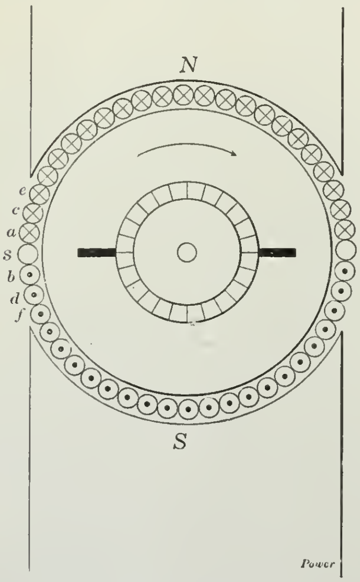


Fig. 75. Armature Currents with Symmetrical Field

In other words, if the rate of current flow is 10 amperes in one part of any series circuit, it is 10 amperes everywhere else in that circuit, and all of the lamps, motors, or whatever forms the "load," must be able to pass 10 amperes. Now, if one device is expected to use 100 watts, then it must use up 10 volts of the electromotive force generated by the dynamo, whereas another device, requiring 500 watts for its operation, must take 50 volts, because the current is restricted to 10 amperes.

It is obvious, then, that the distinction between the parallel or multiple circuit supplied by the shunt-wound dynamo and the series circuit supplied by the series-wound dynamo is that in the former, all of the receiving devices, such as lamps and motors, require the same voltage and the current taken varies according to the power required, whereas in the series circuit all of the devices take the same current and require different voltages according to the power they utilize from the dynamo.

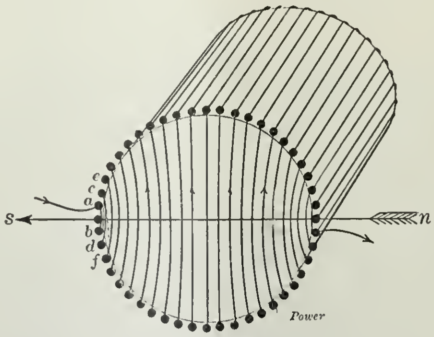


Fig. 76. Electrical Equivalent of the Armature Winding

The Series-wound Dynamo

From the foregoing, it should be evident that the series-wound dynamo must be able to maintain a constant current in the load circuit and to vary its voltage according to the demands of the load. If the rated current of the system is 10 amperes and the load requires 5000 watts (5 kilowatts), then the dynamo must generate 500 volts plus enough extra volts to overcome the resistance of its armature winding and of the line wires which connect it to the load; if the load requires 100 kilowatts, then the dynamo must generate

$$\frac{100 \times 1000}{10} = 10{,}000 \text{ volts}$$

net, because the current must be kept at 10 amperes. This is one of the drawbacks of the series system—this widely varying pressure required. Besides that, a dynamo cannot be made to supply variable voltage and constant current satisfactorily without sacrificing a good deal in the way of efficiency.

Referring to Fig. 71, it will be evident that if some of the load should be cut out by short-circuiting it (which is the only way it can be done in a series circuit), this would reduce the resistance of the circuit, and if nothing were done to reduce the electromotive force of the dynamo the current in the circuit would increase; as the field-magnet winding is in the main circuit, any increase in the current would increase the magnetizing effect of the winding and thereby increase the voltage generated by the armature; this would again increase the current,

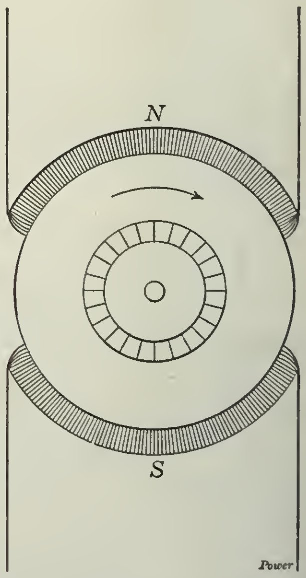


Fig. 77. Symmetrical Magnetic Flux

which would still further increase the magnetization of the field magnet, and so on, until a condition of equilibrium was reached, at which any further increase in the magnetic flux of the dynamo would require a larger increase in current than the increased flux would enable the armature to supply, as described in the case of the shunt-wound dynamo last week.

The series-wound dynamo, then, if built at all on the lines of the shunt-wound machine magnetically, would tend to increase its voltage and, therefore, the line current whenever the load was reduced, instead of reducing the voltage so as to keep the current constant. For this reason, series-wound dynamos are not proportioned magnetically at all like shunt-wound machines. The field magnet is worked at much higher magnetic densities and the armature is wound so as to exert a strong demagnetizing effect on the field magnet. Then, when the current increases, the increase in the armature winding weakens the field considerably and prevents the magnet from building up so rapidly. Even with this construction, the dynamo is not anywhere near self-regulating, and an external mechanism has to be used to change the voltage when the load changes.

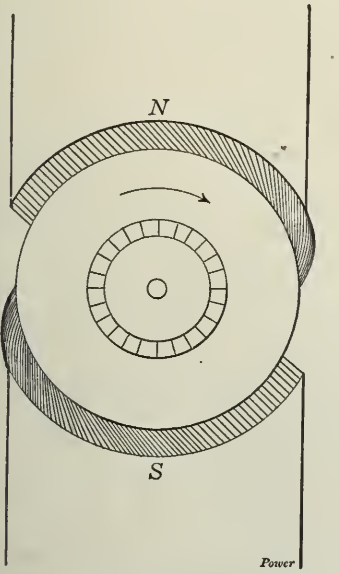


Fig. 78. Distorted Magnetic Flux

The reaction of the armature upon the magnetic field in the airgap may be understood by studying Figs. 75 to 79. Fig. 75 represents an armature having 48 conductors, for simplicity; actual armatures of this class have several hundred conductors. If the magnetic flux in the airgaps were not distorted, the brushes would be set as shown and the currents in the two halves of the armature winding would flow as indicated by the crosses and dots in the circles representing conductors; the crosses indicate current flowing away from you and the dots represent currents flowing toward you. The two clear circles at *n* and *s* represent the coils short-circuited by the commutator brushes at the moment. Now, a winding with current flowing through it as indicated here will make a magnet of the armature core, exactly as the winding on the field magnet makes a magnet of that core. The result, magnetically, will be exactly the same as though a coil of wire were wound around the core in the way shown in Fig. 76, and the direction of the magnetic flux that such a winding would produce, if undisturbed, would be as indicated by the large arrow in that diagram. Consequently, the point on the armature core marked *n* in Fig. 75 would be a north pole and the opposite point *s*, a south pole.

Two distinct magnetic fluxes cannot exist in the same space or material, any more than two distinct currents of water can exist in the same stream; they are bound to mix and form a resultant current. Therefore, when the armature winding sets up a magnetic force at right angles to the magnetic force of the field magnet, the two produce a *resultant* magnetic force having a direction somewhere between the two directions of the original forces. That means that the magnetic flux passing between the polefaces and the armature, instead of going straight across and being evenly distributed, as indicated in Fig. 77, will be distorted into something like the distribution indicated in Fig. 78, which will have the same effect on the armature winding as though the field magnet had been twisted out of line. This will shift the neutral point on the commutator, and the brushes will have to be set further around, somewhat as shown in Fig. 79.

Now, supposing that the field distortion caused the brush setting shown in Fig. 79 to be correct for a current of 10 amperes in the armature (5 amperes in each half), the wires *g, h, i, j, k, l, u, v, y, z* are exactly the equivalent of a coil of five turns wound around the armature core so as to cross the ends parallel with the line *x, x*, and carrying a current of 5 amperes. The current in these wires produces a magnetic force directly opposed to that of the field magnet, and any increase in the armature current will exert a direct demagnetizing effect on the field and tend to prevent the increase by reducing the voltage generated in the whole armature winding. Moreover, an increase in armature current would further distort the resultant magnetic field in the airgaps and reduce the inductive effect on the armature conductors as a whole, still further increasing the tendency to keep down the increase in current.

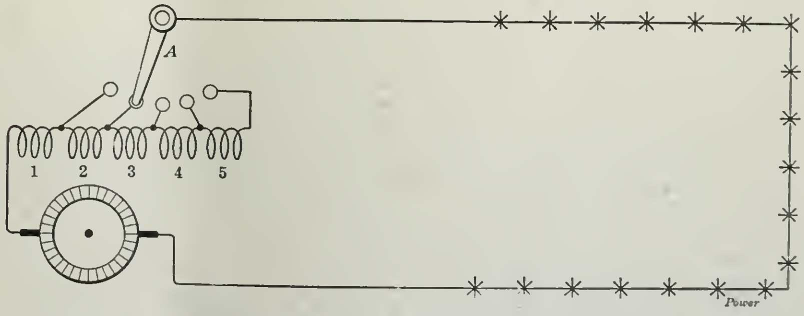


Fig. 80. Regulation by Varying Field Turns

As already stated, however, armature reaction alone cannot make a series-wound dynamo entirely automatic—that is, enable it to keep the line current actually constant, although it assists considerably in that direction. In order to keep the current constant, either the field excitation must be reduced when the load decreases or else the brushes must be shifted around the commutator to a point where the e.m.f. generated in one part of the winding is neutralized by an opposite e.m.f. generated in another part of the same half of the winding.

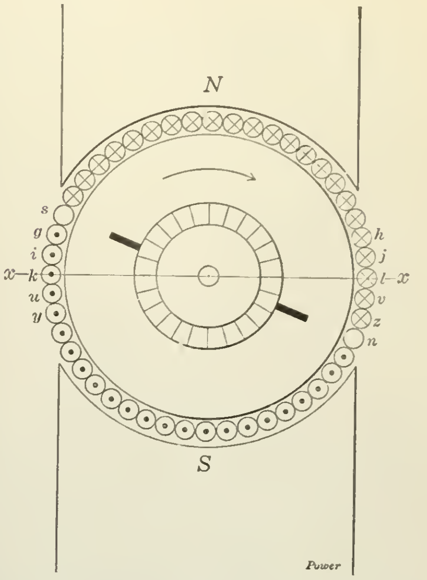


Fig. 79. Armature Currents with Distorted Field

Adjusting the Field Strength

The excitation of the field magnet may be adjusted in either of two ways, by cutting parts of the winding in and out of circuit, or by shunting more or less of the current away from the winding. Fig. 80 illustrates the method of cutting in and out of circuit parts of the winding. With the arm *A* resting as shown, the current passes only through sections 1 and 2 of the field winding, the other three sections being out of the circuit; by moving the arm one point to the right (to the third button) section 3 of the winding would be included in the circuit, and the magnetizing effect would be increased, would be 300 turns active, and a current of 10 amperes would produce 3000 ampere-turns, or 50 per cent. more magnetizing force than before. This method of regulation is never used on machines of appreciable size, because of the flash-

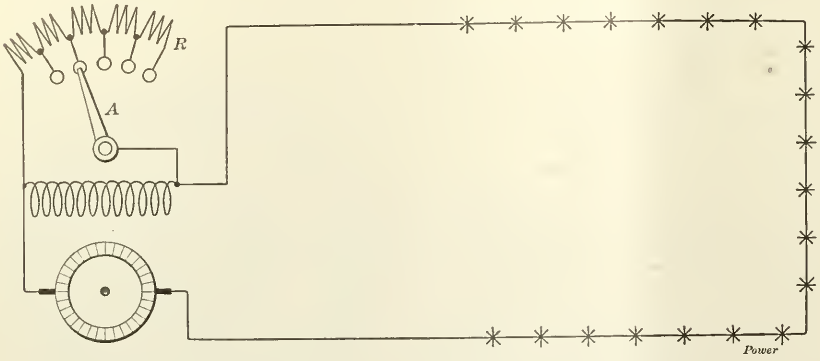


Fig. 81. Regulation by Shunting Field Winding

because the magnetic force of a winding "carrying" a given current is directly proportional to the number of turns or convolutions in it. For example, if each section of the winding contained 100 turns, sections 1 and 2 would contain 200 turns and a current of 10 amperes through them would give 2000 ampere-turns (the magnetizing force of a coil is directly proportional to its ampere-turns); then, if section 3 were cut into circuit by moving the arm to the right one step, there

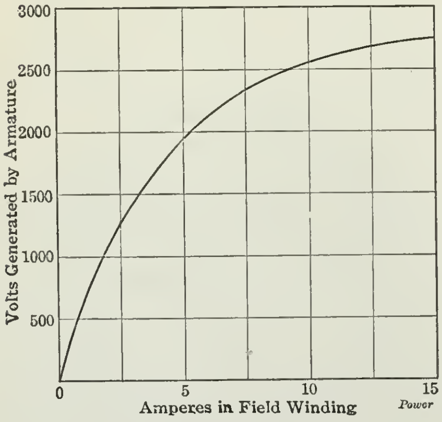


Fig. 82. Excitation Curve

ing that would occur between the arm and the contact button upon moving the arm away from a button in such a direction as to cut out the section connected to that button—from the one on which it rests in Fig. 80 to the one at the left, for example. The cause of this flashing will present it is sufficient to say that it is too destructive to make this method practical.

Fig. 81 illustrates in elementary fashion the method of regulating the field strength by shunting current away from the winding. A series of coils of high-resistance be explained in a future lesson; for the wire are connected to the regulating switch in exactly the same way that the field winding sections are in Fig. 80. With the arm *A* in the position shown, two sections of the resistor[*] are connected across the terminals of the field winding and take part of the current away from it. If the arm be moved to the next button on the right, the third section of the resistor will be cut in, thereby increasing the total resistance of the path through the rheostat and therefore reducing the amount of current shunted out of the field winding; obviously, if less current is shunted away, more current will flow through the winding and the strength of the magnetic field produced by the winding will, of course, be increased. For example, if the resistance of each section of the resistor were 100 ohms and the resistance of the field winding were 10 ohms, with a constant current of 10 amperes in the outside circuit and the armature, the current would divide between the field winding and the rheostat as follows:

Resistor Sections in Circuit.	Amperes in Resistor.	Amperes in Winding.	Total Amperes.
1	0.909	9.091	10
2	0.476	9.524	10
3	0.323	9.677	10
4	0.244	9.756	10
5	0.196	9.804	10

In practice the resistor is divided into many more than five sections and the resistances of the sections are usually different, in order to give a more even graduation of the current shunted out of the field winding; and the arm which cuts the sections of the resistor in and out of circuit is ordinarily moved by an automatic mechanism in response to load changes.

Fig. 82 is the excitation curve of a 2500-volt 10-ampere (more accurately, 9.6-ampere) dynamo running at slightly less than its normal speed. The current in the field winding was supplied from another source and the armature was run without any load. It will be noticed that although 5 amperes in the field winding enabled the armature to generate almost 2000 volts, increasing the field current to 10 amperes increased the armature voltage to only 2550, and 15 amperes in the field winding produced only 2730 volts at the brushes. This indicates that the iron and steel in the field magnet and armature core were "worked" at very high magnetic densities, where the permeability was low and decreased rapidly with higher degrees of magnetization. The curve was plotted from a dynamo of practically obsolete type, but it serves just as well to illustrate the principle of magnetic saturation as a curve from a modern machine would.

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[*] A "resistor" is a conductor used only to add to the resistance of a circuit, or to form a circuit useful only for its resistance. While all conductors have resistance, a conductor is not called a resistor unless its resistance is the only property for which it is used.

LETTERS

Why Does the Motor Make the Lamps Flicker?

At the plant where I am employed there is a 75-horsepower, three-phase, 60-cycle, induction motor which causes the lights to flicker when it is running at full load but not at light load, with the voltage regulator either on or off.

We also have a 60-horsepower 60-cycle motor, and some time ago, when one of the brushes for the starting resistance worked loose, that caused the lights to flicker; after it was tightened the flickering stopped. Thinking the 75-horsepower motor was affected the same way, we took off the brushes, cleaned them up and put more tension on the springs, but the lights still flicker when the motor is fully loaded.

I will appreciate suggestions from other readers of Power as to why the lights flicker when the motor is running at full load.

Louis J. Gorilla.

Ironwood, Mich.

The proper maintenance of hoisting ropes demands, says *The Irish Builder,* constant attention, and efficient lubrication is essential. A lubricant which is recommended by many authorities is formed by mixing a bushel of fresh slaked lime into a barrel of cold tar, or of pure tar and tallow. An equally good mixture is obtained from tar, summer oil, axle grease and a little finely powdered mica, the whole being mixed to a consistency that will work its way into the center of the rope and yet will not dry nor peel. The mixture must, moreover, be of such a character that the rope can be readily inspected, and it should be only thinly applied after the first application. Another mixture recommended by *Mines and Minerals* is formed of graphite or pulverized asbestos mixed with grease.

A man thet works on perpetshul motion is jist like a dog thet chases his tail—he does a lot uv hard hustlin' an' don't get nowhere.

Gas Power Department

Emergency and Alarm System for Ignition Circuits

By William T. Garlitz

The accompanying diagram shows an arrangement of ignition-circuit connections and auxiliary apparatus that I have found very useful and of a considerable protective nature. A battery is used while starting the engine up, and the ignition circuit is thrown over to a dynamo after the engine reaches its normal speed, but the change from battery to dynamo is made through a solenoid switch instead of directly by the hand switch.

Referring to the diagram: when the three-pole switch is closed upward, it connects the ignition circuit to the positive terminal of the battery through the spark coil, the solenoid switch being on its lower contacts, as shown, and thereby connecting the negative battery terminal to the ignition circuit. The hand switch, when up, opens a little auxiliary switch, the use of which will appear later.

> **Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.**

When the triple-pole switch is closed downward, it again connects the positive battery terminal to the spark coil, but it also connects the negative terminal of the switch solenoid to the negative terminal of the generator, through the wires *m m*, and the solenoid lifts its plunger, opening the negative battery connection at the contacts *B* and closing the positive generator connection to the spark coil at the contacts *G*. This is the normal operating condition. Should the generator fail for any reason to give the proper current for ignition, its failure will "kill" the solenoid of the magnetic switch and the plunger will drop again to the position shown in the diagram, closing the contacts at *B*, and thereby putting the battery back on the ignition circuit and preventing a shutdown. The contacts at *A* are also closed, and as the main switch is closed downward, the hinge *h* and jaw *j* are in contact and the alarm bell circuit is closed through a part of the battery, the circuit from the + post of the bell leading through the contacts *A*, the wire *a*, the switch points *j* and *h*, the wire *b*, the battery cells, to the — post of the bell. The ringing of the bell, of course, calls attention to the fact that the generator is not working.

At *W* are two contacts that are bridged by a metal strip when the valve controlling the water-jacket supply is open. The object of these is to sound an alarm if the water is not cut off when the ignition circuit is cut off. The latter operation is done by opening the double-throw switch, and when this is done the solenoid drops its plunger, of course, closing the contacts *A*. With the main switch open, this

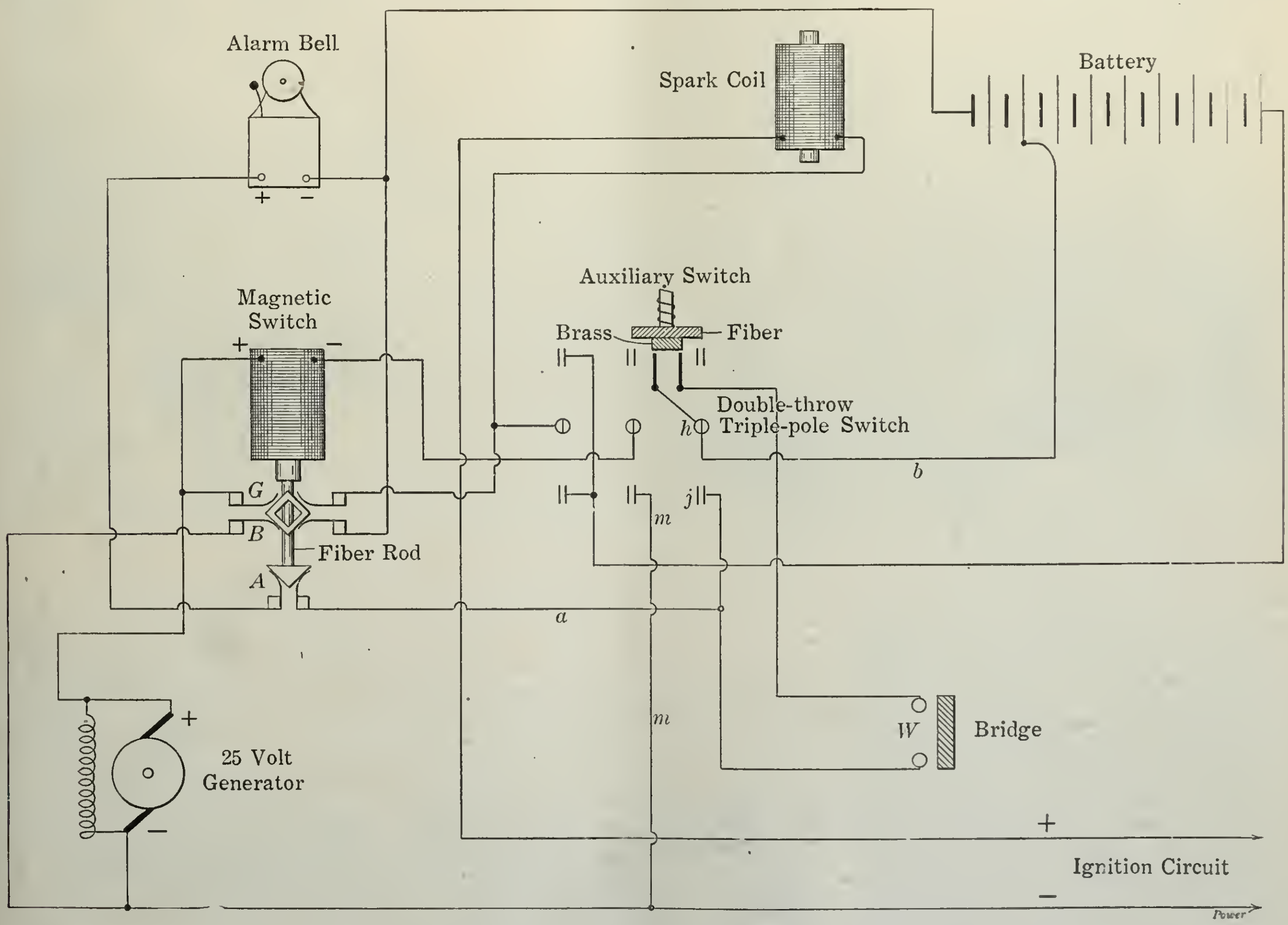


Diagram of Emergency and Alarm System for Ignition Circuits would not complete the bell circuit without the assistance of the auxiliary switch and the valve contacts *W*. If the water valve is left open, the contacts *W* will be closed, and the wire *a* will be connected to the wire *b* through the contacts *W* and the auxiliary switch, completing the alarm circuit and ringing the bell until the cooling water is shut off, which opens the contacts *W*. This part of the alarm system also serves to call attention to neglect to open the main switch, should that occur when the engines are shut down otherwise than by cutting off the ignition.

A New De La Vergne Engine

Our readers are doubtless familiar with the Hornsby-Akroyd oil engine as built for the past ten years or more. The principal characteristic feature of that engine was a vaporizing chamber connected to the cylinder by a very constricted passage; the oil was injected into this vaporizer at the beginning of the compression stroke, and the heat of the compressed air forced into the chamber by the piston, added to the heat from the walls of the vaporizer, ignited the fuel at the proper moment. Obviously, it is not an easy matter to proportion things so that the oil will be ignited at just the right moment by heat due partly to the hot walls and largely to the compression of the air, especially in large cylinders and with varying loads, which render uncertain the amount of heat that is going to be given up by the vaporizer walls. In small engines this is overcome by very careful design of the constricted passage between the vaporizer and the cylinder.

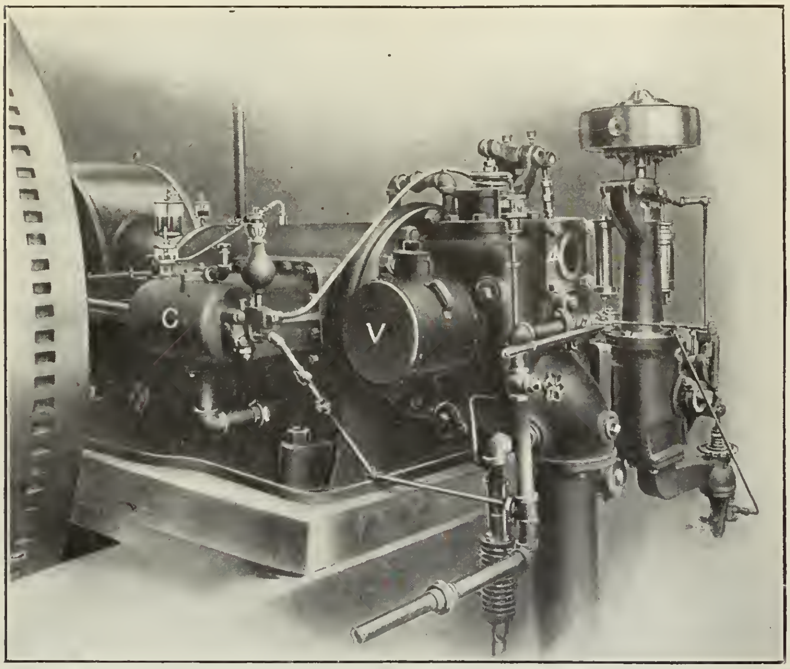


Fig. 1. Compressor Side of De La Vergne Cylinder

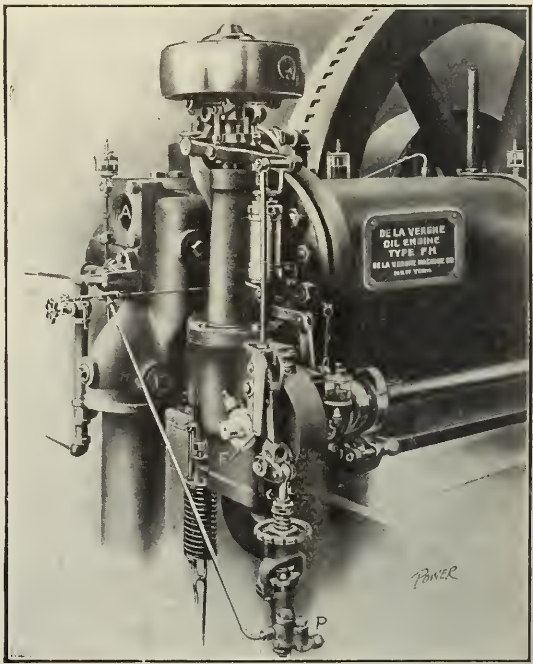


Fig. 2. Governor and Pump

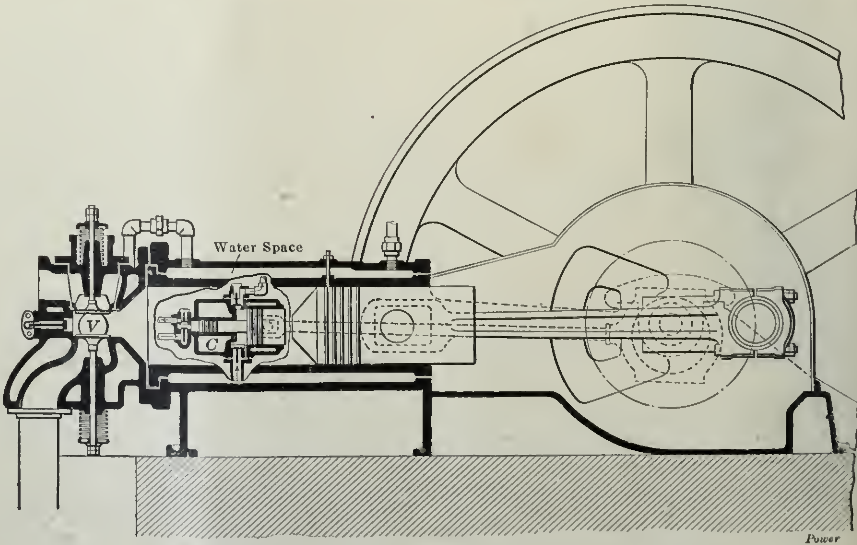


Fig. 3. Section through De La Vergne Engine

For large sizes the De La Vergne Machine Company, builder of the Hornsby-Akroyd engine in this country, has departed materially from the old design. The vaporizer opens into the combustion space of the cylinder without any constriction, the oil is not injected until the moment when ignition is desired, and the compression pressure is carried up to such a point as to insure prompt ignition by the heat of compression plus the heat given up by the vaporizer wall. The oil is injected by a compressed-air blast instead of mechanically, and a small air compressor for this purpose is included in the structure of the engine.

The construction of the engine is clearly illustrated by the accompanying engravings, from which the operating processes may also be deduced. Figs. 1 and 2 are quartering views of the rear end of the engine, in which the air compressor *C*, the vaporizer *V*, oil pump *P* and the governor mechanism are clearly illustrated. Figs. 3 and 4 are longitudinal and cross-sections of the engine, the former indicating the relation between the reciprocating parts, the latter the valve gear, and both, the cylinder construction.

The cylinder head, containing the combustion chamber and valve cages, is removable, but as it has an entirely independent water jacket no water packing is required between the barrel and the head and leakage from the water jacket to the interior of the cylinder is impossible so long as the walls remain unbroken. The vaporizer opens into the side of the combustion chamber, instead of the end, as in the old type, and there is no constriction of the opening. Diametrically across the combustion chamber from the vaporizer (see Fig. 4) the fuel nozzle is located. Whether the oil spray is blown actually into the vaporizer or is ignited before it gets across the intervening space it is difficult to decide, but the distinction is of no practical importance to the man who owns or runs the engine. The fuel is ignited when blown in by the air blast.

The nozzle is of peculiar construction.

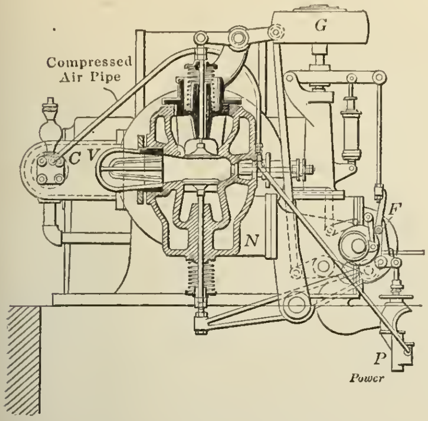


Fig. 4. Valve Mechanism

A sleeve, formed as shown at *B*, Fig. 5, fits snugly into the bore of the housing *A* and a pin valve, mounted in the housing cap shown at the bottom of the picture, fits into the bore of the sleeve *B* and is normally held by the spring *G* against a seat in the far end of the housing *A*. When the valve is drawn off its seat by a cam on the lay shaft, compressed air enters at one side of the housing *A* and oil is forced in at the opposite side, and the oil and air pass to the combustion chamber through the network of grooves around the sleeve *B*, which breaks up the oil into fine particles and mixes it very thoroughly with the air. The fuel is ignited in the cylinder, probably by the heat in the air already in the cylinder; part of this heat is due to the compression of the air in the cylinder, which produces a pressure of about 300 pounds per square inch; the remainder is supplied by the vaporizer, which retains heat from the previous combustion. For this purpose it is formed with thin ribs on the interior wall, as shown in Fig. 6.

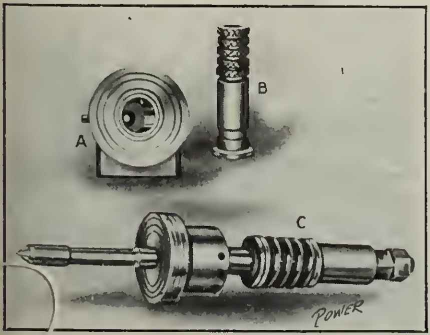


Fig. 5. Oil Nozzle and Valve

The foregoing explanation of the ignition process is based on the assumption that the fuel is ignited before reaching the vaporizer. As stated before, it does not matter whether that is true or not. If the oil does not ignite until it reaches the vaporizer, then the explanation is that the ribs of the vaporizer retain heat from the previous combustion and this heat is supplemented by the heat of compression, so that the ribs are above ignition temperature when the compression stroke is completed and the oil blown in. In either case, the air which blows the oil into the cylinder combines with it to form a combustible mixture, which burns in the favorable atmosphere of heated air in the cylinder.

Speed regulation is effected by varying the quantity of mixed oil and air admitted to the cylinder. The governor controls the air supply to the nozzle and simultaneously adjusts the stroke of the oil pump *P* (Figs. 2 and 4), thereby varying the amount of oil delivered to the nozzle. This is done by means of a movable fulcrum roller between the two cam levers which actuate the pump plunger. This roller is shown in Fig. 4 at *F*. The lever to the left of it is moved by a cam on the half-speed shaft and its motion is transmitted by the roller to the lever at the right, which forms one arm of a bell crank; the other arm is connected to the pump plunger. When the governor draws the roller upward, the stroke of the pump plunger is decreased, and *vice versa*.

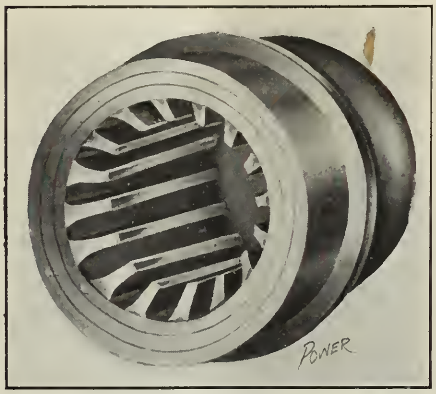


Fig. 6. De La Vergne Vaporizer

In Fig. 3 the cylinder wall is shown broken away, in order to reveal the sectional view of the air compressor. This is a two-stage machine, with two single-acting plungers arranged as shown at *C*. It is driven by an eccentric on the main crank shaft, set about 90 degrees ahead of the crank. The air compressed by the first-stage plunger is delivered to a tank at about 150 pounds pressure and this tank is automatically kept filled to that pressure; from it is taken the air for starting the engine. The high-pressure plunger of the compressor also draws from the tank and the quantity of air taken is controlled by a valve actuated by the governor in such a way as to maintain almost constant the quality of the mixture delivered by the nozzle; it is somewhat richer at full load than at no load.

The piston of the engine draws in the same quantity of air at every suction stroke, and the compression pressure is therefore almost constant. At full load this air is necessary to the complete combustion of the fuel mixture, but at light loads there is an excess. The indicator diagrams, Figs. 7 and 8, illustrate the approach to uniformity of compression pressures. The apparently higher pressure at no load is probably due to the more nearly instantaneous combustion of the charge. Fig. 8 shows a pressure rise of only 100 pounds, from compression to maximum, with a short horizontal combustion line, giving the full-load diagram area with much lower maximum pressure than the usual sharp-peak diagram would show for an equal area.

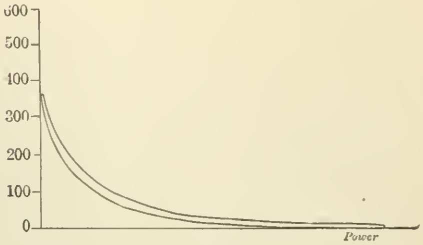


Fig. 7. No-load Diagram

The engine is built in single-cylinder form from 85 to 175 horsepower and of correspondingly doubled rating in twin-cylinder form. The guaranteed maximum consumption of crude oil, distillate or cheap fuel oil is six-tenths of a pound per brake horsepower-hour at full load and three-quarters of a pound at one-fourth to one-half of full-load rating.

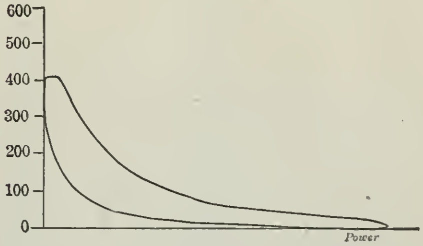


Fig. 8. Full-load Diagram

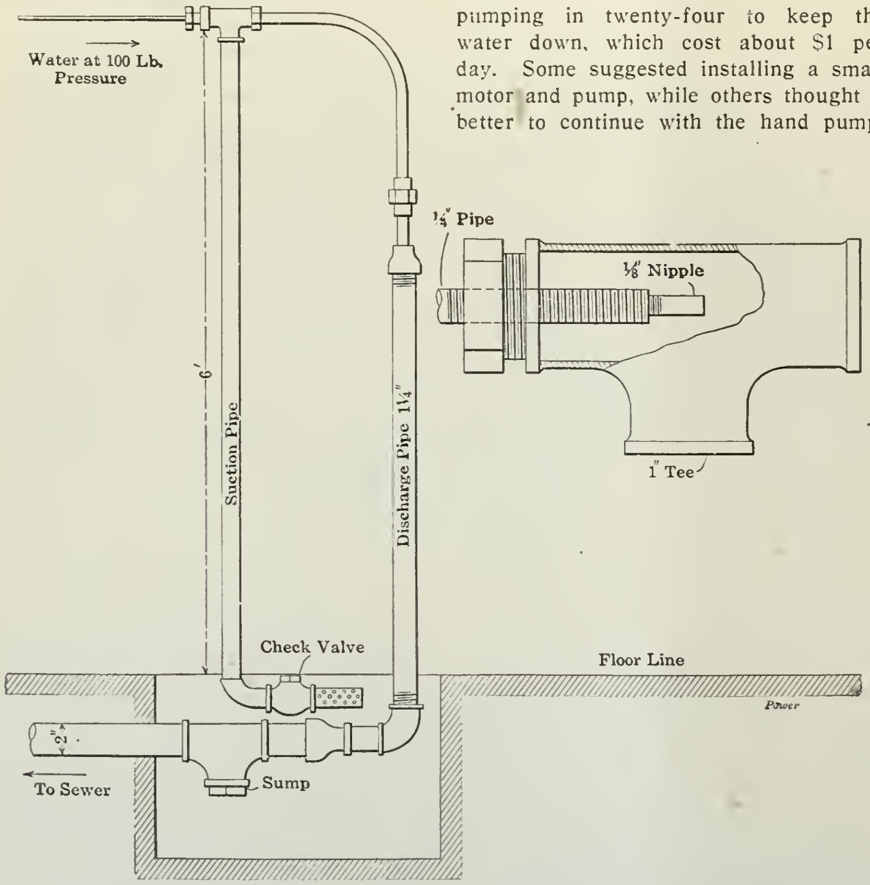
A paper read to the American Society for Testing Materials by Charles Macnichol, describes a method of obviating the trouble caused by the action of cement upon the oil in paints employed for covering concrete surfaces. The author recommends the application of zinc sulphate dissolved in an equal weight of water, the solution to be laid on by an ordinary brush after the concrete has become dry. Doctor Cushman expresses the opinion that the effect of the zinc solution is to fill the pores of the concrete with mixture of gypsum and zinc oxide, neither being injurious to oil paint.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Removing Seepage by Siphon

A building had been in process of construction for about eighteen months, and when finished it was found that the trunk sewer in the street had not been completed, and would not be opened for several months. A number of drains and some surface water emptied into the sewer and this water began to back up into the building, the installation of a back-water valve having been overlooked when the drainage piping was laid out. In the early fall the water would at times be an inch or two in depth. Not being familiar with the ground, the sewer and the layout of piping, I was at first somewhat puzzled at the rise and fall of the water, as I visited the building only once or twice a week and had not had an opportunity for close observation.



Relative Location of Siphon

It was also found that the seepage water into the basement amounted to about one barrel per hour.

When the fall rains began and the building was occupied, there was immediate complaint, as the water backing into the basement interfered with the operation of the boiler, and the building had to be kept warm at any cost. In an attempt to keep the water out the regular inlet was plugged with a plumbers' test plug and a temporary riser of 1¼-inch pipe was connected where the cleanout plug had been. A common well pump was then installed and operated by hand, and it was found that it took four hours' pumping in twenty-four to keep the water down, which cost about $1 per day. Some suggested installing a small motor and pump, while others thought it better to continue with the hand pump.

Then I decided to get the water out upon my own responsibility. I procured a 1-inch tee, a 1x¼-inch bushing, cut a long thread on a ¼-inch pipe, screwed it into the bushing, and the bushing into one end of the tee. The ¼-inch pipe, however, was tapped on the inside for a ⅛-inch pipe, and a ⅛-inch nipple was screwed through so as to slightly pass the branch of the 1-inch tee. A bend of 1-inch pipe was then fitted into the other end of the tee, and with the union it was coupled onto the 1¼-inch temporary riser. It was found that the arrangement did not work successfully until the smooth bend was put on, after which it worked nicely.

The tee was placed about 6 feet above the floor, as it was thought at that time that the water might back to that hight in the basement before the sewer could be opened. The suction pipe was then put in the branch of the tee with a check valve and strainer. The ¼-inch pipe was then connected to one of the service pipes in the building, and when the valve was opened it was found that the water could be lifted 6 feet without any trouble. The water pressure was 100 pounds per square inch.

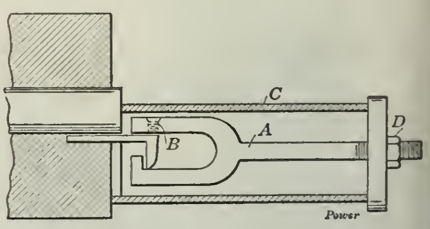
I believe that it would be possible to keep the basement dry no matter how high the water rises in the sewer, or as long as the seepage does not exceed four or five barrels per hour, which I do not anticipate. However, if more than that amount should come in, the capacity of the siphon could be increased.

H. R. Rogers.

Seattle, Wash.

Key Jack for Corliss Valve Stems

The accompanying sketch shows a convenient method of removing the key from the valve stem of a Corliss engine. The hook *A* is made of soft steel and contains the set screw *B* the head of which is flush with the jaw of the hook. The collar *C* is a piece of pipe, bearing against the end of the stem and steam arm, and cut to fit these so that it will stand out when the key is being drawn. By turning the nut *D* the key may be gradually drawn out.



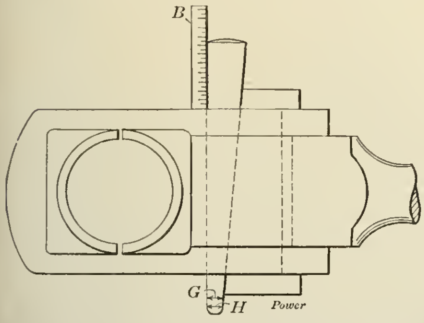
Key Jack

C. W. Greenleaf.

Lynn, Mass.

Driving Keys

When driving keys it is almost universally the custom to make a scratch with a knife or some other sharp instrument, thus permanently disfiguring the key, which, after a while, becomes scratched all over, making it almost impossible to tell which was the latest mark. When driving a key for the first time, if there is no sideplay or if the connections are so large that they cannot be moved sideways in order to determine the degree of tightness, the key should be driven solid and then backed out about a quarter of an inch depending, of course, upon the taper. The taper of the key should always be noted as the greater the taper the less the key will have to be driven.



Scale in Position for Measuring Distance Key Is Driven

While visiting an engine room recently, I became interested in the manner in which the engineer adjusted the lost motion in the crosshead and crank-pin boxes of his engine. In this instance the engineer instead of scratching the key, merely laid a rule on the strap, and taking his pencil drew a line across the key, using the rule as a guide. After the boxes have once been brought up snug it can be determined very accurately how much the key must be driven each time. This I have found with the average key to be about 1/16 inch, although in some cases 1/32 inch is all the boxes will wear; yet the inexperienced engineer often hits the pin with a blow from a hammer, as it comes around.

Another method is to take a scale or rule and set it up in front of the key as at *B* in the accompanying sketch. After driving the key, again note the reading on the scale and the distance it has been driven can be easily ascertained. When a key has been driven so far that liners are required, the thickness of the liners can be found by measuring across the key close to the bottom of the gib as at *G*, and then again at the bottom of the key at *H;* the difference between these measurements will be the thickness of the liner necessary to bring the bottom of the key up to the bottom of the gib. When taking apart to put in liners there is no way of marking the key to know just how far to drive it. When replacing again, it should be driven solid and then eased back.

It should be remembered, however, especially on engines of small clearance that the wear is continually making the rod shorter and bringing the piston nearer the crank end of the cylinder. Putting in liners carries it back, and as both parts of the box have been worn, part of the liners should be put on the strap side or there may be danger of knocking out the back head of the cylinder. Some builders arrange a key one side of which draws upon one side of the pin and the other side on the reverse side of the pin, in which case one shortens the rod and the other lengthens it, thus tending to equalize the wear and keep the rod approximately the same length. It is the practice among many modern builders to make solid ends and put in a wedge to be drawn up with a screw. In such cases it is necessary only to get the number of threads to the inch and ascertain the distance that one turn of the nut will draw the wedge, after which it becomes as simple as a key that is in plain sight.

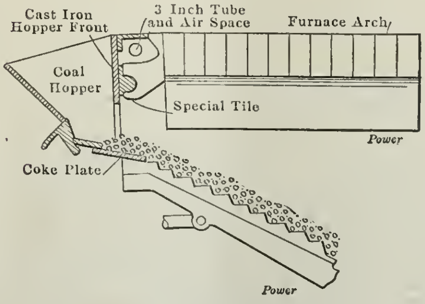
Charles H. Taylor.

Bridgeport, Conn.

Furnace Alterations

At the plant where I am employed, we have had considerable trouble from the furnace fronts of the boilers burning out. The boilers are arranged in pairs, separated by a dividing wall, and the two outer sides are finished with glazed brick. Each boiler is fitted with a mechanical stoker of the front-feed inclined-grate type.

The first cause for concern came when the glazed-brick setting at the side of the furnace front bulged and cracked. This was caused by the expansion of an angle iron, which extended the full width of the two boilers and acted as a support for the cast-iron fronts of the coal hoppers. Our bricklayer then increased the thickness of the wall at this point. This prevented the wall from cracking, but gave us another source of worry in that instead of the angle expanding lengthwise it bowed down. The cast-iron front of the hopper has a lug riveted to it which acted as a support for the special tile shown in the sectional view.



Section through Furnace

The groove in the tile engaged the lug, and the other end rested against the furnace arch. When the angle bowed, the tile fell below the arch, exposing the cast-iron front to the heat and resulting in a burnt plate. This allowed the coal to fall into the furnace and be wasted, besides involving a considerable expense for repairs. The remedy was effected by inserting a piece of 3-inch boiler tube in the wall, as shown in the sketch, and having the tile placed with a small space between. This allows the draft to draw air through the tube, over the tile and into the furnace through the cracks. This circulation of air keeps the angle and tiles cool and the cost of repairs is reduced to a minimum.

Although some of the experts on CO_2 will no doubt say that we are wasting fuel, the waste resulting from the unburnt coal that found its way into the furnace through the burned-out front and the expense of repairs far outweighed the reduced economy from the slight reduction in CO_2 caused by the extra air.

One of the boilers was repaired in this way eight months ago and the results have been so satisfactory that we are effecting the same change in the remaining boilers.

H. R. Blessing.

Philadelphia, Penn.

Removing Clinkers from Furnace Walls

Probably most engineers and firemen have experienced trouble with clinkers adhering to the side and bridgewalls of furnaces, and, although the remedy for this may be known to many, there are, no doubt, many who have not heard of it. To better illustrate it I will cite the first case of my own.

A number of years ago I took charge of the steam plant in a small hotel containing two horizontal tubular boilers, 16 feet long and 54 inches in diameter. It was in the afternoon that I took charge and before I had had time to look around the plant the fireman on watch told me that his relief had arrived and that he was going home. I at once went into the fire room and met the other fireman, who was a young man but had held his present job for some time. In view of this, I thought it best to let him do as he had been accustomed until I had become better acquainted with both the men and the place; therefore, I went into the engine room and sat down.

Before long, I heard the rattle and rasping of the fire tools and supposed that the fireman was about to clean the fires, but was soon surprised to hear a dull thud at intervals of a few seconds. I roamed around the engine room, listening, and finally went out into the fire room. The young fellow was cleaning the fire in one of the boilers and as I came around the corner he made a jab at the side wall with a slice bar and before I had a chance to say anything he withdrew the bar to arm's length and slammed it into the wall again. I at once stopped him and then looked into the firebox. The fire was pushed to one side and the side wall and bridgewall had a solid clinker about 6 inches above the grate, extending out over the grate about 8 inches, thus reducing the size of grate surface considerably. I asked him what he was doing and he replied, "I am trying to get these clinkers off the wall." I told him that he would knock down the wall by going at it in that way, and directed him to leave the clinkers there for the time being.

Upon returning from supper that evening I happened to stumble over a half barrel of oyster shells and wondered why they had been left in the passageway. As I was sitting in the engine room a few minutes later, it occurred to me that I had read in an old Farmers' almanac that a handful of oyster shells if thrown in the kitchen stove would effectually clean the brick lining. I reasoned that if such were the case they ought to clean the lining of a boiler as well. I had the fireman get the oyster shells and told him to throw a few shovelsful along the side walls and across the bridgewall about an hour before he cleaned the fire. He did as directed and in three days the walls were free from clinker. I have tried this many times since with both oyster and clam shells and have never seen it fail, even in the most stubborn cases of clinker.

W. N. Wing.

Brooklyn, N. Y.

Faulty Design

I used to envy the man who sat in an office, drew plans on paper, smoked black cigars and carried a slide rule in his pocket, but I have since changed my views, partly as a result of the experience which is related in the following.

Not long ago I took charge of a new plant which had just been finished and tested, and was ready for operation. The machinery had been installed by one firm of mechanical engineers, the heating system by another and a consulting engineer had been employed to design the plant and to superintend the installation. All equipment and material used were first class.

The first mistake made was that of installing the equipment before the building was near completion. This subjected the machinery to moisture and dirt, and consequently it was in very bad condition when the plant began operation. We learned that the plant had been tested the winter before and upon firing up, found that the piping had not been drained as it should have been after testing and consequently a number of fittings and valves burst. The water ends of both feed pumps were frozen and burst and new water ends had to be ordered. When all broken fittings and valves had been replaced, a cold snap came and steam was called for.

The overhead system of heating was used, an 8-inch pipe carrying the steam to the top of the building, a distance of about 100 feet, from which point it was distributed. Upon turning on the steam, it was found that this pipe was filled with water and that no bleeder had been provided to drain it. Consequently the pipe had to be tapped and a bleeder put in. Steam was again turned on and upon trying to start the return pump it was found that this had not been packed. The return pump and tank were located in a pit with only about 6 inches clearance between the walls of the pit and the pump, so that it was necessary either to take the pump out or to dig the pit larger in order to get at the pump to pack it. As there was plenty of waste space around the pump pit, we at once set to work and enlarged the pit.

The engines and generators were much too large for the work they were required to perform, one of the two machines being able to carry the peak load alone, although it ran for about two-thirds of the time at 30 per cent. of its rated capacity.

The exhaust pipe from the engines was carried about 3 feet below the floor, which placed it below the sewer, so that the water and oil from the exhaust steam could not be drained into the latter. In trying to remedy this condition a drain had been attached directly to the bottom of the separator and run up 2 feet into the blowoff line of the boilers. We, of course, had to disconnect this drain, put in a trap, dig a pit to catch the discharge from the trap and install a small jet to lift it to the sewer.

The steam header above the boilers was drained by a trap the discharge of which was connected directly to the sewer. This was changed and made to return directly to the boiler. The drain from the hotwell was connected directly to the house sewer; hence, when a boiler was blown down, the steam and hot water passed through the plumbing all over the building. This also had to be changed and an independent sewer put in.

The main steam line from the steam header to the engine was run straight from the header to the engines with no provision made for expansion, and the blowoff pipes from the boilers passed through a concrete floor which was cast around the pipes so that there was no room for them to move as the boiler expanded or contracted.

The water-column pipes were run directly through the furnace wall and into the side of the boiler, with no protection whatever from the fire.

The heater used for heating the water supply for the building was too small and another thousand-gallon tank had to be installed for that purpose.

A bypass had been put in around the pressure-reducing valve, as is usually done, so that the steam may be run through the bypass while the reducing valve is being repaired; but no valve had been put on the high-pressure side of the reducing valve, hence the steam could not be shut off from the high-pressure side of the reducing valve while repairing it.

The feed-water pumps were both connected to the same exhaust pipe, with no valve between the two connections. This necessitated both pumps being shut down in case the steam end of either had to be repaired.

The consulting engineer had failed to mention in the specifications the kind of service required of the generators; hence, there was a drop of about five volts every time the elevator load was put on, thus causing a bad flicker of the lights. Also, when the elevators arrived it was found that 220-volt motors had been specified, while the voltage carried in the plant was only 110. This necessitated a change in motors, costing about $475.

An ice-water circulating system was installed and the ice water was circulated by means of a steam pump, the water and steam cylinders of which were so close together that the steam cylinder heated the ice water as it passed through the pump. This pump had to be taken out and a motor-driven rotary pump installed, which effected a saving of about $1 per day in the ice bill.

Several other changes had to be made at a total cost of about $1000 before the plant could be put into successful and economical operation.

Many men are experts at putting things on paper, submitting figures, etc., but are sadly lacking when it comes to the working out of details. This type is not confined to the ranks of the consulting engineer by any means. I have seen operating engineers who could keep their engines running without a sign of a knock, who knew the steam tables by heart, who were experts with the indicator, and knew valve setting from A to Z, but who paid no attention to the little things. I have known these same men to run all day with the cylinder cocks open and have steam traps leaking steam for months at a time.

The expert is the man who is "onto his job"; it matters not whether he be a consulting engineer, operating engineer, fireman, or oiler. The man who is expert on the small job is the man who will be expert in a higher position.

R. L. Rayburn.

Kansas City, Mo.

A new use for electric welding is described in the engineering supplement of the London *Times* of September 14, 1910. A steel chimney 56 feet high and 4 feet 3 inches in diameter, was put together entirely by electric welding, and was completed before erection. It was erected in about three hours.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Economic Engineering

In the September 27 issue there appeared an editorial on "Economic Engineering." The article is good and furnishes food for thought, but I cannot say that I wholly agree with one statement made.

In speaking of the improvement of equipment and methods of operation the article reads: "The economic engineer is in a much better position to produce an effective solution of the problem than the superintendent or chief engineer, because with the latter it is often very hard to break away from long established customs, while the economic engineer is free from local prejudice and has a vast amount of data, collected from various other plants, to aid him in making intelligent recommendations."

I do not doubt that this may be true in a good many cases, but when the writer made this statement, I am inclined to think that he did not have in mind the superintendent or chief engineer who is "onto his job."

I have found a great deal more unwillingness on the part of employers to furnish new equipment with which to improve the methods of operation, than unwillingness on the part of operators to break away from "old established customs."

The man who is "onto his job" knows his plant from A to Z, as no other man can know it without having had experience in operating that particular plant. He knows the condition of every piece of apparatus in the plant and its relation to the economy of the plant as a whole. He has a system of records and reports and knows not only the total output of the plant, but also the power consumed and the exact hours of service of each machine under his charge. He not only knows the total output, fuel and water consumption, cost of labor, supplies, etc., but makes a study of the relation each bears to the other and to the expense of operation of the plant as a whole. He is able to tell each day exactly what the expense of operation was for the previous day.

He is always looking for new apparatus and ideas for the improvement of the service and economy of his plant. He visits other plants in search of new ideas, improves every opportunity of discussing mechanical topics with men of his own calling and eagerly scans all the mechanical papers he can get his hands on for any information which will make him a more efficient man and thus improve the efficiency of his plant.

He not only reads the mechanical papers but carefully studies the advertising pages in search of any new apparatus or ideas which may be applied to his plant.

It is easy to forget. One often forgets little things which might be used to advantage in his own plant. The man who means business generally carries a notebook in his pocket, in which he puts down every idea which there is any possibility of his being able to apply to his own plant.

R. L. RAYBURN.

Kansas City, Mo.

Accuracy of Water Meters

I have read with interest numerous articles in POWER in regard to the accuracy of water meters of the disk type.

In June, 1908, I wished to run a few boiler tests on the boilers in my charge and as we had no means at hand of accurately measuring the water for two 250-horsepower boilers, the superintendent borrowed two 2-inch new disk-type meters from a local water company. These meters were of a well known make and such as are used in water-works service. As two boilers were to be tested at once, one meter was temporarily connected to each boiler and the tests made with water at a temperature of 80 degrees, as hot water will warp the disk and make this type of meter unreliable.

After the tests (four in number) were made, the meters were taken down and calibrated in the following manner:

A barrel was placed on a platform scale and weighed, then the meter was connected up with a piece of 2-inch pipe and a length of 2-inch hose with a valve on the inlet side near the meter. A certain number of cubic feet of water was measured into the barrel and weighed at different rates of valve opening, the water being at a temperature of 80 degrees. The pump pressure was about 100 pounds. The different rates of flow through the meter were made by regulating the valve opening from one-quarter to full open. For each cubic foot of water registered by meter *A* the average weight was 65.8 pounds. The weight ranged from 63.1 pounds with the valve nearly closed to 69.3 pounds per cubic foot with the valve wide open.

The average weight per cubic foot registered by meter *B* was 69.1 pounds. The weight ranged from 66.3 to 76.8 pounds with different rates of flow. Both meters were new and had never been used before. In other instances also I have noticed the same thing, that is, that the meter does not register as much as it should, the error being from 1 to 11.5 per cent.

J. CASE.

Hyattsville, Md.

Boilers of Large Capacity

With a given firing method and a given type of boiler, the size of the boiler unit does not affect the efficiency at all; so the only argument for enormous boiler units is economy of space, and a slight reduction of first cost. But, on the other hand, an enormous boiler unit is undesirable for the simple reason that whenever a tube needs replacing, or whenever cleaning is necessary, an undue proportion of the boiler plant must be shut down. At Detroit, two gigantic boilers supply a 14,000-kilowatt turbine, so half the boiler plant is out of use whenever repairing or cleaning is required; while if four units of normal size were substituted for each of these boilers, only one-eighth need be shut down at a time.

With this type of boiler, where the entire ground space is occupied by the furnaces, and the water tubes and drums partly overhang above the firing floor, the division of each boiler into four single-ended units would make practically no difference in the amount of ground space required, not much difference in the first cost and no difference whatever in the capacity or efficiency.

Gigantic boilers units of practically identical size and design were installed in London in 1904; but such huge boiler units do not appear to have been adopted elsewhere in Europe, for the above reasons of convenience of small boiler units.

While the Stirling boilers at the Delray station, described in the issue of October 11, have curved tubes and 23,040 square feet of heating surface each, the Hornsby "Nesdrum" boilers in London have straight tubes arranged in nests, but in similar broad and shallow banks, and have 21,700 square feet heating surface each. The latter have shallower grates with only 336 square feet of grate surface and have a rated output of 100,000 pounds of steam hourly. This involves a higher rate of combustion than in the case of the Stirling boilers which are reported to raise 130,000 pounds of steam hourly, using stokers with 480 square feet of grate surface. The London boilers were described in Mr. Patchell's paper of December 7, 1905, to the Institution of Electrical Engineers. The general opinion in the discussion ensuing was that gigantic boiler units had no advantage over normal boiler units, and nothing has occurred in the five years since to alter this opinion.

Charles Erith.

London, England.

Does the Crosshead Stop?

In the October 4 issue the question is asked, "Does the Crosshead Stop?"

On page 172 of "Stevens' Mechanical Catechism" the following question and answer are given:

"Does a crosshead stand still at dead-center points?"

"No. The crosshead center could stand still only if the crank pin moved around it as the center. The accompanying figure shows how the reality differs from such a case. The points H and H_1 are the positions of the crosshead-pin center when the crank-pin center is at D and D_1. The circle shows the real movement of the crank pin; the two curves indicate the circles in which the crank pin would have to travel as long as the crosshead stood still.

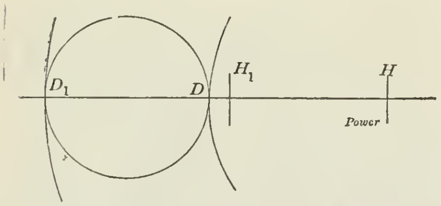


Diagram of Crosshead Travel

"The movement near the dead centers is comparatively slow, but as the crank pin does not stand still at the dead center, but is moving either toward or away from it, the crosshead, moving with it, does not stand still either. The dead center is an imaginary point having no dimensions; thus it cannot be said that the crank-pin center *remains* at the dead-center point any *time* or that it takes the crosshead any time to change its direction of stroke."

The pendulum of a clock does not stand still at either end of its arc of oscillation. No time intervenes between the end of one year or month or hour and the beginning of the next.

The question I wish to ask is, if the crosshead does not stand still, how can it stop? And, how can it change direction without stopping when it has to go back on the same line that it came out on?

H. T. Fryant.

Jackson, Miss.

Foaming in Boilers

In the issue for October 4, page 1790, it is stated that "Foaming in steam boilers is a violent ebullition which carries water in the form of foam to the steam space and main steam pipe. It can be prevented by using clean water in a clean boiler."

Is not this statement incorrect? Are there not mineral waters supposed to be clean which will cause a boiler to foam? Also, will not soft (distilled) water cause violent foaming under certain conditions?

A. L. Hoover.

Ottumwa, Ia.

One Valve Performing the Functions of Two

As to Mr. Potter's proposed scheme for making one valve perform the functions of two, described in the October 18 issue, I would say that it is perfectly feasible, and that his diagram has the problem pretty well solved. In practice, however, he will find that it is rather difficult to keep the valve tight, unless a means of adjustment is provided to compensate for the wear on the valve and its seat. This can easily be done by making the pressure plate adjustable, which he has provided above the valve.

This is not altogether a new scheme as I have operated compound engines in which the control of all of the events in both cylinders was accomplished by means of one valve.

J. M. Row.

Fort Flagler, Wash.

Feeding Boilers

The article by A. Rathman under the above heading in the issue of October 18 is, to say the least, misleading and the arrangement of feed pipes there mentioned is out of date, dangerous and one that would be condemned by the State inspectors and boiler-insurance companies in this part of the country. I am taking it for granted that he is writing about the ordinary return-tubular boiler. The practice in this State and the one that is compulsory according to the Massachusetts State boiler rules is "that the water must be discharged at about three-fifths the length of the boiler from the front head over the central row of tubes and below the lowest water line." This, I believe, is the correct way to arrange the feed pipes and from my experience I believe it to be the best. His arrangement, by which the water is discharged at a distance of only 2 feet from the front head, will certainly cause more or less local contraction of the shell plates and sooner or later he will have a leaky seam or cracked plate.

His statement that this is the coolest part of the boiler is, I think, wrong. If, when a boiler is under steam, there is such a condition of things that one part of the boiler is cooler than another, it surely is not the front end, for that is directly over the fire and should by all means be the hottest part.

How his arrangement can in any way have a tendency to lessen his troubles from scale I cannot see. In fact, to my mind, if he has water that deposits considerable scale he will have it in the most undesirable part of the boiler, directly over the fire, where it can do the most damage, causing the sheet to burn, bag or blister and the tube ends in the front head to leak.

Of course, I must admit that he may not have these troubles for my experience has been that results are not always what we think they ought to be. Still, the danger exists and it is good practice and common sense to guard against it. No one at the present time would think of feeding a boiler through the blowoff, that is, for a steady feed. Yet I have run boilers that were fed that way for 21 years and the boilers were to all appearances as good as they were the day they were put in operation. Circumstances favored the boilers in this case. The feed after passing through the heater was at a temperature of 210 to 214 degrees Fahrenheit, according to whether we were using any back pressure or not on the engine. All the returns from a high-pressure heating system entered the boiler through the same pipe and the temperature of the feed water on entering the boiler must have been very near or quite at the temperature due to the pressure carried on the boiler, 90 pounds. To say that this was the only correct and best way to feed a boiler would be foolish, for conditions in most plants would make it entirely out of the question.

In another plant that I took charge of there were four 60-inch by 18-foot return-tubular boilers. The feed pipe entered the boilers through the top of the shell about 10 inches back from the front head, dropped down to within about ½ inch of the tubes, branched to each side, with a pipe extending back about 2 feet toward the back head. This arrangement was put in by a "bright" master mechanic against the protests of the engineer. Very soon after being put in operation the girth seam over the furnace started to leak and no amount of calking would keep it tight. The night watchman was blamed for the trouble and all manner of schemes were tried to catch him putting cold water into the boilers during the night, but he steadfastly maintained that he used the injector at all times and never the city pressure. The following Sunday, after I had taken charge, the boiler-insurance company's inspector made me a visit. As I had been inside the boilers before he came and had seen the feed-pipe arrangement, I had made up my mind as to the cause of the trouble. The inspector's attention was called to the feed pipe but he would not order it changed and would only suggest that a change be made. He, like the rest, was inclined to blame the night watchman and cold water. This did not satisfy me and to prove that my idea was right I ran the pipes in one of the boilers back to within about 3 feet of the back head. I calked all of the leaky seams and under a hydraulic test found everything tight. In less than two weeks the seams in the other three boilers were leaking. The one in which the feed pipe extended to the back head was tight and, in fact, is so at the present day. After this demonstration the other three feed pipes were changed and no more trouble was experienced from leaky seams.

I have seen boilers piped in all manner of ways, being fed with all kinds of water and at all temperatures. But, for me, I want the feed water to discharge in the back part of the boiler and at as high a temperature as it is possible to get it.

S. E. Willie.

Lowell, Mass.

Engineers' Wages

Mr. Harris' letter in the October 18 number inspires a few comments.

If there is a single employer who does not realize that money invested in a *good* man is the best investment that he can make, that employer is in the "has-been" class and it is a poor proposition to remain with him even if you bluff a raise in wages out of him by unionism or any other form of coercion.

Further, if so situated, and your present employer is short-sighted or totally blind, if he has an engineer who is one of those "Johnny-on-the-job" fellows, the fact will be advertised gratis for some distance. This kind of engineer usually has several voluntary offers of other jobs per year. Do you? Mr. Harris asks what is the standard by which men are to be measured. A man is worth the prevailing wages paid for unskilled labor until he begins to think. Then, his thoughts are not worth thinking about until he begins to practice. Then, his practice is not worth much until he begins to *get results*. Then, his boss usually gets busy at the pay-envelop end before some other employer gets his man.

Will Mr. Harris cite an instance where a union organized to better conditions, including the wage scale, ever improved anything except the wage scale of the mediocre workman?

Trusts, corners or any other illegal combinations in restraint of trade are apt to be profitable to those who make them successful, that is, generally, the promoters, not the "suckers." But they are generally held in contempt, lately, and seldom referred to as things whose virtues are to be emulated. Then, let us not express a desire (even if we feel it) to imitate what our Government is even now busy prosecuting. It really does not sound nice to say, "Come on, let us do something the doing of which threatens others with iron bars and gloomy walls, and let us do it in just the same way in which they have done."

I really believe that if I had been a member of a labor union when I read Mr. Harris' article I should have withdrawn my membership. I have heard that labor unions disclaim any connection with violence, and say that such is the work of thugs, in no way interested in their cause. In fact, I have read the statements of officers of unions wherein they say that no friend of their union would do anything to reflect on the union and I have believed that they were sincere.

Mr. Harris certainly does not voice the sentiment of the majority of the readers of Power. Many prefer to feel that they have *earned* what they get, not extorted or coerced it.

W. R. Smith.

Alton, Ill.

Boiler Feeding by Centrifugal Pumps

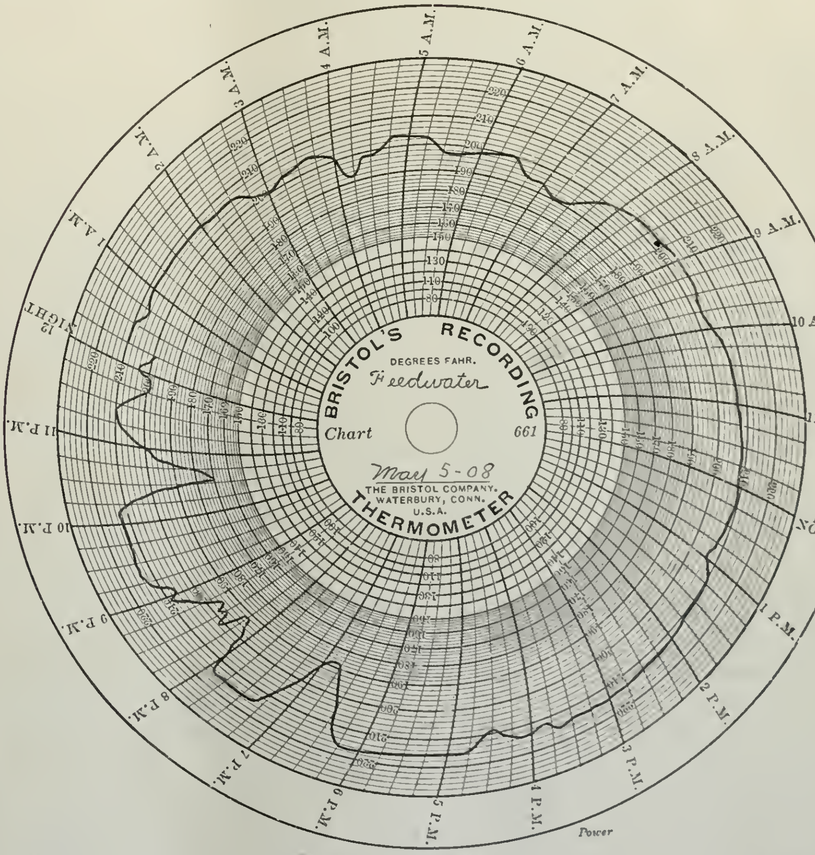


Fig. 1. Feed-water Temperature with Manual Control of Feed Valve

Referring to the editorial in the October 11 issue of Power on "Boiler Feeding by Centrifugal Pumps," it appears to me that this is very timely, inasmuch as American engineers do not seem to appreciate the advantages of the turbine-driven centrifugal pump for boiler-feeding purposes. However, there is one statement, on the use of boiler-feed regulators, to which I wish to take exception.

It is stated that "If a feed-water regulator is used, as is common in large plants, the speed of the pump is automatically controlled, but when the speed of the pump is reduced the amount of steam entering the feed-water heater is also cut down, and the water enters the boiler at a lower temperature."

From the quotation it is evident that the proposition in mind concerns a feed-water heater using only the exhaust from the boiler-feed pumps, but the fact seems to have been lost sight of that "when the speed of the pump is reduced," less water is being pumped through the heater. It is therefore evident that the amount of steam exhausted by the pump into the feed-water heater fluctuates exactly in proportion to the amount of water handled, and therefore the feed-water regulator tends to maintain a practically *constant* temperature.

In practically all power plants there are other pieces of auxiliary apparatus, such as circulating pumps, mechanical-draft fan engines, exciter engines, etc., besides the boiler-feed pump which exhausts into the same feed-water heater. Now, when an overload comes on, some of this auxiliary apparatus is also subject to a greater load and consequently more exhaust steam enters the heater. This overload at the same time makes it necessary to feed the boilers faster in order to maintain the correct level, the latter being automatically looked after by the feed-water regulator. Thus, the feed-water regulator always tends to keep the amount of water going through the heater proportional to the amount of steam in it. These are conditions which, of course, tend to higher final temperature.

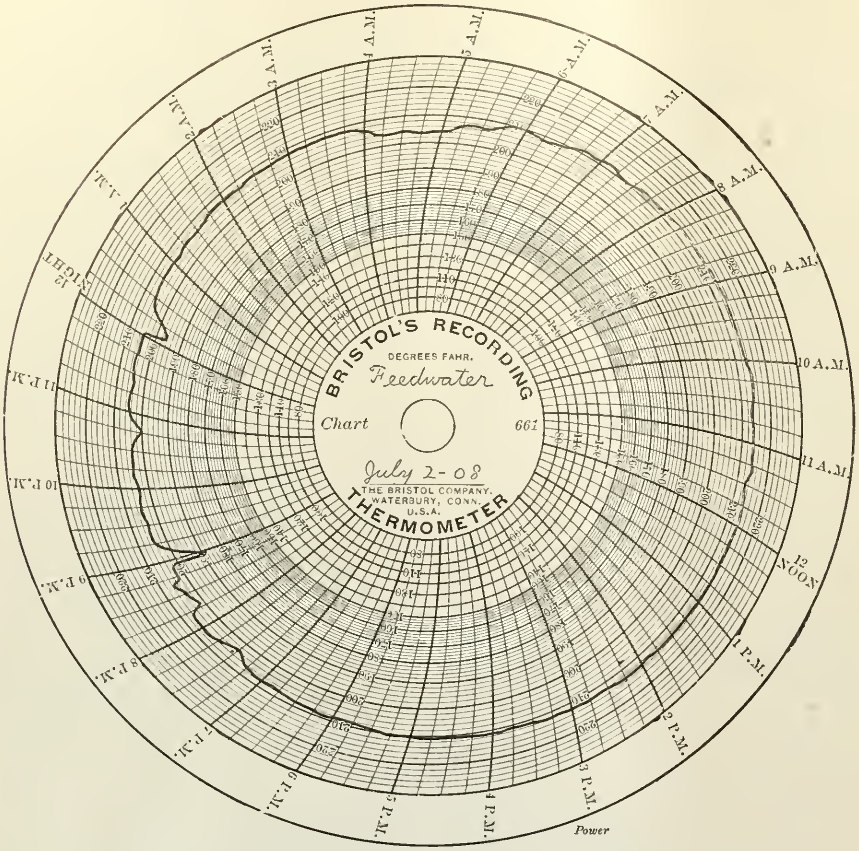


Fig. 2. Feed-water Temperature with Automatic Regulator

The above points are perfectly evident on theoretical grounds but are also demonstrated by actual operation. Figs. 1 and 2 are temperature charts for a feed-water heater. Fig. 1 shows the exit temperature of the water when the boiler-feed valves were operated by hand. Inasmuch as an attendant never operates the feed valve directly in proportion to the rate of evaporation in the boiler, there are times when the valves are opened wide and the water passes through the heater at a much greater rate than that demanded by the load; thus the feed-water temperature under these conditions is low. If the feed valve is then nearly shut off, the auxiliary apparatus still supplies steam to the heater, but as only a small amount of water passes through it the temperature becomes excessively high. This is exactly what is shown by the chart.

Fig. 2 shows the temperature readings from the same feed-water heater when the boilers were equipped with boiler-feed regulators. It is readily seen that the regulator acts in such a manner that the feed-water temperature is maintained very nearly constant. By integrating these two curves I have also found that the average temperature in the second case is nearly five degrees higher than in the first, which amply supports the statements just made. An average increase in temperature of the feed water of five degrees means a fuel saving of about ½ per cent., which, when figured out for a year, shows a substantial sum of money saved.

George F. Fenno.

New York City.

Size of Reducing Valve Required

In the October 18 number, G. S. Sprague asks advice regarding the size of reducing valve to install in a steam-heating system.

I agree with him that a 2-inch valve would be about right. With a 2-inch bypass, which should be put in to supply steam in case it should be necessary to make repairs on the reducing valve, the bypass can be used in case more steam is required than the reducing valve is capable of supplying. We have our system arranged in this manner, and frequently use the bypass on cold nights when there is little exhaust steam.

W. O. Perkins.

Bristol, Conn.

Water in the Ashpits

The question of using water in boiler ashpits, raised in the issue of October 18, is a much broader subject than would appear at first glance. The practice is not new and has been in use for many years both in stationary plants and in marine practice.

What are the advantages? In marine boilers located on boats such as harbor tugs, lighters and boats of that class, the advantage of having water in the ashpit lies primarily in the fact that in the removing of ash it eliminates dust. In the majority of boats dust can do more damage in wear and tear on the machinery than the actual work of operation, and in boats of the class above referred to the engine is frequently located adjacent to the boiler where dust from it would be a costly nuisance.

Using water in the ashpits of boilers located on land has no special general indorsement. If hard coal is used, the present standard grades of which usually contain from 5 to 10 per cent. of moisture, water in the ashpit has no beneficial effect.

Of the soft coals there are classes which might be slightly benefited by the usage of water in the ashpit, but it would largely depend on the type of grate bars on which the coal was burned as to just what that benefit would be.

My own theory of how this practice originated on land is that it is either a copy of the usage of water in ashpits of steamboats, or that it is done with the idea of merely killing the live ash as it works through the grate thus preventing the heating of the grate bars on the under side, to their detriment.

One theory of the usage of water lies in the reduction of clinker, and is copied from the feature of using steam blowers in the ashpit, which has been found to eliminate objectionable clinker and at the same time accelerate draft.

It would be quite an argument to prove that the presence of water in the ashpit gave any considerable portion of moisture to the air supplied to the grates, if dependent solely upon the water being picked up by the air in its transit from the ashpit door up through the grate bars. If, however, a very open grate bar were used and the fires worked down, the live ash dropping into the water would produce steam which would have more or less effect on the clinkering of the coal and might produce to some extent the same result that a steam blower would.

The value of using water in the ashpit, therefore, is affected by the quality of the coal burned, the grate bar used and by the thickness of the fire carried on the grates, and is also somewhat affected by the draft conditions. I believe its advantage or disadvantage to a plant could only be proved by actual tests carefully carried out. If any water is to be used I would suggest that the following features be carefully studied:

If water is put in the ashpit it should not be to a greater depth than one or two inches. The ashpit should be properly prepared so that the presence of water in it will not affect the brickwork or settings of the boiler. Its preparation should consist in having the entire pit lined with not less than one inch of first-class concrete extending as high as the bearing bars. If this is not done the water will work into the bricks of the setting and deteriorate them and the mortar in which the bricks are set.

If it be an iron setting, such as is found in some of the self-contained boilers, all of the iron should be carefully covered with the cement lining above cited, as ash and water together cause an action which will effectually injure the iron in a very short time.

CHARLES H. PARSON.

New York City.

Corn Cobs as Fuel

In response to the request in the October 18 issue for information as to burning corn cobs under boilers, I see no reason why anyone should not burn cobs with as little injury to the boilers as with coal. I have burned lots of them without bad results of any kind.

They make a quick, intense fire, which does not last long. Regulation of such a fire is best accomplished by restricting the inflow of air by keeping the ashpit doors properly adjusted, thereby obtaining a steady heat with less liability of damage to the boiler due to sudden changes of temperature such as would occur with excessive air openings in the ashpit.

I favor the ashpit regulation in the case of cobs, natural gas and shavings, more than with coal, for there is more danger of buckling the fire sheets with those fuels. With a boiler experience of 30 years and having handled most every kind of fuel, I have never had a bagged or corrugated boiler bottom. I attribute my good luck in this respect to keeping the boiler clean inside.

In burning cobs the grates will, if the firing is strong, become covered over in a day's run with a tough leathery slag that adheres tenaciously to the grates and can hardly be removed until cold.

S. E. EVANS.

Anderson, Ind.

In answer to the Western correspondent's query about burning corn cobs and slack coal together, I can give the results of 14 years' experience. My first 4 years of firing was with cobs alone and after that, cobs mixed with Jackson Hill and Hocking slack. The tubes were cleaned once a week and the ashes cleaned out of the back end of the boiler once a month. Very seldom was more than an ordinary wheelbarrow load taken from the grates for a 10 hours' run. Forced draft was used some of the time owing to a short stack. No damage has been done that we can discover that can be laid to anything but ordinary wear and tear.

The cobs make a good hot fire and it is not necessary to use a bar with clear cobs, and very seldom with cobs and coal. This is quite a corn country and in the few steam grist mills that are left they burn all the cobs that they can get.

H. M. ROWLEY.

Adrian, Mich.

About 14 years ago I worked in a plant where they fired four boilers with 75 per cent. cobs and 25 per cent. slack coal. With the kind of coal we were using, it would have been impossible to generate the required amount of steam without the cobs.

About eight years ago I took charge of a plant that had five 125-horsepower boilers, under which mine-run coal costing $1.50 per ton was being used. The cobs were being burned up in the open, for the owners were of the same opinion as our Western friend—that the cobs would ruin the boilers. I commenced using cobs and slack coal, for which we paid 80 cents per ton; a mixture of half cobs and half coal was used and after the firemen got used to handling the cobs and coal they could keep up the required pressure easier with the coal and cobs than they could with the mine-run alone.

During a miners' strike, lasting six months, we fired with nothing but cobs. In all of my experience in using cobs as a fuel I have never noticed any injurious effects to either the boiler shell, tubes or settings that I thought was caused by using the cobs.

Cobs are a little harder on grate bars than coal alone, but by keeping the ashpit well filled with water, one can offset this.

J. W. VARDEN.

Evansville, Ind.

Other letters were received on the subject of "Corn Cobs as Fuel." In all of them the opinions expressed were substantially the same as those in the letter published herewith. The writers of the unpublished letters are, S. Kirlin, New York City; R. A. Cultra, Cambridge, Mass.; W. A. Field, Anoka, Minn., and C. A. Moore, Piqua, O.

Valves and Packing to Withstand Acid

Referring to Mr. Demarle's inquiry in the October 18 issue under the above heading, I would say that probably the best results could be obtained by substituting lead valves for rubber. Valves made from lead are soft enough to prevent leakage and at the same time are very little affected by acids. For the pistons in the acid ends of the pumps either lead rings or metallic packing could be used. This metallic packing, which is shredded babbitt metal mixed with graphite and oil, could also be used to advantage in the stuffing boxes.

JOHN FRENCH.

Washington, D. C.

Jealousy and Plant Management

I agree with B. P. Gage's sentiments as expressed in his letter in the October 25 issue under the title "Why Are There Not More Writers among Engineers?" It would seem as though older engineers, by not giving younger ones the benefit of their experience, display unreasonable jealousy.

The following incident serves to illustrate the point. A fireman who was young and ambitious to progress, once asked the assistant engineer on his watch what the number of stages was of a certain air compressor in the plant. The engineer said that it was a four-stage machine. The fireman, not being entirely ignorant on the subject, doubted the accuracy of the engineer's statement and when a favorable opportunity occurred he put the same question to the chief engineer who told him that it was a two-stage compressor.

Mr. Gage inquires about handling men. I will give him the benefit of the experience I have had during five years' service in the United States Navy. I am convinced that in any plant of whatever size a fixed system should be evolved and put into effect. Perhaps the following may meet with approval:

1. Every man must be at his post ready for duty on time.
2. Waste, oil and grease to be issued only on order from the assistant engineers as required for their shifts.
3. Every man, from the first assistant engineer down to the oilers in the engine room, and from the head fireman down to the firemen in the boiler room, to have a definite amount of bright work, steel work, floor area, etc., to be cleaned by a certain hour each day for inspection by the chief engineer.
4. Keep records for each day, showing the quantity of coal and water used, the nature of all repairs made and the quantities of supplies issued.
5. Liquor to be allowed in the plant under no circumstances.
6. Each man to inform his relief as to the condition of the apparatus under his care and a description of any repairs that may have been made.

The chief engineer should have no "pets." The work should be divided with fairness and all orders should be obeyed.

W. G. VEDDER.

Fort Logan, Colo.

Boiler Explosion at Mount Victory, O.

By Osborn Monnett

Explosion of sawmill boiler of portable locomotive type, which had long outlived its usefulness, killed four men. The metal was badly crystallized. Boiler was of size not covered by Ohio inspection laws.

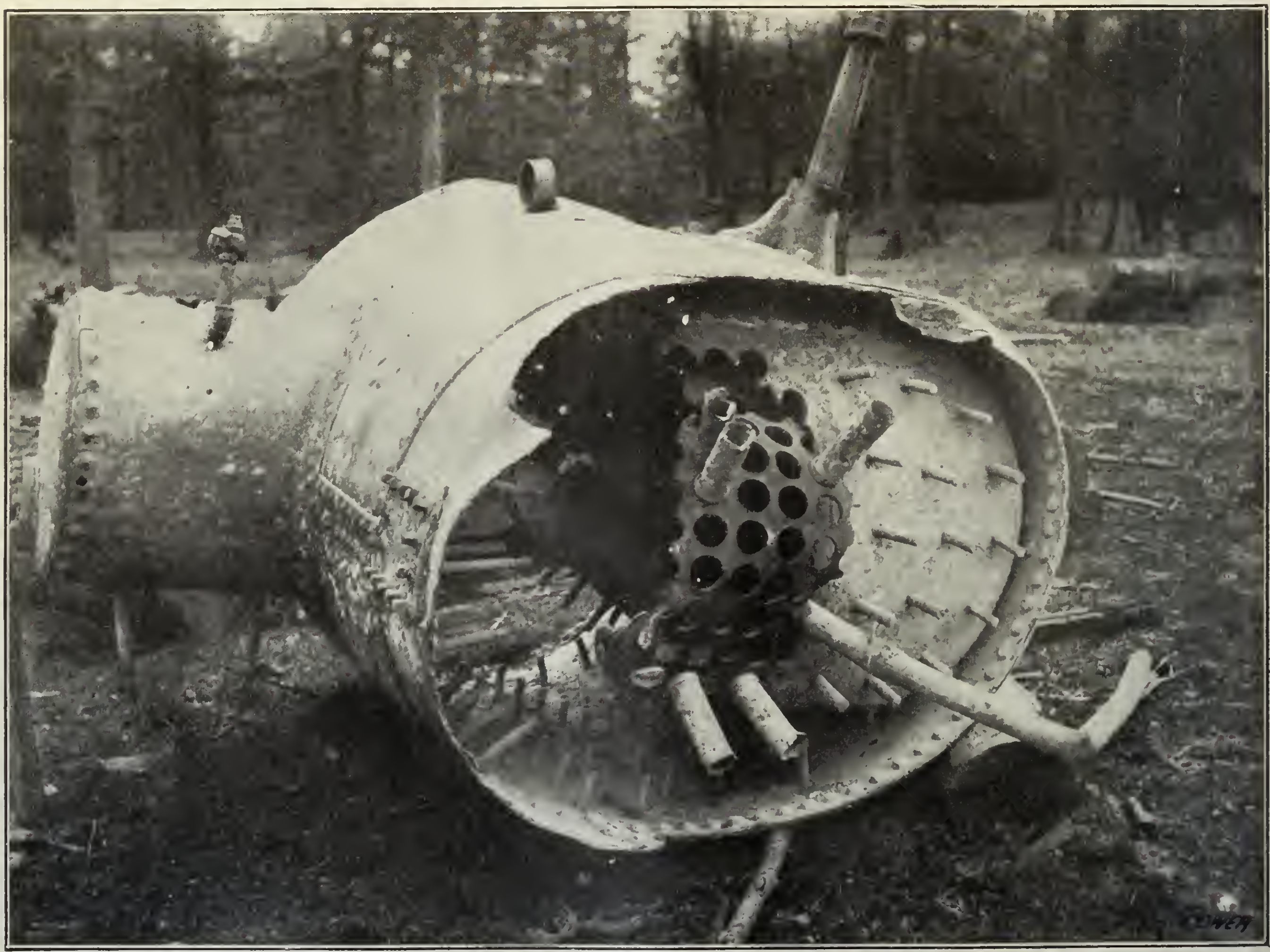
An old sawmill boiler exploded about noon, November 11, on a farm 5 miles southeast of Mt. Victory, O., a small village in the west-central part of the State. The boiler was of the portable locomotive type mounted on wheels, the axles being bolted direct to the shell at the firebox end. The engine was mounted on the boiler shell and was of the throttling type with semi-rotary valves.

The owner, who was also the engineer, had shut down for the noon hour, and on departing for a farm house ½ mile distant for lunch, remarked to one of the workmen about the mill that there was an unusually hot fire in the firebox and asked that those remaining at the mill to eat lunch keep watch of the water in the boiler.

The owner had just returned from lunch when the explosion occurred, killing him and three others. Several men who were returning to the afternoon work report that at the instant of the explosion they could see at a distance one of the mill hands pouring water upon the injector, indicating that they had experienced trouble in getting water into the boiler.

It is presumed that the water got low, overheating the crown sheet and causing the explosion. The latter, as shown in the accompanying illustration, collapsed, every staybolt pulling out at the threads. The staybolts, which were 7/16 inch in diameter, were badly corroded and at a point about corresponding to the water line had wasted away to about one-half their original area. The bolts did not let go at this point, however, but pulled out at the threads in every instance. There was no soft plug in the crown sheet and no provision had ever been made for one.

The front head (firebox end) of the boiler was blown out clean, and the barrel of the boiler tore completely around its circumference at a point near the throat seam, being thrown in an opposite direction to that of the head. The boiler was fitted with a 1½-inch safety valve of the pop variety, which, on examination after the accident, was apparently in good condition, no evidence of corrosion being seen. The metal of the boiler was in a badly crystallized condition, and where the sheets were torn, laminations of the metal were easily discernible. The outfit had been obtained second-hand by its last owner. The metal in general was in the most wretched condition and apparently the entire outfit had long outlived its usefulness. This boiler was of a size that does not come within the regulations of the Ohio license laws and emphasizes the need for closer supervision of steam boilers and operators of these portable outfits.



Firebox End of Boiler, Showing Collapsed Crown Sheet

Proposed Changes in Mass. Boiler Rules

By Benjamin Baker

A lively and at times somewhat acrimonious hearing took place before the Massachusetts Board of Boiler Rules at Boston, November 3, on a petition from certain boiler manufacturers of the State asking for certain changes in the present rules. The proposed changes were as follows:

It is proposed to omit the words in *italics* and to insert those in **black face.**

No. 1, page 47, paragraph 67. A horizontal tubular boiler over seventy-eight (78) inches in diameter (*shall*) [**may**] be supported independent of the boiler side walls from steel lugs by the outside suspended type of setting. Where three (3) supports are necessary on each side of a boiler, an equalizer shall be used.

No. 2, page 40, paragraph 32. A horizontal return-tubular boiler, having a manhole below the tubes, shall have one or more stays on each side of the manhole, the rear ends of which shall be attached to the rear end of the boiler, and the front end shall pass out through the front head and shall be secured with nuts inside and out. (*The center line of such stays at the front head shall not be below the center line of the manhole*) [**or the front end of the stays may be attached to the front head in a similar manner as to the rear head.**]

No. 3, page 46, paragraph 60. (*A manhole shall be located in the front head below the tubes of a horizontal return-tubular boiler sixty* (60) *inches or over in diameter*).

[**The placing of a manhole in the front head below the tubes of a horizontal return tubular boiler shall be optional.**]

No. 4, page 21, paragraph 5. The rule states that the tensile strength in pounds per square inch of firebox steel shall be from 52,000 to 62,000. The amendment proposes to substitute the figures 55,000 to 65,000.

No. 5, page 21, paragraph 3. "There shall be three classes of open-hearth boiler plate and rivet steel; namely, flange or boiler steel, firebox steel and extra-soft steel, which shall conform to the following limits in chemical composition." The suggested amendment recommends that no quality of steel shall be used of a less composition than that required of fireboxes, or that the rule be so amended that on and after January 1, 1912, no flanged steel shall be allowed in any plate or head of a boiler.

No. 6. Suggests amending the same paragraph by making the acid steel contain only the same amount of phosphorus as the basic process.

No. 7, page 12, paragraph 1, section VI. When a boiler is tested by hydrostatic pressure, the maximum pressure applied shall not exceed one and one-half (1½) times the maximum allowable working pressure, except that twice the maximum allowable working pressure may be applied on boilers permitted to carry not over twenty-five pounds pressure per square inch, or on pipe boilers.

[**the minimum pressure shall be one and one-fourth (1 1/4 times the working pressure.**]

No. 7, page 31, paragraph 4. The section, which is too lengthy and involved to quote, relates to the staying of boiler heads by the use of angle irons, and the amendment suggests the addition of examples or rules for '25 and 150 pounds pressure both for the 30- and 36-inch boilers.

No. 9, page 48, paragraph 9. The feed water shall discharge about three-fifths (⅗) the length of a horizontal return-tubular boiler from the front head (except a horizontal return-tubular boiler equipped with an auxiliary feed- water heating and circulating device), and at about the central row of tubes above the tubes when (*the diameter of the boiler exceeds thirty-six* (36) *inches, and*) the pressure allowed exceeds twenty-five (25) pounds per square inch. The feed pipe shall be carried through the head or shell with a brass or steel-boiler bushing, or the opening reinforced, and securely fastened inside the shell above the tubes.

this rule does not apply to boilers thirty-six (36) inches and less in diameter.]

No. 10. Also advise the best method of feeding water into boilers of thirty-six (36) inches in diameter and less when made for working pressures exceeding twenty-five (25) pounds and not having a manhole.

It appears quite certain that the first four amendments and also Nos. 7 and 8 will not be adopted by the board. Articles 5 and 6 met with no opposition from any source, and are not unlikely to be adopted. Articles 9 and 10 are aimed chiefly at more definitely defining the required practice in order to end some slight controversies over the interpretation given by the inspectors of the board to the rules.

The interests of the operating engineers and the outside public were effectively presented at the hearing by representatives of the National Association of Stationary Engineers and the International Union of Steam Engineers.

The hearing opened with a lively discussion as to just what interests were represented by the petitioners. It was finally made clear that the changes were asked for by a committee of five, consisting of one member from each of the five large boiler-manufacturing firms in Massachusetts. The New England Boiler Manufacturers' Association in April, 1909, appointed a committee on boiler rules; this committee in turn appointed as a subcommittee, with full power to act on the Massachusetts rules, the five members of Massachusetts firms who represented the petition given above.

It was made clear therefore that the petition represented the wishes merely of five Massachusetts manufacturers, and that the suggested changes did not have behind them the authority of the New England Boiler Manufacturers' Association.

No technical reasons were presented by the petitioners in support of the first three changes. For the first amendment the only reason advanced was that in a few cases the requirement of the outside suspension type of setting had made it impossible to enlarge a boiler plant without moving one or more of the existing boilers. The operating engineers present objected to any weakening of the present rule for mere reasons of convenience such as were presented.

In favor of the second change the only reason was the asserted fact that certain engineers designed boilers with the lower tubes in such relation to the manhole that the stays had to be put below the center line of the manhole, a practice which is forbidden by the Massachusetts rule. Discussion very quickly brought out two strong points against the amendment; first, that a lower position of the stays than that now required would weaken the boiler head, and second that the adoption of the amendment as to the method of attaching the stays to the front head would leave the rules entirely without specification as to the manner of fastening at either head.

Strenuous opposition from the operating engineers was aroused by amendment No. 3. It was argued by the engineers that a manhole in the front head below the tubes of large tubular boilers is absolutely essential to proper inspection and cleaning of the interior of the boiler; and was required for the safety of both the operating men and the general public. On this question, Mr. Stevens, of the board, representing the boiler-using interests, referred to articles on boiler explosions in the November 1 issue of POWER, which he called the leading engineers' paper of the country. Mr. Stevens declared that the chief article of faith of the operating engineer was that boiler safety lay in the engineer's ability to inspect every part of the boiler inside and out. The adoption of amendment No. 3 would remove from the rules so much of the guarantee of safe operation as is secured by the existence of such manholes in large boilers, and of the internal inspection which these manholes permit.

On the question of raising the tensile-strength limits merely for firebox steel, the main reason in favor of the proposed change appeared to be the fact that the steel mills sometimes found it inconvenient to meet the limits, 52,000 to 62,000 pounds, imposed by the present rules. Mr. Stevens, of the board, referring to a recent visit he had made to some of the Krupp mills at Essen, reported that the new German Blue Book of boiler rules, applying to boilers for all pressures, set a maximum tensile-strength limit of 56,000 pounds. German practice had shown, it was said, that in boiler plates of higher tensile strength there was a dangerous liability to cracking. Mr. Stevens also referred to the practice of the Babcock & Wilcox Company in adopting 56,000 pounds as the maximum tensile strength used in calculations.

Under the present Massachusetts rule the tensile strength of plates as they are actually turned out by the mills ranges from 56,000 to 60,000 pounds. German practice is somewhat more rigid than that of Massachusetts; and it appears that the tendency of the best practice is to preserve a sufficient degree of ductility by limiting the tensile strength.

Amendments 5 and 6 while they preserve certain opportunities to boiler manufacturers, are in the direction of securing better material for boilers. At present, basic steel is used almost exclusively in Massachusetts boiler manufacture for the reason that it is very difficult to produce acid steel with the low phosphorus and sulphur content, 0.04 in each case required by the rules for firebox steel. Occasionally, however, and a specific example was presented from the experience of one of the petitioners, a boiler designer will specify acid steel with the low phosphorus and carbon content required of firebox steel. In bidding on such specifications one manufacturer sometimes has acid plates of the required quality, and the amendment was designed to give such manufacturer the advantage in bidding to which he was entitled by his having a particular steel specified, especially since the specification of this particular acid steel gives a plate as high in quality as the grade of firebox steel provided for in the existing rules.

Amendment 7 was not favored by the operating engineers, and, in fact, little or nothing was advanced in support of the change. The present rules prescribe no minimum; and the use of a hydrostatic test at all is left to the judgment of the State inspectors. It was pointed out that the proposed minimum pressure was so much higher than the allowable working pressure on many boilers that might be tested, that the application of the proposed minimum test pressure might cause weakness in the boiler that would be recognized only after this weakness had caused an accident.

Amendment 8, if adopted, would have the effect of making the rules accept internal angle bracing for the heads of boilers above the tubes when the working pressure is as high as 150 pounds. The rules now limit the use of such bracing to boilers not over 36 inches in diameter and with a working pressure not over 100 pounds. The board holds that such bracing is not adequate on large boilers at higher pressures than 100 pounds; since angle bracing is used only on boilers which are without manholes, and which, therefore, do not permit of internal inspection.

Amendment 9 was professedly aimed at removing certain variations in the way the State inspectors interpret the existing rules; but the adoption of the amendment, which is quite unlikely, would remove from the rules the present requirement that "the feed pipe shall be carried through the head or shell with a brass or steel boiler bushing (or the opening reinforced) and securely fastened inside the shell above the tubes." Comment by a member of the board showed that the board would not permit the removal of this part of the rule, since without it feed pipes might be made to depend for support wholly on the threads of an opening in the head or shell—a scheme which is considered inadequate.

Amendment 10 seeks an expression of opinion from the board as a guide both for the makers of boilers and for the inspectors in the case of boilers of smaller diameter than 36 inches; these are not now specifically covered by the rules.

Large Pumping Engine for Wheeling

> *The engine is of the vertical triple expansion crank and flywheel type with steam cylinders 42, 74 and 110 inches in diameter and water plungers 33 inches, the low-pressure steam cylinder being the largest ever used on a pumping engine.*

There has recently been completed, in the shops of Allis-Chalmers Company, one of the most interesting pumping engines ever built. In point of power it practically equals the largest, that built by the same company for the city of Nashville.

The new engine is for the city of Wheeling, W. Va., and is of the vertical triple-expansion crank and flywheel type with steam cylinders 42, 74 and 110 inches in diameter and water plungers 33 inches in diameter. The low-pressure cylinder is the largest cylinder ever used on a pumping engine. The cylinders are all steam jacketed. Steam for the jacket of the high-pressure cylinder is bypassed around the admission valve. That for the intermediate and low-pressure cylinders is taken from the first and second receivers, respectively. The steam from the jackets of the high-pressure and intermediate cylinders is trapped into the second receiver and does useful work in the low-pressure cylinder.

Admission valves on the high and intermediate cylinders and the exhaust valve of the high-pressure cylinder are of the Corliss type while the exhaust of the intermediate and both admission and exhaust of the low-pressure cylinder are of the poppet type. The valves are driven from eccentrics on a lay shaft supported on brackets on the main frame of the engine. This lay shaft is driven by drag cranks and connecting rods from each end of the main shaft.

The engine is designed for a steam pressure of 125 pounds and a vacuum of 27 inches. The stroke is 72 inches and at the regular speed of 18⅓ revolutions per minute the piston speed will be 220 feet per minute. Two flywheels, each weighing 90,000 pounds, are provided. The condenser is of the surface type and is located in a bypass of the discharge line of the pump with cutoff valves on either side. The engine is provided with a free exhaust for use when necessary. The condenser pump is driven from an arm on the low-pressure plunger.

The valve chambers are 72 inches in diameter, which are the largest ever built, and are of cast steel. The pressure chambers are also of cast steel. The valve deck is steel and the 186 valves in each chamber are placed directly in the deck without the use of cages. The valves are of a special weighted type which obviates the need of springs.

The water end of the pumping engine is self-contained and self-supporting. The steam cylinders are supported on double A frames, one end of which rests on the water ends and the other on a pier. The pit in which the water end is located is 46 feet deep and the total hight of the engine from the base of the water end to the top of the steam end is 84 feet 8 inches.

Each piston has a single piston rod which connects with a cast-steel crosshead into which it is screwed and held by a lock nut. Four steel distance rods

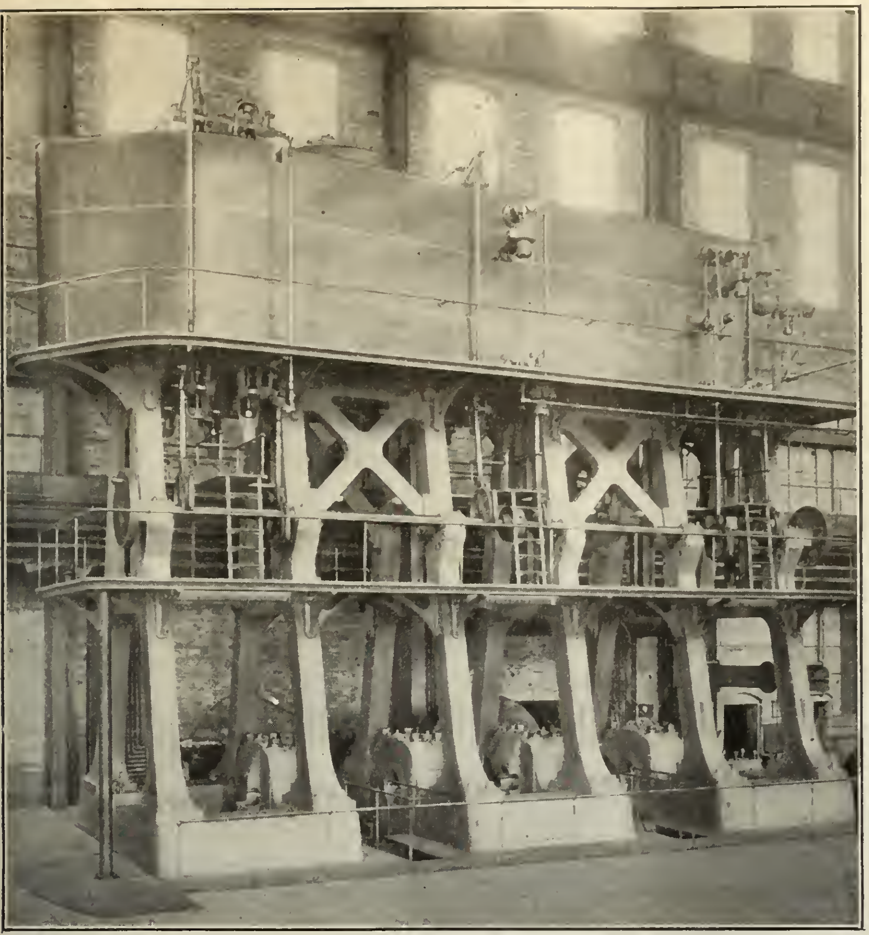


FIG. 1. STEAM END VERTICAL TRIPLE-EXPANSION PUMPING ENGINE

connect the steam crosshead with the water crosshead which is of cast steel. This is connected to the water plunger by means of heavy cast-iron pipe sections fitted with adjustable guides.

Galleries are provided at the level of the valve deck and at the plunger chambers in the pit. Above the engine-room floor are two main galleries. The lower gives access to the crossheads, eccentrics, etc., while the upper allows inspection of the valve gear and permits the use of the indicator motion. The tops of the cylinders are connected by platforms for reaching the upper poppet valves.

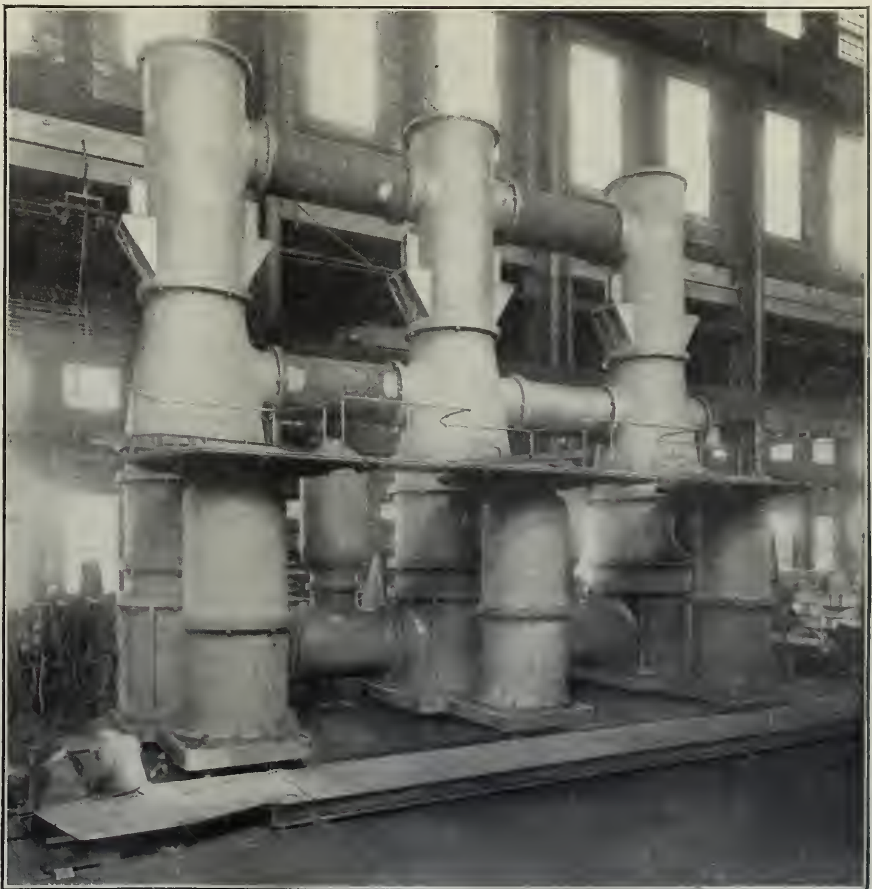


FIG. 2. PUMP END OF UNIT HAVING DAILY CAPACITY OF 20 MILLION GALLONS

Hen Simpkins, th' ingineer uv th' creamery, got a overdose uv buttermelk an' sez he'z swore off. Wun uv th' schule marms kum in ter his ingin room last week an' after she'd watched th' dashpots uv th' ingin jumpin' up an' down fer awhile she pinted at 'em an' ast Hen how it wuz posserbil fer them t' make so much butter with jist them two little churns.

Hen wuz so gol darnd flabbergasted thet he tipped over backards an' fell inter a tank full uv buttermelk. He says he ain't hed any hankerin' fer th' dumd stuff since, an' thet ever time he sees a woman headin' fer th' ingin room it gives him a chill.

Third International Congress

The second international congress of the refrigerating industries, which has just closed a successful meeting at Vienna, Austria, decided to hold the next, or third international congress, in the United States, in 1913. This will offer a great opportunity to all those in the United States who are interested in closer commercial relations with other countries in those products that are best handled at low temperatures. It will enable them to meet experts from many foreign countries, and probably many representatives of foreign governments.

It will enable the refrigerating engineers and experts in this country to meet men engaged in similar pursuits in other countries, and learn from them in what way many of the ordinary or extraordinary difficulties in the construction and operation of refrigerating plants are met or overcome in other lands. It will offer opportunity to exchange experiences with those who have studied refrigerating problems from standpoints somewhat different from those which obtain in America.

The burden of responsibility for suitably entertaining the refrigerating experts of the world is no light one. However, so many departments of commerce and industry are directly or indirectly interested in the science and practical application, in one form or another, of artificial refrigeration, that assistance from most branches of industry may be expected.

Inquiries of General Interest

Duplex Pump Valve Setting

What effect will increasing or decreasing the amount of lost motion in duplex-pump valves have on the action of the pump?

D. P. S.

Increasing the lost motion lengthens the piston stroke and reducing the lost motion shortens it.

Compression in Fitchburg Engine

How can the compression at one end of the stroke of a Fitchburg engine be altered, and how are the exhaust valves attached to the valve stem?

C. F. E.

The degree of compression may be varied by giving more or less lap to the valve on the end of the cylinder where the change is desired. The valves are loosely fitted to the stem and are held in position by a nut. The crank-end valve abuts against a collar on the stem and its position may be altered by means of washers between the collar and the valve. The head-end valve is separated from the other by a distance piece and is also adjustable along the stem by means of washers.

Setting Safety Valve

If a safety valve is set to blow at 100 pounds pressure and it is desired to have it blow at 75 pounds, what is to be done?

S. V.

With spring-loaded safety valves a slight change in the release pressure may be made by a change in the spring tension. But a change from 100 to 75 pounds will necessitate changing to a spring suitable for the pressure. With a lever valve the weight may be moved nearer to the fulcrum.

Distance of Grates from Boiler

Can you give me any rule for finding the proper distance between the grate and the shell of a horizontal return-tubular boiler?

D. G. B.

The proper distance between the grate and the shell of a boiler depends upon the nature of the fuel burned, but it should in all cases be enough so that complete combustion can take place without the flames striking the comparatively cold boiler sheets.

With anthracite coal the ordinary return-tubular boiler setting is approximately correct; but where bituminous coal is used, the distance between the grate and the boiler should not be less than the diameter of the boiler.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Efficiency of Joints

On page 2002 of the issue of November 8, an inexcusable error was made in the reply to the question on the efficiency of a single-riveted lap seam. In giving the answer it was thoughtlessly assumed that the shearing strength of the rivets was equal to the strength of the plate between the rivet holes and the general formula applied. With a shearing strength of 42,000 pounds per square inch of rivet area the strength of the rivets would be

$$\frac{0.518 \times 42{,}000}{3.5 \times 0.375 \times 55{,}000} = 0.287$$

or 28.7 per cent. of the strength of the uncut plate.

Brass Polishing Paste

Will you please give me a recipe for a good brass polish?

B. P.

A very good brass polish is made by mixing rotten stone and soft soap or sweet oil into a thick paste. Apply with a woolen rag and rub vigorously.

Water Hammer

What is water hammer?

W. H.

Water hammer is the violent projection of a body or slug of water against the end of a pipe or fitting, or against the inside surface of a boiler or receiver. It is usually occasioned by the condensation of steam in some part of the system and the violent rush of steam and water from other parts of the system to fill the void caused by condensation.

Difference in Gage Readings

In an air-tight tank full of water under pressure with gages at the top and bottom, which will indicate the greater pressure, the top or the bottom one?

D. G. R.

The gage at the bottom of the tank will indicate a pressure of 0.432 pound more than the upper one for each foot of difference in hight.

Cause of Reduced Pump Speed

Why is it that after overhauling and repairing the boiler-feed pump, I fail to get as high a speed as formerly?

R. P. S.

If your pump runs slower than it did before overhauling, it is because it is working against more resistance. This may be due to either too tight a piston, or to the fact that the piston fits well enough so that some of the water does not leak past it but all goes to the boiler.

Slipping of Main Belt

If the main belt slips on one of the pulleys will the engine require more steam to do the work?

S. M. B.

If the resistance against which the driven shaft revolves is not increased, the engine will require less steam, because it is doing less work.

Decimal Reduced to a Common Fraction

How may a decimal be reduced to a common fraction?

D. C. F.

Write the decimal as the numerator of a fraction, the denominator of which is 1, followed by as many ciphers as there are digits in the decimal and reduce to the lowest terms. For example,

$$0.375 = \frac{375}{1000} = \frac{3}{8}$$

Smokestack Dimensions

What should be the diameter and hight of a smokestack for three horizontal tubular boilers 6x18 feet?

S. S. D.

From such meager data only a general answer can be given. If the stack has a cross-sectional area one-seventh that of the grate and a hight 25 times its diameter, it will approximate the dimensions established by good practice. The grate area for a 6x18-foot boiler will be nearly 33 square feet, say 98 square feet of grate surface for the three boilers. This would give 14 feet for the cross-sectional area of the stack. The diameter of a circle of 14 feet area is 4.2 feet and

$$4.2 \times 25 = 105 \textit{ feet,}$$

the hight of the stack.

Oil under Changes of Temperature

Is there any lubricating oil that will not grow thick in cold weather?

O. C. T.

So far as is known there is no oil that will not change in consistency under extreme ranges of temperature.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for October, 1910, *was* 132,000.

November 1............ 33,000
November 8............ 33,000
November 15............ 33,000
November 22............ 33,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power
and The Engineer

Editorial

Depreciation

Into the cost of running a power plant there enter a number of items which, like the salesman's new suit in his expense account, "are there but you can't see them." You can see the coal going in, and you know you have to pay for that, and for oil and waste and labor and repairs, and, perhaps, for water. The office knows, if the engineer does not, that taxes and insurance have to be paid upon the power plant, and that the greater the cost of the plant and the fewer the horsepower-hours gotten out of it in a year the greater will be the cost of taxes and insurance per horsepower-hour or year.

But there are a couple of other items which are not so well understood. The owner knows that he is paying interest on the cost of the power plant. Perhaps he had to borrow the money with which to build it, in which case the interest cost is as simple as taxes or insurance. If he took the money out of the bank or out of some investment where it was already earning interest for him, his purchase of the steam plant with it deprives him of that interest and puts him in the same position as if he let this remain invested and borrowed new capital at the same rate of interest with which to buy his plant, in which case there would be no question of the interest charge. When the desirability of the plant as an investment is considered, however, this interest which it is earning upon its cost must be taken into consideration. If the owner must borrow the money, the plant must earn six per cent. of its cost for somebody else before it commences to pay him anything. If it is built with his own money it must earn six per cent. before he is any better off than as though he had not built it, but had loaned his money to somebody else.

The most confusing item, however, is depreciation. One can see the coal being used and new coal coming in, and he can tell how much coal he uses per horsepower-hour, but he cannot see his engine being used up. He does not see new engines constantly coming in, and does not stop to think that this engine will last only a certain number of years, and produce in its lifetime a certain number of horsepower-hours, and that for every horsepower-hour which it produces he is using up so much engine, as well as so much coal and oil.

This applies not only to the engine, but to the boiler, the auxiliaries, the chimney, the building, and perhaps, even to the ground.

If a man buys a piece of ground for $5000 and puts a flour mill upon it, and after a while the mill burns down or is moved off and he sells the land for $3000, then every barrel of flour made in the mill must be charged with its quota of that $2000 which the use of the land during the life of the mill cost the owner.

This falling away in value is called "depreciation." In the case of land there may be "appreciation," or rise in value, but in the case of buildings and machinery ruin and the scrap heap are inevitable. It is only a question of time, and that time determines how much of the cost should be charged to the power plant each year.

If one pays $20,000 for an engine and it lasts twenty years, he is using up a thousand dollars' worth of engine a year, and must include this in the power cost or he will fool himself in regard to profit in the long run, although it may look rosy until the need of renewal begins to be apparent.

The way to meet this would appear to be to make the most intelligent estimate possible of the time that the plant will last. Many things may affect this. It may become obsolete while it is still in good condition; that is, there may be something so much better brought out that it would pay to scrap this and put in the more efficient installation. This is called "obsolescence." The plant may become outgrown and will, even if in good condition, bring but a fraction of what it cost installed and ready to run when offered second hand. It may live its life, serve its full time and go to the scrap heap. One must estimate as best he may how long the present outfit will last and when it may be expected to be "all in." But suppose one knew that the engine which cost him $20,000 today would last twenty years; how would he charge it into the yearly expenses? Shall he take out a thousand a year, charge it to the power plant and put in aside so that at the end of twenty years he will have the fund at hand to buy the new engine which will be required? If he does and puts the money out at interest, he will have too much at the end of the twenty years, for he will not only have the principal, enough in itself, but the compound interest on the successive sums. One thousand dollars put away at the end of each year, at six per cent. compound interest, would accumulate in twenty years to the sum of nearly $37,000, and to accumulate a sum of $20,000 would require an annual deposit of only about $540. This is called "amortization," or the killing off of a debt or charge.

The usual practice is to charge something like five per cent. of the investment per annum for depreciation when computing the over-all cost of horsepower production and kindred problems.

Are You a CO_2 Engineer?

Most engineers are interested in indicator diagrams. This is evident from the large number of cards mailed to POWER each month. The diagrams are usually from two classes of engineers: those who wish to know how to figure the horsepower indicated by the diagrams, and others who have learned how to take them and figure the horsepower, besides making proper valve adjustments. The latter diagrams are not accompanied by a request to tell what there is wrong about them, but are sent more in a spirit of "here they are, better them if you can."

Both kinds of diagrams show that the engineers are taking an interest in their work, and that they realize the importance of knowing how to indicate an engine, set the valve and figure the horsepower developed.

Strange as it may seem, no such interest is taken in the performance of the boiler furnace. It is seldom that a letter is received inquiring about the working of a CO_2 recorder or how better results may be obtained from certain furnace conditions. Engineers cannot be blind to the fact that the great saving to be made in the power plant in the future will be made in the boiler room. What, then, is the reason for this lack of interest?

The old method of throwing coal into the furnace in a haphazard manner is fast passing, and the idea that good combustion is taking place because the ashpit doors are wide open and the damper regulator operates is fast being dispelled. In fact, such methods of firing belong to the time when tallow was the lubricant for steam-engine cylinders and when hemp rope was used for piston packing. The man who holds to the old ways of doing things is fast becoming a back number.

Are you going to keep on in the same old way? Are you going to allow your firemen to throw in the coal and regulate the draft by guesswork, or do you intend to get on the track of progress and operate your boilers on a new plan?

Why not get as interested in the development of CO_2 gas as in indicating a steam engine? To operate a steam engine properly the diagram is indispensable, and the making of CO_2 is just as important for efficient steam-boiler service.

It would appear that a great many engineers do not realize the importance of flue-gas analysis, for if they did, they would have more to say about the subject. It can hardly be possible that there are very many who do not know about the important gains to be made by the intelligent firing of a boiler and by properly regulating the air and the coal supply. This fact is known to most engineers, but how do they propose to find out when their fires are being operated to the best advantage?

A CO_2 recorder will tell when fuel is being wasted; it will also tell the engineer when he is obtaining the greatest number of heat units from the fuel. It will act as a check on the fireman, and, if properly handled, will indicate just what performance is taking place in the furnace.

Publicity usually means the making or breaking of a new device and its method of operation. If the CO_2 recorder will do what is claimed for it, there must be some engineer who has had interesting experiences with the apparatus, who has found out that by analyzing the flue gases he has made a saving in coal consumption.

An account of such experiences would be of much interest to others and would doubtless assist some in getting started toward operating their boiler plants more economically than they have hitherto.

Myrine

With flaring headlines in type over an inch high, the *Catholic Union and Times* in a recent issue informs the waiting world that "the greatest invention since Edison and Tesla" has been made by Charles H. Myers, of Buffalo, N. Y., who is heralded as "the most wonderful inventor of the century." We do not happen to know how Edison and Tesla rank as inventions—in fact, we did not know they had been invented—they look like human animals of spontaneous growth. Neither do we know how much Mr. Myers paid the *Catholic Union and Times* for printing the foolish statements contained in the article; but whatever the price, it was far too much.

What Mr. Myers has done as the basis of the *Union and Times'* ebullition is to concoct a very ingenious liquid fuel for internal-combustion engines, heating and lighting purposes. Please note that we say "ingenious," not revolutionary or economical. The composition of his fuel Mr. Myers refuses to disclose. He calls it "Myrine" and claims that it is about twice as efficient as gasolene and costs about the same per gallon. The liquid is slightly oily in appearance, evaporates more slowly than gasolene and has an odor reminiscent of benzol. It is poured into what the inventor calls a generator, which appears to be merely a carbureter of the tank type, containing an absorbent material which holds the liquid much as a sponge would. The engine draws air through this generator, which evidently saturates the air with the liquid and makes a too-rich mixture, because an auxiliary air intake at the engine is used.

While "Myrine" is not a "fake" fuel, like Carroll's carbon dioxide, we are strongly of the opinion that its industrial merits are highly exaggerated, whether ignorantly or intentionally we do not pretend to say. The appearance and odor suggest a petroleum distillate somewhere between commercial gasolene and kerosene, but if the statements as to fuel consumption have any foundation whatever in fact, the liquid must have a much higher heat value than gasolene. This leads to the surmise that it may be either dilute picric acid or acetylene gas "dissolved" in acetone or some low-grade ether; if the heat value is not as high as is claimed, the liquid may be a mixture of benzene and denatured alcohol.

A feature of the enterprise that naturally engenders suspicion is that the liquid cannot be bought at any price until the "State rights" are all sold. Mr. Myers' representative told the writer that "Myrine" would be sold to the State companies at fifteen cents a gallon. If it has any such heat energy as is claimed for it, we do not believe that it can be made for anywhere near that price; there is no known volatile hydrocarbon, higher in heat value than gasolene, that can be marketed at such a price, and we decline to believe that Mr. Myers has discovered any new hydrocarbon.

We advise would-be investors to require reliable proof of the heat value in the fuel per dollar of manufacturing cost before putting any money in State rights. In gasolene, one can buy about seven thousand heat units for a cent; if Mr. Myers can sell the consumer a million heat units for a dollar and make money, his scheme is sound; if he cannot, he is simply deluding himself and the purchasers of his State rights.

In order to satisfy some critics who thought that the test of the Manning boiler made at the works of the Bigelow Company last July, and reported in our issue of July 26, was not entirely representative, in view of the fact that the openings for the handholes had not been cut out of the reinforcing plate, the test was repeated on October 25 with such openings cut out and the section of the boiler exactly in the condition in which it is regularly built. The pressure was run up to over 900 pounds, and the boiler failed by collapsing a tube, showing that the barrel, reinforced in the manner under discussion above the staybolts, is by no means the weakest part of the structure and is entirely adequate for the service for which it is built.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Surrell Double Draft Boiler

This boiler is designed to be used in connection with any return-tubular or other type of steam boiler. Its function is that of a steam generator and down-draft dutch-oven furnace.

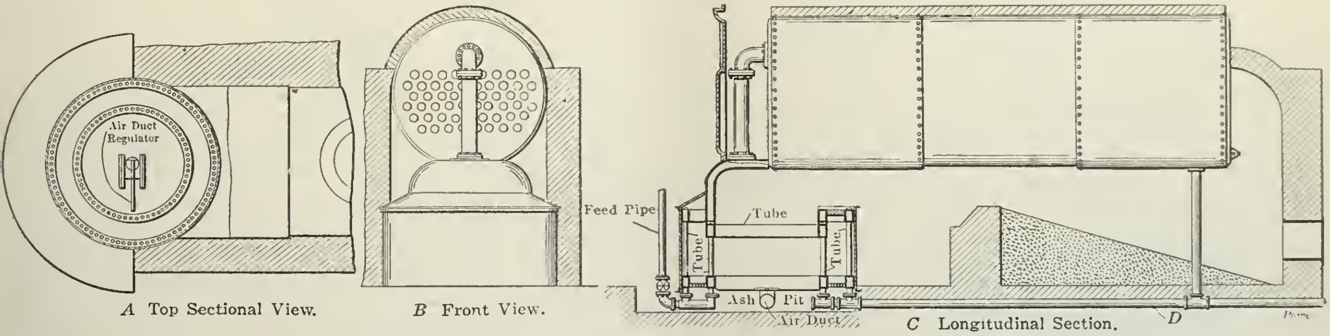
Section *A* shows a top sectional view of the boiler and furnace. The two rows of tubes shown connect the top and boiler header, the water circulating through the tubes. The furnace is placed between the two rows of tubes, and the draft is down through the fuel, the air entering through the top of the furnace and between the outside row of tubes. As the tubes in the inner row are distanced about one-half their diameter, a passageway is provided between them for the hot gases of combustion to pass to the inner combustion chamber. This chamber is connected by means of a flue to the space under the steam boiler usually occupied by the grate. The heat passes along the bottom of the boiler shell, and escapes to the stack after passing through the boiler tube.

The circulation of the water is through the pipe *D*, up through the two rows of tubes into the top header, shown in sections *B* and *C*. From the upper head, the water is carried to the top of the return-tubular boiler through a single pipe, which can be fitted to a tube opening or special flange hole, as desired.

Owing to the design of the Surrell boiler, it is claimed that the heat radiation is greatly reduced, and that its application to a steam boiler will greatly reduce the fuel consumption obtained under ordinary practice.

This boiler is made of cast iron and of steel for low and high pressures respectively. It is also made in complete units for use other than in connection with a high-pressure steam boiler.

The boiler is made in several designs by the Surrell Double Draft Boiler Company, 85 Liberty street, New York City.

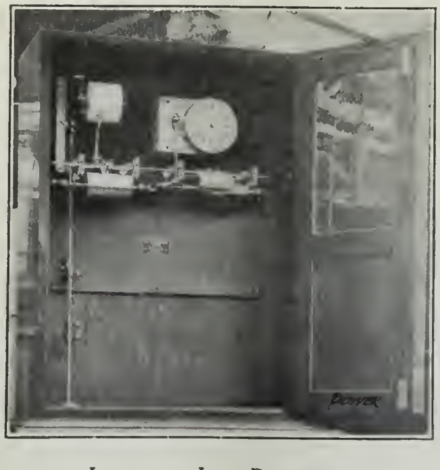


A Top Sectional View. *B* Front View. *C* Longitudinal Section.

THREE VIEWS OF THE SURRELL DOUBLE-DRAFT BOILER

Improved Lea Recorder

Following is a description of the improvements recently made in the Lea water recorder. In the case of the standard Lea recorder, the total weight of water passed is obtained from the area of the chart record by means of a planimeter or other means. While this is a very simple and accurate operation, and one which in many cases is quite sufficient, there are many instances where it is a great convenience, and in some cases an absolute necessity, for the total quantity of water passed to be read off directly and quickly. To meet this demand for an integrating or weir meter, an improved instrument has been devised.



IMPROVED LEA RECORDER

The accompanying illustration shows a general view of the device as now made. The driving clock is placed inside the casing. There are three counting dials which register in units, tenths, hundredths and thousandths.

The integrating action may be described as follows: The powerful eight-day clock is driven by a falling weight. This clock drives the vertical spindle, and this spindle, by means of bevel gears, drives the horizontal shaft at a constant speed.

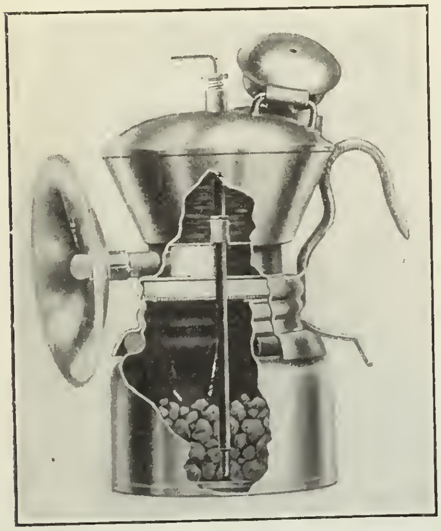
Upon the axle there is a toothed or corrugated drum the teeth of which are cut away in a regular spiral manner at the right-hand side. Above this toothed drum is a gear box, or case, containing the train of gears. This drum is driven by a pinion and engages intermittently with a few or greater number of teeth upon the toothed spiral drum, according to its lateral position. This position of the gear box and pinion is determined by the amount of movement of the long horizontal slider bar to the left from the zero position, the motion of this bar having been rectified and made directly proportional to the rate of water flow by means of the irregular spiral drum.

When no water is flowing over the weir and the instrument is at zero, the toothed spiral drum is out of contact with the driving pinion of the counting dials. When the rate of flow is, say, half the maximum, the driving pinion comes in contact with the toothed drum over half its circumference. When the rate of flow is at its maximum the pinion is driven continuously by the toothed drum, and so on for all intermediate rates of flow between zero and the maximum the counting pinions and the dials are driven in a proportionate manner, and thus, as the tooth drum revolves continuously at a regular speed, the total quantity of liquid passed is integrated or counted up in a very simple manner.

A written record of the flow is produced on the chart, daily or weekly, from which the total can be easily deducted from the area of the diagram just as in the case of an indicator diagram from an engine. The total quantity of liquid passed during any period can be read off directly whenever required, and as the total quantity passed can be arrived at in two different ways, namely, from the chart or the counting dials, there is a double check on all measurements made. This device is made by the Lea Recorder Company, 28 Deansgate, Manchester, Eng.

Baldwin Acetylene Lamp

The accompanying illustration shows a semi-sectional view of this lamp. It consists of but four parts—a carbide container, a water tank, burner and a wire agitator. This wire agitator is placed in a pipe which projects from the water tank into the carbide container, and there is just enough free space between the wire agitator and the sides of the tube to permit the proper amount of water to flow to the carbide which is placed in the lower receptacle. The water supply is shut off or turned on by the adjusting knob on top of the water tank.



Baldwin Acetylene Lamp

The illustration shows the general construction. The water is placed in the top chamber and flows through the central tube to the carbide. As the gas is generated it rises through the filtering bed shown between the outlet to the burner and the carbide chamber. The ignition of the gas escaping from the burner, shown in front of the reflector, produces a clear, white, brilliant light. The reflector is removable and the lamp is fitted with a hook and support braces for steadying purposes.

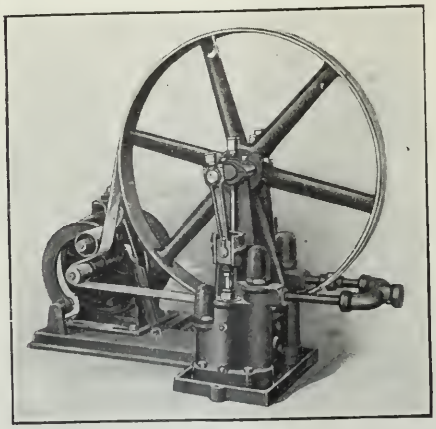
This lamp is suitable for various purposes and should be of great convenience to the engineer who is obliged to inspect a boiler either internally or externally. The flame will not blow out under ordinary conditions, there is no smoke, grease or oil, and when the lamp is held close to the point of inspection the reflector prevents the light from shining in the eyes of the person making the inspection.

Often the steam gage or water glass is located in a dark corner, but by placing this lamp at some convenient point the reflector can be made to throw light upon the gage, thus enabling the engineer to ascertain without doubt the steam pressure carried and the hight of the water in the glass.

The lamp is made of brass and can be carried in one's pocket. It will burn for a time if laid on its side or even held in an inverted position. It requires but one-half pound of carbide per day to operate the lamp, which will burn about two hours before it is necessary to recharge the reservoir with carbide. This lamp is manufactured by John Simmons Company, 102-110 Center street, New York City.

Sanitary Duplex Power Pump

These pumping outfits have been designed with special reference to service in office buildings, where noiseless operation is one of the prime considerations. They are also arranged to be operated in connection with pressure-tank systems, in which case they may be supplied with a small air pump in connection with the regular pumping equipment. The pumps are of the double-acting type, brass lined, and the pump body is built in one single casting, making the minimum number of joints to be packed or to require attention. The valves are readily accessible and can be reached by removing the air chambers. The piston rods are of brass and the stuffing boxes on these rods are the only joints requiring packing. This type of pump reduces the number of bearings to a minimum, dispenses with countershafts and other complications, and in connection with its noiseless operation is especially adapted to location where vibration must be eliminated.



Duplex Power Pump

The pump is made by the Sanitary Pump Company, Dayton, O.

The Bush Steam Generator

The accompanying illustrations show the improved Bush steam generator. It consists of two headers, as shown in Fig. 1, which are connected together by six boiler tubes, one end of which is threaded into the front header, the other end of each tube extending to the rear header or drum. This drum is connected to the blowoff outlet by means of a tee and flange connection, the lower outlet of the tee being connected to a mud drum on the blowoff pipe. The front header is connected to two holes made in the boiler head at a point above the water level, and are equal in capacity to that of the six tubes extending between the headers. The single tube between the blowoff connection and the drum has a capacity equal to the six pipes, shown in Fig. 1, connecting the headers.

The operation is as follows: The water in the boiler circulates down through the blowoff connection into the header through the large pipe, and passing through the six heating tubes enters the front header and passes into the boiler through the two vertical tubes.

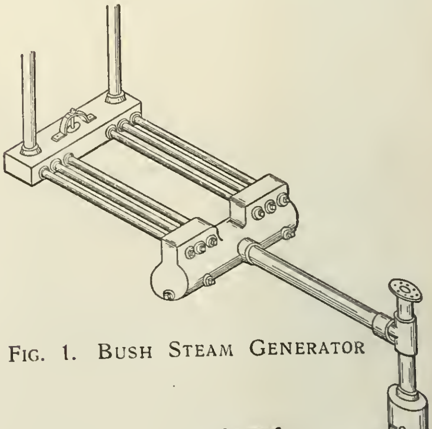


Fig. 1. Bush Steam Generator

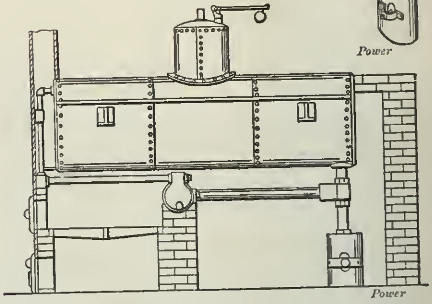


Fig. 2. Generator Connected to Boiler

This gives a rapid circulation and as the tendency of scale-making material is to settle, much of it finds its way to the mud drum shown at the bottom of the combustion chamber.

The front header is fitted with a handhole, while the drum is fitted with two plugged holes and a blowoff connection.

This device can be installed with any type of return-tubular boiler and is said to greatly improve the circulation and efficiency of the boiler. It is manufactured by the Bush Steam Generator Company, 61 Ann street, New York City.

New Building for Practical Mechanics at Purdue

Purdue University dedicated on November 12 a new building for the department of Practical Mechanics. The building, or rather a series of buildings, provides facilities for instruction in mechanical drawing, descriptive geometry and shop work. Ground was broken July 22, 1909, and the completed structure turned over to the university June 15, 1910. The main building contains 25,000 square feet of floor space; can accommodate at one time 400 students in drawing, and has locker accommodations for 1200 students. The lecture room seats 300 and there are two class rooms, each having a capacity of 60 students. The shops cover 43,000 square feet of ground and are capable of accommodating a group of 350 students at one time. The machines, tools, benches, lockers, in fact all the details of the equipment are modern while many of the special features are unique.

The American Institute of Steam Boiler Inspectors

This organization held its second meeting on October 26 at Boston, papers being presented by Professor Whiller, of the Massachusetts Institute of Technology, chief inspector Joseph McNeal, ex-inspector Thomas Hawley and George R. Richardson, of the Hartford Steam Boiler Inspection and Insurance Company. Abstracts will appear later in our reading columns.

The institute was organized last April. C. D. Noyes, of the Hartford company, is president; J. F. Mulloy, of the Mutual, vice-president, and E. R. Doherty, of the Mutual, secretary.

Lecture Course for Operating Engineers

The Modern Science Club, 125 South Elliott place, Brooklyn, N. Y., is taking a step in what appears to be the right direction. It is instituting a lecture course for operating engineers. The course consists of 20 lectures to be given every Friday night, commencing December 2, 1910, and extending to April 28 1911. One advantage of the course lies in the fact that the lectures will be given in a logical order and bear definite relations to each other. The following are some of the topics which will be treated: "Principles of Combustion," "Draft," "Flue-gas analysis," "Evaporation," "Boiler Design" (from the practical standpoint), "Boiler Rating and Efficiency," "Scale and Scale Treatment," "Steam-engine Indicator," "Planimeter," "Steam-engine Calculations," "Single-, Double- and Triple-expansion Engines," "Condensers," "Cooling Towers," etc. The course will be open to nonmembers.

NEW PUBLICATION

MECHANICAL ENGINEERS' POCKETBOOK. By William Kent. Published by John Wiley & Sons, New York. Morocco; 1500 pages, 4x6¾ inches; 218 illustrations; many tables. Price, $5.

This is the eighth edition of Mr. Kent's well known reference book; it contains 326 pages of text more than the preceding edition, but as 159 pages of the old edition were dropped entirely, this means that there are 485 pages of new material. The principal new material is in the sections devoted to Steam, Steam Engines and Turbines, Iron and Steel, Materials (other than iron and steel), Internal Combustion Engines and Water (including hydraulics). The Steam section contains an entirely new table of steam properties based on Marks' & Davis' values and tables on the flow of steam in pipes, condensed from Sickles' tables.

The section on Heat has been improved by substituting modern values of temperature-indicating colors of iron and steel for those published seventy-five years ago by Pouillet. The Marks & Davis definition of the British thermal unit is accepted here, which is justifiable on the score of convenience, but the compiler neglects to explain that the definition accepted by scientists is the quantity of heat required to elevate by one degree the temperature of a pound of water at its greatest density, corresponding to the definition of the French calorie. The specific heats of air are given to five places of decimals, although there are differences between various determinations of the specific heat at constant pressure and that at constant volume has never been determined authoritatively at all. Moreover, in the Air section the ratio of the specific heats is taken at 1.406 (Professor Wood's value), which would make the constant-volume specific heat 0.1689, whereas on page 435 it is given as 0.16847. The specific heat of olefiant gas ($C\ H_2$) at constant volume is about 50 per cent. too low, but as this member of the olefiant series is of no importance except to a chemist, the error will not affect many users of the book. It the compiler intended to discuss ethylene gas (C_2H_4), which is of engineering importance and has almost the same physical properties, it is a pity he did not get the symbols and specific heat right. There is a slight discrepancy between the Centigrade and Fahrenheit absolute zeros throughout the book; if the Centigrade zero be taken at — 273, then the Fahrenheit value must be — 459.4 and not — 459.2 as given. Since modern investigators have about agreed that — 459.6 is nearer the correct absolute zero of the Fahrenheit scale, and the calculated value for the gas thermometer is 459.7, and further since nobody knows what is right, it would seem justifiable to assume the round number of 460 for all engineering purposes instead of fussing over tenths of a degree. Whatever value is used, however, the Fahrenheit and Centigrade zeros should, of course, be consistent with each other. On page 565 the compiler says that Poole's formula for the mean temperature difference between the steam and the water in feed-water heaters "is based on an assumption that the transmission of heat [through the tube walls] is directly proportional to the temperature difference." As the rate of transmission does not enter into the formula quoted the statement is untrue. What Mr. Kent evidently had in mind was the complete formula for heat transmitted per square foot per degree difference, quoted on page 910 of this edition, but he does not say so.

The succeeding section on Thermodynamics is a new one. It contains a very skilful explanation of entropy, obviously written with the object of making the discussion easily intelligible, but the statement that "entropy is length on a diagram," etc., is certainly a somewhat extreme means of simplification. In defining pressure, one would scarcely go to the length of saying "pressure is hight on a diagram," etc. However, entropy is about the most difficult function, property or ratio (or whatever one regards it) to define or describe, and almost any method which clarifies its significance is justifiable. The arrangement of matter on pages 574 and 575 is confusing to the eye.

The Air section has been improved by giving several representative figures for the composition of air, but suffers a little from the persistent use of — 459.2 as the Fanrenheit absolute zero. The Murphy tables and formulas also add greatly to this section. In the discussion of air compression and expansion the formulas are said to apply to one pound of air, whereas they apply to any weight whatever.

The discussion of water power contains some valuable new material on ordinary types of turbine and on impulse wheels; and the succeeding section on pumps has been improved greatly by additional material on centrifugal and turbine pumps. The records of pumping engines, however, could have been enlarged profitably with a few more recent results.

The Fuel section contains a good deal of excellent new material, including, notably, abstracts of tests made by the United States Geological Survey and by Lord and Haas, and a subsection on producer gas. The coal analyses are more compactly and logically arranged than before, and Dulong's formula, as applied to coal, has been properly extended to include sulphur. The use of percentages instead of decimal parts of fuel constituents, however, is scarcely commendable because of the tendency to confusion. The heat of combustion of CO is incorrectly stated in the table on page 754; the heat there given is that of $2\frac{1}{3}$ pounds (the quantity formed by the incomplete combustion of one pound of carbon), not one pound. The heat per pound of CO, of course, is 4350 B.t.u.

The sections on Steam, Boilers and Engines have been very extensively revised, as previously intimated. Of course, the fundamentals remain the same, but questions of the practical application of those fundamentals have been pretty well brought up to date. The discussion of engine-parts dimensions might profitably have been shortened by omitting such data as have been superseded. The subsection on the slide valve is greatly improved; that on turbines is entirely rewritten and enlarged nearly five hundred per cent., but it could have been considerably enhanced in value by giving more details of the Stott-Pigott tests and some figures from the Bernon Mills installation, now about a year old.

The section on Internal Combustion Engines consists chiefly of abstracts and condensations from Poole's elementary book, "The Gas Engine," which the compiler forgot to mention, although he laboriously itemizes, as "references," a list of other textbooks to which he is not indebted for any considerable quantity of material. In the course of the section he gives off-hand credit by saying "Poole states," "according to Poole," and so on, but carefully refrains from mentioning the printed source of the data utilized. In scanning the Poole book for material Mr. Kent apparently overlooked probably the best practical formula in it, that is, the one for mean effective pressure, using r_c^{n-1} as the equivalent of K^{d} (pages 87 and 88). Referring to the Diesel engine the compiler includes amongst its distinguishing features the compression of air alone and the injection of oil by compressed air; the brief description of the De La Vergne engine on the same page shows that it does exactly the same things, which, therefore, cannot be "distinguishing" features of the Diesel engine. As to electric ignition systems the compiler says: "of these, the make-and-break and the jump-spark systems are in use." What others are there? The quotation from the *Iron Age* of seventeen years ago of producer-engine performance, while showing good economy, might have been omitted and the space devoted to something more modern; seventeen years is almost a gas-engine century. Other excellent test figures, of recent date, are given, making the classic antique still less worthy of the space accorded it.

Although the section on Gearing has been enlarged by five pages, the data and information relating to friction clutches and other frictional drives are disappointingly meager. Some very useful material on worm gearing has been added.

The section on Friction and Lubrication has been improved considerably by the addition of material on ball and roller bearings and on tests of lubricants.

Ice-making Machines has been extended fourteen pages, which include new tables of ammonia properties; the Machine Shop section has grown about twenty pages, and the Foundry section has six pages of new material and five of old.

Electrical Engineering has also grown; although eleven pages of old material have been dropped entirely, the section is seventeen pages longer than before. It has been improved by the elimination of the archaic method of calculating dynamo proportions and windings which spoiled this section of the old edition, but suffers a little from the compiler's efforts to interpret some fundamental propositions and definitions. For example, the definition of the C. G. S. unit of current, at the bottom of page 1344; some of the "modifications" in the table on page 1347; the definition of resistivity on page 1349; the use of "conductivity" where "conductance" is meant, on page 1351; the explanation of series circuits and joint resistance. There is considerable repetition in the section and the method of calculating field-magnet excitation here given has been obsolete at least fifteen years. The discussions of storage batteries and metallic-filament incandescent lamps are inadequate. The exposition of the principles of the magnetic circuit is excellent as to quality, but the compiler manifests here again a disposition to give as little credit as possible for "lifted" material; the Sheldon table on page 1384 is taken bodily from Foster's Pocketbook, but there is no reference to that fact. The subsection on alternating currents remains practically unchanged and is very satisfactory. A subsection on Electric Motors has been added, and that is also excellent, though somewhat prolix for a book of this character.

All of the tables in the book have been set in a particularly legible and durable style of type and the mechanical work throughout is unusually good. It is scarcely necessary to say that the book is practically invaluable to anyone having to deal with mechanical engineering and closely allied branches of work.

In making the adverse comments in the foregoing review the enormous difficulties in collecting, arranging and revising data for a work of this sort have not been ignored. The purpose and intention of the reviewer were to point out only the more important defects and those that might have been avoided easily, not to carp over minor lapses. The merits of the book immeasurably outweigh its defects.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies Building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K Morse; sec., E. K. Hiles, Oliver Building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres. William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore Md.; sec. and treas., D. L. Gaskill, Greenville, O.

power
and The Engineer

Vol. 32 NEW YORK, NOVEMBER 29, 1910 No. 48

THE ability to reveal our thoughts and emotions to others by word of mouth is given to most of us. It is comparatively easy to communicate with one another in this way because the voice can be so modified and inflected, accentuation and tone can be so varied as to impart the most subtile shades of meaning. In addition to these vocal variables there are the stage settings—so to speak—facial expression and gesticulation. And yet, in spite of all these, how often are our remarks misunderstood! Who has not had the sad experience of unwittingly offending or hurting the feelings of some good friend by a well meant joke or bit of persiflage.

Think then, what a difficult thing it must be to make a written statement perfectly clear. Few who have not been put to the test realize how hard a job it is to write exactly what is intended so that the reader will get what is meant. Simple thoughts or ideas, of course, do not offer much difficulty. But, let the subject matter be extensive enough to require several consecutive sentences, and the trouble will begin.

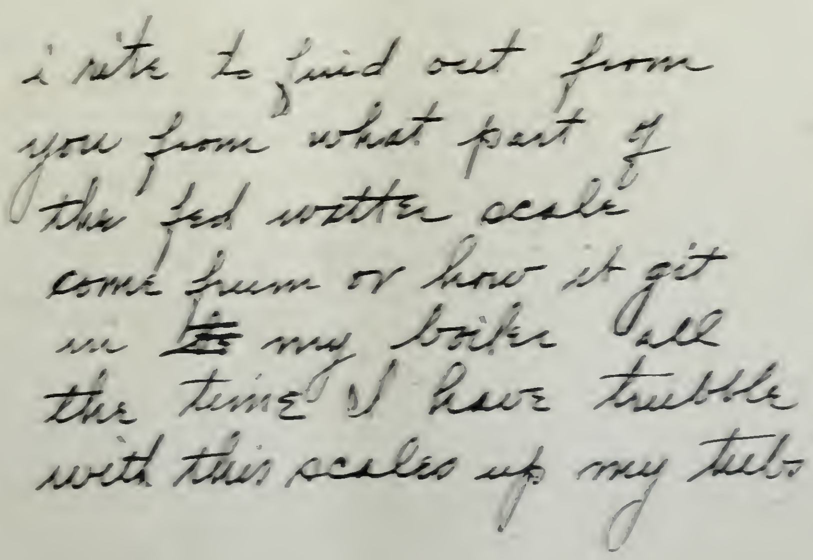
These very paragraphs will be misconstrued by hundreds. It probably is no exaggeration to say that each one who reads them will find a meaning at least slightly different from that found by anyone else.

Are all agreed, then, that it is hard to write correctly?

* * * * *

There is more credit and benefit in doing with a fair degree of success something which is difficult, and improving the performance each time, than there is in doing something which is easy and requires little effort. Therefore, when you get an idea—an idea which you want to give to others or an idea that you want help—put it into writing, the best writing you can muster, and send it to us. The practice will be good, for it is practice that makes perfect. If you do not come off with first "honors," what of it; you have had the practice.

The writing shown herewith is fictitious but is a fair sample of letters occasionally received at this office. The writer at least conveys his meaning and shows that he is interested in his work. Therefore we are always glad to receive such a letter.



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the time I have trubble
with this scales up my tubs

When a man gets busy with the pencil, no matter how poor a speller or grammarian he may be, hope is not lacking. There must come a time when he will be better at spelling and at a good many other things.

The first thing is to get busy; the next, to keep busy.

Compound Engine Proportions

By Alwin Hofmann

> *By means of the diagrams any one of the factors—initial pressure, back pressure, cylinder ratio, cutoff and mean effective pressure referred to the low-pressure cylinder—may be determined if the others are given. This refers to both compound condensing and noncondensing engines.*

For a number of years the writer has been using the accompanying diagrams in connection with calculations relating to compound engines, and as they have been found of considerable help they are given herewith with the thought that they may prove useful to engineers and others who are engaged in this kind of work.

It is now common practice to proportion compound engines by figuring first the low-pressure cylinder, as if all the work were done in that cylinder. After this has been done, the high-pressure cylinder is calculated in accordance with the desired cylinder ratio or the kind of work which the engine is called upon to perform, as will be explained later.

On the left-hand vertical scale of the diagrams, of which one is for condensing and the other for noncondensing engines, the cylinder ratio is given, and on the right-hand vertical scale the mean effective pressure referred to the low-pressure cylinder; that is, the mean effective pressure which would be obtained in that cylinder if all the work were done in it. The curves plotted across the diagrams represent cutoffs in the high-pressure cylinder, and the straight lines represent different gage pressures at the throttle.

The diagrams can be used for a number of different purposes, which can best be explained by citing some practical examples of their use. First, consider the case of an engine where it is desired to determine the mean effective pressure that can be developed with a known steam pressure. Suppose this is a condensing engine carrying 26 inches vacuum, 150 pounds gage steam pressure at the throttle with a cylinder ratio of 4 to 1 and cutoff at one-quarter stroke. Looking on the left-hand scale of the diagram, Fig. 1, opposite the cylinder ratio of 4 to 1, follow the horizontal line to its intersection with the one-quarter cutoff curve. From this point proceed vertically as far as the steam-pressure line of 150 pounds, then horizontally to the right-hand scale where will be read a mean effective pressure of 28 pounds referred to the low-pressure cylinder. The pressure lines are marked for gage, not absolute pressures.

The chart for noncondensing engines is used in the same manner. The mean effective pressures represent the actual figures which one may expect to obtain from an engine with Corliss or similar type of valve gear running at a piston speed of 600 to 800 feet per minute, allowance having been made for cylinder clearance, loss of pressure through ports, etc.

The vacuum is taken as 26 inches at the exhaust flange of the engine, but for each inch increase or decrease in vacuum, 0.38 of a pound can be added to or subtracted from the mean effective pressure obtained.

The engines are assumed to have neither reheating receivers nor steam-cylinder jackets. Where a reheating receiver is used, 4 per cent. can safely be added to the mean effective pressures obtained. Where both reheater and steam jackets are used, 10 per cent. should be added to the mean effective pressures.

For piston speeds of less than 600 to 800 feet per minute, the mean effective pressure will increase slightly. For high-speed engines or similar types the mean effective pressure may be reduced by about 5 per cent.

The noncondensing diagram is figured on the basis of the low-pressure cylinder exhausting freely into the atmosphere at an altitude of approximately sea level. Where an engine operates against a back pressure, the mean effective pressures should be increased about 0.85 pound for each pound of back pressure. At altitudes above sea level, 0.4 pound per 1000 feet altitude should be added.

The numbers given on the diagrams along the steam-pressure lines are the absolute pressures which will be obtained at the end of expansion in the low-pressure cylinder for any mean effective pressure. Attention is called to the fact that this terminal pressure is not dependent upon the cylinder ratio nor the cutoff, but is determined solely by the steam pressure and ratio of expansion, or the mean effective pressure referred to the low-pressure cylinder. The terminal pressures are especially useful in proportioning noncondensing engines as will be explained later.

The second use of the diagrams is that of determining the cylinder sizes of a compound engine, and this can also be best explained by giving a few examples. In proportioning the cylinders, the first thing to do is to figure the size of the low-pressure cylinder—the cylinder ratio is an after consideration—the important thing being to figure the low-pressure cylinder so as to have the proper mean effective pressure referred to that cylinder in accordance with the results desired. Too much stress cannot be laid upon this point, because many engineers are inclined to overlook this.

The cylinder ratio alone is no indication of economy of an engine. A high cylinder ratio may or may not mean an economical engine, and under certain conditions a high ratio may even be a detriment and reduce the economy rather than help it. For instance, consider a condensing engine designed so as to require a comparatively high mean effective pressure referred to the low-pressure cylinder. If a high-pressure cylinder which will give a high cylinder ratio is used here, it will not help the economy because the high-pressure cylinder will then operate with a late cutoff and therefore the port friction will be excessive; in other words, the high-pressure cylinder is too small and the ports therefore not large enough to properly handle the amount of steam needed to do the work.

No definite rule can be laid down giving the proper mean effective pressure upon which to figure the low-pressure cylinder, as this depends upon the results desired. For instance, in the case of a condensing engine, if a highly economical engine is wanted a high ratio of expansion should be used or what amounts to the same thing, the mean effective pressure referred to the low-pressure cylinder should be low. This does not mean that the lower the mean effective pressure the higher the economy, because a point is soon reached where increasing the ratio of expansion or reducing the mean effective pressure adds nothing to the economy on account of the increased amount of cylinder condensation encountered when using a high ratio of expansion.

A high ratio of expansion also means a large low-pressure steam cylinder, and therefore high first cost. On this account, it is the engineer's business to so choose the mean effective pressure as to make the engine commercial, or to design it in accordance with the degree of economy desired. Consider a condensing engine operating at 150 pounds steam pressure and 26 inches vacuum. If a highly economical engine is desired without regard to first cost, it would be good practice to use a mean effective pressure of about 25 pounds, or a ratio of expansion of, approximately, 19. Increasing the ratio of expansion above this would not improve the economy appreciably, therefore it would be bad practice to go beyond this. After deciding upon the mean effective pressure and knowing the horsepower required, the stroke and the speed, it is a simple matter to determine the diameter of the low-pressure cylinder from the following formula:

$$I.h.p. = \frac{D^2 \times N \times S \times P}{252{,}000}$$

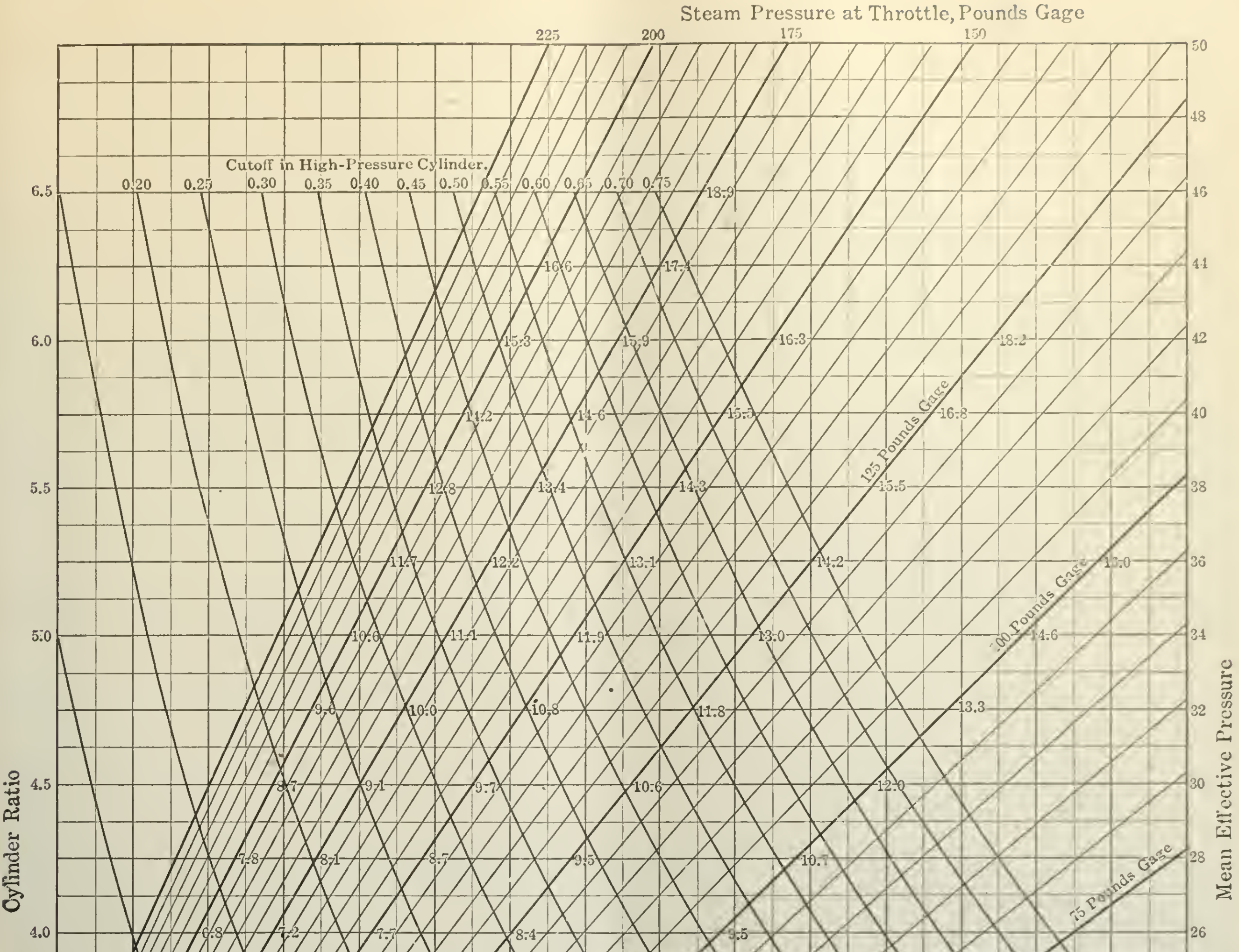


FIG. 1. MEAN EFFECTIVE PRESSURE CURVES FOR CONDENSING ENGINE

where,

I. h. p. = Indicated horsepower;

D = Diameter of the low-pressure cylinder in inches;

N = Revolutions per minute;

S = Stroke in inches;

P = Mean effective pressure in pounds, referred to low-pressure cylinder.

In determining the size of the high-pressure cylinder, the cutoff in that cylinder is first decided upon. This should not occur too early in the stroke, as in that case the cylinder condensation will be excessive; and it should not occur too late in the stroke or the pressure losses through the ports will be too great. In the case in hand, where economy is the first consideration, a cutoff of between 25 and 30 per cent. will give good results. Referring to the diagram, Fig. 1,

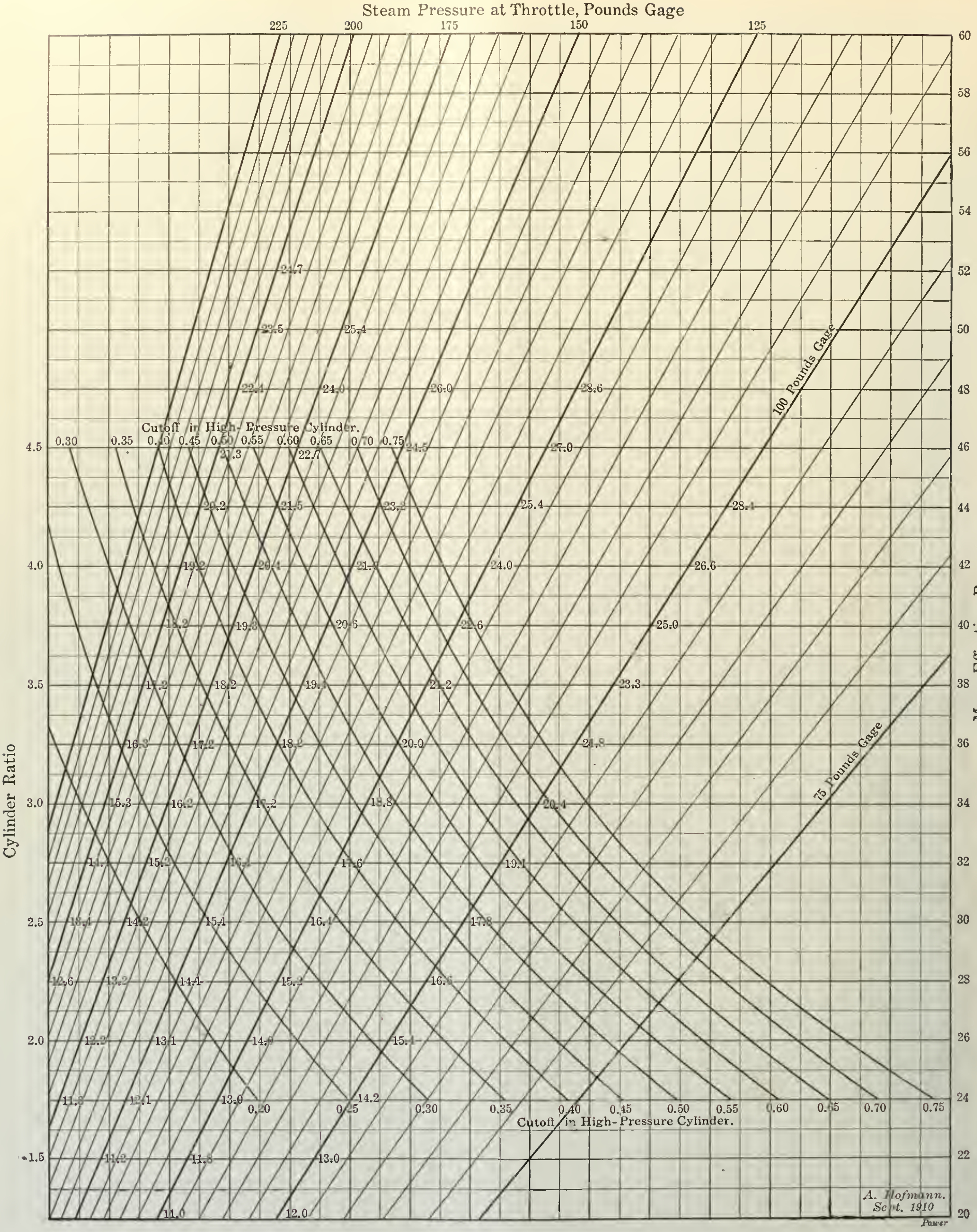


FIG. 2. MEAN EFFECTIVE PRESSURE CURVES FOR NONCONDENSING ENGINES

it will be found that with 25 pounds mean effective pressure, 150 pounds steam pressure and, say, 27½ per cent. cutoff, the cylinder ratio will be 5 to 1. Having determined the cylinder ratio, it is a simple matter to figure the diameter of the high-pressure cylinder, the size of the low-pressure cylinder having already been found.

In the case of the foregoing example, it has been assumed that the engine is to operate with a comparatively constant load. On the other hand, if the load is variable a higher mean effective pressure should be used in order that when the engine is operating under a light load the condensation losses will not be excessive, also with a variable load a comparatively low cylinder ratio should be used so as to give an early cutoff at normal load, thus giving the engine a large overload capacity.

If, instead of a highly economical engine a commercial engine is desired, that is, one in which economy may be sacrificed in favor of price, the low-pressure cylinder is proportioned by using a higher mean effective pressure. This requires a smaller low-pressure cylinder. In the case of the example cited, a mean effective pressure of 28, or even 30 pounds, could be used, and a cylinder ratio of 4 to 1, which would give a good engine but still not one of maximum economy.

In figuring the cylinders of a noncondensing engine, the operation proceeds along somewhat different lines. The main object here is to have the steam expand down close to atmospheric pressure in the low-pressure cylinder, for when this condition exists maximum economy is obtained. Therefore, in determining the proper mean effective pressure in the case of a compound noncondensing engine, it is necessary only to find the terminal pressure given along the steam-pressure lines in the diagram, and then choose the mean effective pressure so as to give a terminal pressure of about 15 pounds or a little higher, in which case the steam will expand down practically to atmospheric pressure; that is, there will neither be a loop in the low-pressure indicator card nor excessive drop, neither of which is desirable in a noncondensing engine. The size of the high-pressure cylinder can be determined the same as in the case of condensing engines, except that with a noncondensing engine a later cutoff must be used in the high-pressure cylinder; otherwise that cylinder will be too large and most of the work will be done in the high-pressure cylinder instead of being divided equally between the two cylinders.

Another use of the diagrams which is evident, but which attention is specially called to here, is the following: Assume an existing engine with a cylinder ratio of 3.5 to 1, operating noncondensing, the indicated horsepower to be developed being such as to require a mean effective pressure of 28 pounds, referred to the low-pressure cylinder. It is desired to know if this engine will operate at 125 pounds steam pressure, provided it has a cutoff gear which will permit only a cutoff at 40 per cent. of the stroke and no later. Referring to the diagram, Fig. 2, it will be found that under these conditions the cutoff in the high-pressure cylinder will be just a little short of 40 per cent. and therefore the engine will operate at 125 pounds steam pressure.

Milwaukee Garbage Destructor

Disposing of the garbage, ashes and refuse matter collected from cities is a problem that has cost thousands of dollars with little or no commercial returns to show for the outlay. Various methods have been employed in disposing of these materials, but have proved unsatisfactory, due to the high cost of removal or the failure of the destructor apparatus to give satisfactory results. Milwaukee has probably had the most varied experience in the disposal of its garbage of any city in this country. Almost every known method of garbage disposal has been tried at various times during the last twenty years. Prior to 1882 different methods were used in different wards, the people either burying the garbage, feeding it to poultry, hogs and cattle or dumping it on vacant lots.

These methods were abolished in 1882, when a municipal collection system was installed and an experimental destructor was built which was unsuccessful. The garbage was then dumped in adjoining towns and later in the lake. Then another destructor was built and an attempt was made to use oil and coal to burn the garbage. This plant was operated for six years, when another type of reduction plant was built and operated until closed by the health commissioner in 1890. For two years the garbage of the city was dumped into the lake and then discontinued as it was found to contaminate the water supply.

Then for five years the garbage was removed by water transportation, at the end of which period a system of municipal collection and burying farm was again installed. In 1901 another crematory was built to use coal for fuel, which was operated on and off up to May 12 of this year.

This destructor plant is one of the latest to be put in operation in this country. The garbage is incinerated without the aid of fuel other than that found in the refuse matter. The heat from the destructor is used to generate steam in four 200-horsepower boilers supplying the engine equipment of the plant. Excess steam is allowed to go to waste.

It has been found in England, Germany, Belgium and France that all domestic refuse when mixed will burn without additional fuel. Due to this experience and influenced by a report on the disposal of garbage by Dr. Rudolph Hering, the city of Milwaukee advertised for bids under specifications which besides the usual guarantee of capacity, freedom from dust, smoke or other nuisance, required guarantees of the amount of labor and the quantity of useful steam to be delivered by the plant. A contract was awarded The Destructor Company of New York City for the construction of a Heenan refuse destructor of 300 tons daily capacity, and the plant has been operated by the Board of Public Works since June 17, 1910.

The plant consists of four independent units, each of 75 tons daily capacity, arranged in the four corners of a rectangular two-story and basement building.

Fig. 1 is an exterior view of the plant from the lake side. Fig. 3 is a cross-sectional view through the building showing the general arrangement of the furnaces. Fig. 2 is a view of the clinkering floor where the operation of the plant is carried on. The superintendent's office, electrical machinery and recording instruments are located on this floor. The upper story is used entirely as a feeding floor and for storage of refuse. see Fig. 4, which is hoisted into the building and distributed into the various storage bins by a pair of overhead electrically operated traveling cranes. The basement is used for a clinker railway and doors for cleaning the dust from the furnace.

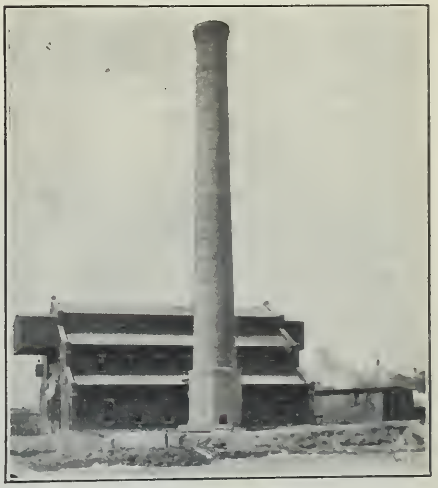


FIG. 1. VIEW OF PLANT FROM THE LAKE SIDE

The refuse is collected in two kinds

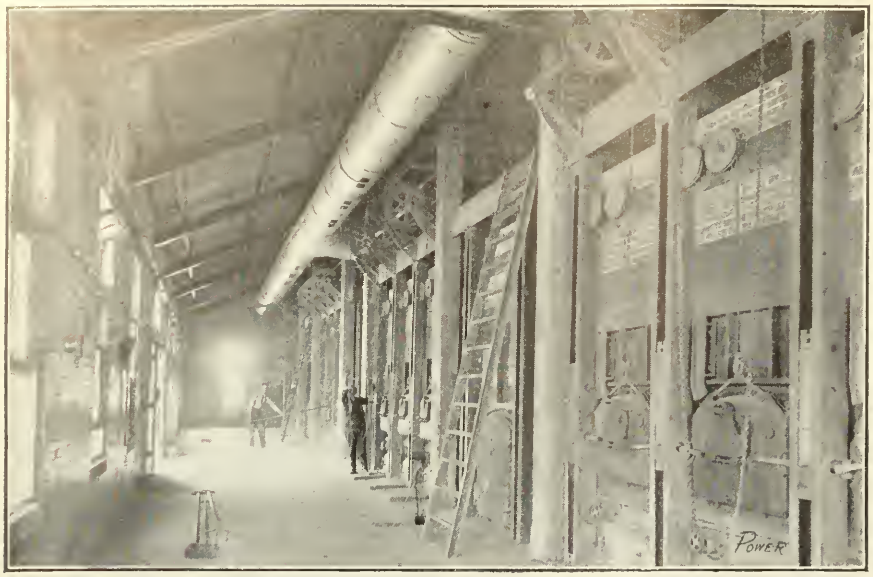


Fig. 2. View of Clinkering Floor

of carts. One type has a removable steel body of about 1½ cubic yards capacity and is used chiefly for collecting garbage. The body is hoisted by a crane and the contents discharged into bins on the feeding floor. The other type of cart is bottom dumping, of about 2½ cubic yards capacity. These carts are used chiefly for the collection of rubbish and ashes and dump into boxes resting in pits at the ground level; the boxes are picked up by the cranes and their contents transferred to the storage bins. From the storage bins, the men who feed the furnaces rake or shovel the material into the mouths of rectangular steel chutes termed "containers." These containers are located directly over the top-feed openings into the furnaces, the sliding doors to the furnace openings also serving to keep the bottoms of the containers closed.

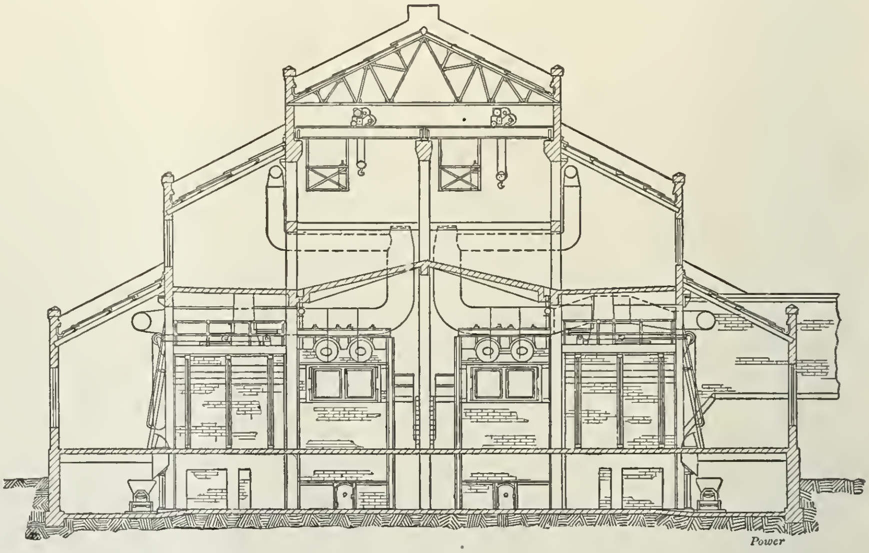


Fig. 3. Cross-sectional View through the Building, Showing General Arrangement

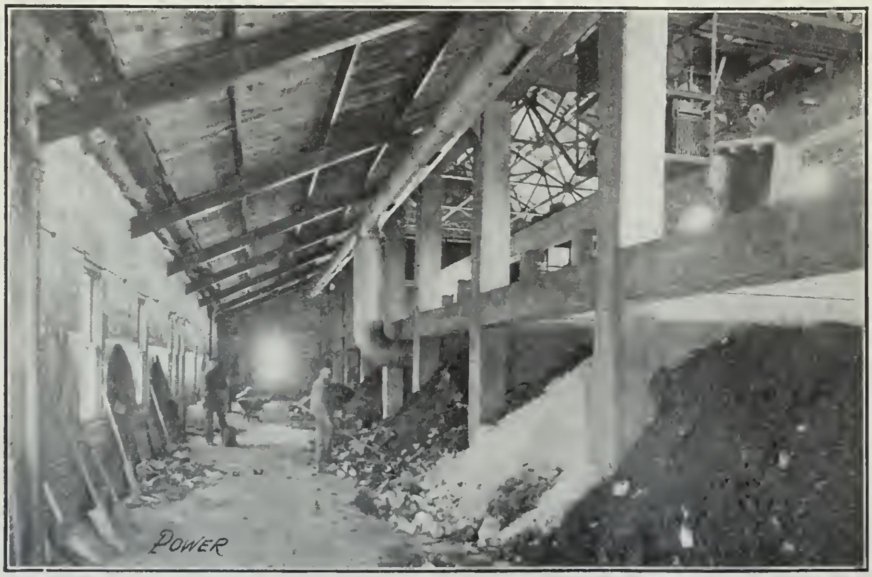


Fig. 4. Feeding Floor

The man feeding from above keeps the containers charged with mixed material at all times and the doors admitting the contents into the furnaces are controlled by the fireman from the floor below. The material when dropped into the furnace by the operation of the sliding doors, falls on a drying hearth where a large proportion of the moisture is carried away with the hot gases. The fireman rakes the dried material forward on the grates as required, for which purpose small stoking doors are provided.

After a sufficient amount of material has been burned on each grate to form a thick body of clinker, the large counterweighted clinkering doors in the front are opened and the clinker, after being broken up by bars, is drawn out of the furnace and dropped through holes in the floor into steel dumping cars on tracks below, whence it is pushed outside the building and allowed to cool on the ground. Fig. 5 is a half-plan view of the furnace and clinkering floor.

Each furnace consists of six separate grates located three on each side of a central combustion chamber through which the products of combustion of all six grates pass and are thoroughly mixed. The velocity through this combustion chamber is so slow that the combustion is entirely completed and dust is allowed to settle. After leaving the combustion chamber, the gases from each unit pass over the heating surface of a 200-horsepower water-tube boiler, and then through an air heater or regenerator on their way to the chimney.

The combustion is entirely controlled by forced draft, all the air being drawn from ventilating ducts in the building by an engine-driven fan, and raised to a temperature of over 300 degrees in the air heater. Fig. 6 shows the path of the gases and air supply from the ventilating ducts. The proportions are such that the amount of air required for combustion when the plant is operating at full capacity is seven times the cubic contents of the building, which insures the renewal of all the air in the building about every eight minutes, and prevents the escape of any gases from the building except through the chimney. Separate ashpits are provided under each grate with controlling valves, so that the rate of combustion and the amount of air supplied may be varied under the individual grates at will. Each unit is fitted with its own fan and engine with independent flues.

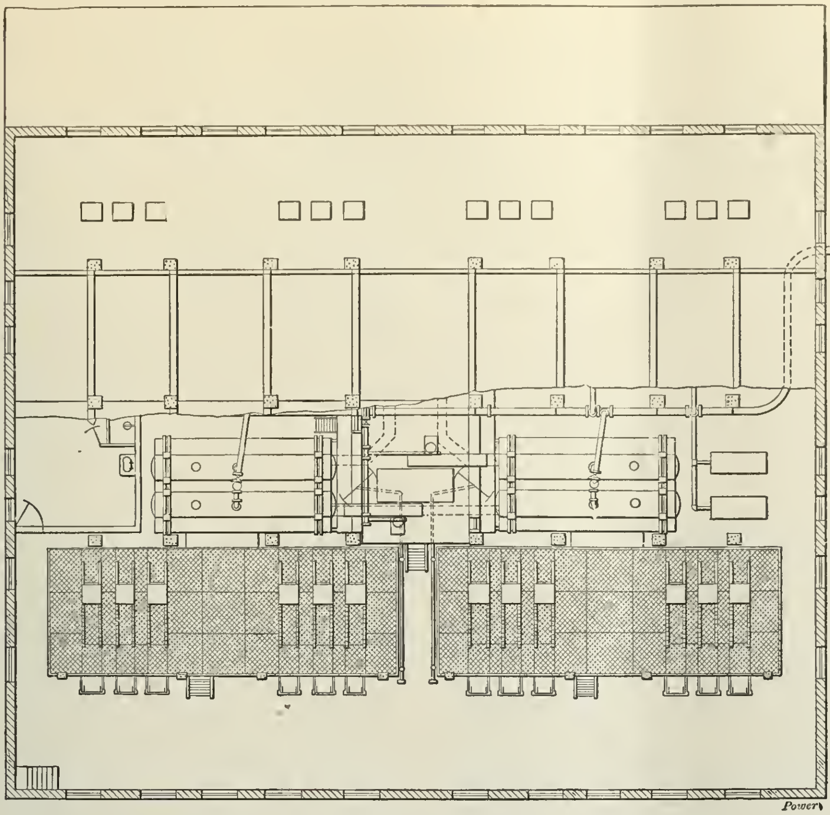


Fig. 5. Plan of Half of Furnace and Clinkering Floor

RESULTS OF TESTS MADE ON PLANT.

Garbage, percentage	41
Ashes, percentage	41
Rubbish, percentage	5
Manure, percentage	13
Dates of Tests: May 26, 27, 30, 31, June 1, **1910.**	
Quantity of refuse consumed in 37 hours, tons	126.81
Contract requirement, in 40 hours, tons	125.00
Excess over guaranteed capacity, per cent	9.66
Rate of burning per sq.ft. area, lb	64.00
Steam evaporated from and at 212 degrees Fahrenheit per pound refuse destroyed, lb	1.34
Guaranteed evaporation per pound refuse destroyed, lb	1.1
Excess over guaranteed evaporation: lb.	0.24
percentage	21.8
Temperature of combustion chamber, average, degrees Fahrenheit	1664
minimum, degrees Fahrenheit	1267
Temperature of air supply, degrees Fahrenheit	351
Cost of Operation:	
Labor per unit of 75 tons rated capacity per 24 hr:	
One feeder, 37 hr. at 25c	$ 9.25
Three firemen, 37 hr. at 25c	27.75
¼ engineer's time, 37 hr., 37½c	3.47
Total cost of labor	$40.47
Cost of labor per ton	0.319
Steam required to operate fan engine and feed pump, per hr. at 130 pounds gage pressure, lb	694
Equivalent from and at 212 degrees per hr., lb	843
Steam consumption per ton, lb	19
Value of steam consumed per ton of refuse incinerated at contract price of 4 cents per 100 lb	$0.088
Labor	0.319
Total gross cost per ton	$0.407
Revenue for steam per ton of refuse incinerated at contract price of 4 cents per 100 lb. for 2680 lb. of steam	$1.072
Net profit per ton	$0.665

At present garbage is taken from the entire city and refuse and ashes from but seven out of 24 wards. This makes the percentage of garbage to be disposed of in the destructor exceedingly high on account of lack of combustible matter. The average annual mixture collected has the composition given in the table.

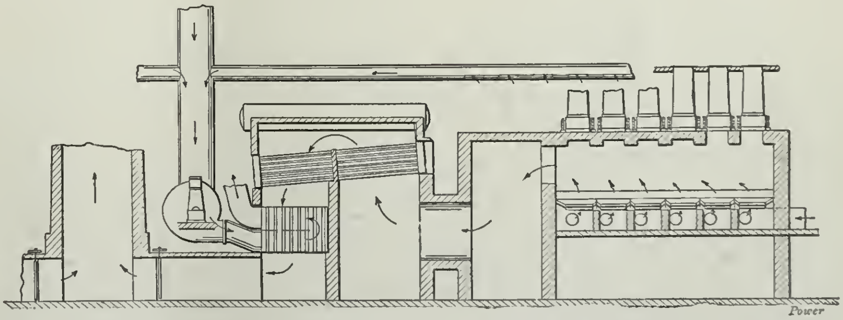


Fig. 6. Showing Path of Gases and Air Supply from Ventilating Ducts.

The tests show that the cost of steam obtained from incinerating one ton of refuse at a contract price of four cents per 100 pounds for 2680 pounds of steam is $1.072, but as a matter of fact the steam made by the boilers in the plant is not used for any commercial purpose other than operating the engine driving the necessary machinery throughout the plant. The excess steam is allowed to go to waste through a reducing valve into a sump filled with rocks to deaden the sound.

It costs the city of Milwaukee $25,000 a year to operate two flushing pumping plants and one sewerage pumping station. It is stated on good authority that the steam allowed to go to waste from the destructor boiler into the sump would be sufficient to operate these three pumping stations, providing the necessary equipment were procured to utilize this power, which would not exceed an estimated amount of from $75,000 to $100,000.

A similar destructor plant to be located on the opposite side of the city has been advocated. This would not only double the possible power output from the disposal of garbage, but would decrease to an appreciable extent the present cost of cartage.

At a recent meeting of the Institution of Municipal and County Engineers in Dublin, says *The Surveyor*, a communication of considerable value by M. Sellars, the town surveyor of Dundalk, on the scraping of water mains was submitted for consideration. In cases where the action is corrosive, of course, the material of the pipe must be adversely affected; while in cases of mere incrustation or a surface deposit there may at the same time be a damaging influence at work on the pipe. As Mr. Sellars points out, the principal cause of reduction of delivery from a pipe which has become incrustated or corroded is not to be found in the reduced bore, but in the roughness of surface which is a necessary accompaniment of a pipe so affected. This roughness produces sudden eddies and cross currents which, even apart from the great amount of head which is lost in friction, tends to an unduly large diminution of velocity. This, then, is the evil, and the remedy points toward efficient pipe scraping, which will thoroughly remove this incrustation.

Factors Affecting Air Pump Capacity

By R. M. Neilson

Although it is common practice to determine the capacity of air pumps by arbitrary rules, such methods should always be checked by calculations based upon the behavior of a mixture of air and vapor under the conditions existing in the condenser.

Air pumps for handling the exhaust from steam engines are often designed by rule-of-thumb methods. There is some excuse for this practice because it is often difficult to ascertain the probable volume of air and vapor which the pump will be required to handle per minute; but with large units the cost of the air pump is such a significant factor that it is worth while to ascertain the minimum size of pump which will suffice. As the function of the air pump is to withdraw the air from the condenser, it would seem that for any vacuum the capacity of the air pump—the effective piston displacement per minute—should be proportional to the air which enters the condenser per minute. The air pump, however, necessarily withdraws steam or water vapor along with the air, and this fact must be taken into account.

Air in a condenser is always associated with steam or water. It is well known that there is a definite temperature corresponding to any pressure of saturated steam. The steam cannot be reduced in temperature without reducing the pressure, and cannot be increased in temperature—unless by being superheated—without being accompanied by a rise in pressure. However, if the steam is in intimate association with water it cannot be superheated.

A fact of equal importance to steam engineers, but less widely known, is that for any mixture of air and steam in definite proportions there is a definite temperature for every pressure. The proportion of the mixture, that is, the ratio by weight of air and steam, cannot be altered by condensing some of the steam unless either the temperature or the pressure of the mixture is altered. To explain the reason of this would involve a dissertation upon the laws of partial pressures of mixed gases, which is somewhat beyond the scope of this article.

It is important to note that the higher the temperature for any given pressure, the greater is the proportion of steam to air, the limit being reached when the temperature is that of the boiling point of water at the given pressure, at which point the ratio of steam to air is infinite. Lower temperatures than that corresponding to any given pressure represent a reduced ratio of steam to air; hence it is desirable to reduce the temperature of the mixture which is to be discharged by the air pump so as to reduce the size of the latter and the power required to drive it. Manufacturers and users of condensers in general are aware of the advantage of reducing the temperature of the mixture; and condensers, both of the injection and the surface type, are now usually arranged on the contraflow principle, so that the mixture of air and vapor, just before leaving the condenser, is cooled by the coldest entering water. In addition to this the air is often passed through a special cooler placed between the condenser and the air pump.

It is often stated that the reason for seeking to thus reduce the temperature of the mixture is to reduce the volume of the air, because the volume of a gas is proportional to its absolute temperature. The real reason, however, is that a reduction in temperature reduces the amount of steam or water vapor associated with the air. As steam, this occupies a large volume, but when condensed into water, its volume is almost negligible. The mere reduction in volume of dry air by reducing the temperature a few degrees is of little account. If dry air at a pressure of 2 inches of mercury is reduced in temperature from 95 to 85 degrees Fahrenheit, the diminution in volume is only 1.8 per cent; but, if air saturated with water vapor at this pressure is reduced in temperature from 95 to 85 degrees Fahrenheit, the volume will be diminished 56 per cent.

The accompanying tables show the volumes of air-steam mixtures such as are handled by air pumps, at vacuums ranging from 24 to 29 inches, referred to a 30-inch barometer. For example, Table 1 deals with a 24-inch vacuum; this means that the vacuum is 24 inches when the barometer stands at 30 inches. Moreover, the table is applicable whenever the vacuum is 6 inches less than the barometer; hence, it would apply when the barometer is at 29 inches and a mercury gage indicates that the vacuum at the exit end of the condenser is 23 inches.

The air contained in water, used for boiler feed or for condensing purposes prior to entering the pump, varies in amount up to about 4 per cent. by volume. Water is actually incapable of retaining air in solution to the extent of 4 per cent., but the mixture of gases found in water differs more or less from atmospheric air, although usually spoken of as such. Moreover, water often temporarily contains more air than it can permanently hold in solution.

A boiler-feed pump or condensing-water pump having air-tight glands and drawing its supply from below a quiescent water level can add nothing to the air already contained in the water.

In the first column of each table is given the percentage by volume of air in water, and the second column expresses this relation by weight. Column *A*, Table I, gives the volume of the air at a temperature of 121 degrees Fahrenheit if removed from the water and perfectly free from water vapor at a vacuum of 24 inches. The figures in this column are in themselves of no practical use in connection with a condensing plant, as the air which has to be dealt with by the pump is never free from water vapor. This column, however, is given to show to what extent the volume of air is augmented by the water vapor.

In column 3, Table I, the conditions as regards temperature and vacuum are the same as in column *A*, but the air is saturated with water vapor. This is the condition of the air when handled by the air pump. By comparing the figures in columns 3 and *A*, the large increase in volume due to the water vapor will be apparent. The amount of water vapor is, however, relatively small at this temperature. In column 5, where the temperature is 10 degrees higher, the volumes are nearly doubled. This increase in volume is due not so much to the increased temperature of the air — this would account for only about 2 per cent. increase in volume — as to the much greater proportion of water vapor which is associated with the air at this temperature. In column 7 the temperature is high and the volume of air and vapor greater. The various columns in the succeeding tables express similar conditions to those in Table I. Figures corresponding to those in column *A*, Table I, are not given in the succeeding tables because they have no direct bearing on the present discussion.

In a jet condenser the air contained in the feed water is of relatively small importance, as a much larger quantity enters the condenser along with the condensing water. In deciding upon the capacity of an air pump for a jet condenser, the average percentage of air in the injection water should be ascertained, if possible, and if there is much variation in the percentage, the maximum percentage should be ascertained. Then from the weight of condensing water to be pumped per minute the re-

TABLE 1. 24-INCH VACUUM.

1	2	A	3	4	5	6	7	8
Air in Water.		Volume of Same Air at 121 Degrees F. and 24 Inches Vacuum, Dry. Cubic Feet per 100 Cubic Feet of Water.	Volume of Same Air Saturated with Water Vapor at 121 Degrees F., and at 24 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 131 Degrees F., and at 24 Inches Vacuum		Volume of Same Air Saturated with Water Vapor at 135 Degrees F., and at 24 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water		Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	2.795	6.88	0.001103	12.18	0.00195	19.5	0.003125
1.0	1.226	5.590	13.76	0.002206	24.36	0.00390	39.0	0.006250
1.5	1.84	8.385	20.64	0.003309	36.54	0.00585	58.5	0.009375
2.0	2.45	11.180	27.52	0.004412	48.72	0.00780	78.0	0.012500
2.5	3.07	13.975	34.40	0.005515	60.90	0.00975	97.5	0.015625
3.0	3.68	16.770	41.28	0.006618	73.08	0.01170	117.0	0.018750
3.5	4.29	19.565	48.16	0.007721	85.26	0.01365	136.5	0.021875
4.0	4.90	22.360	55.04	0.008824	97.44	0.01560	156.0	0.025000

TABLE 2. 26-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 106 Degrees F. and at 26 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 116 Degrees F., and at 26 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 120 Degrees F., and at 26 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	9.8	0.00158	17.4	0.00279	29.4	0.00472
1.0	1.226	19.6	0.00315	34.8	0.00558	59.0	0.00945
1.5	1.84	29.5	0.00473	52.4	0.00838	88.5	0.01419
2.0	2.45	39.3	0.00630	69.5	0.01115	117.9	0.01888
2.5	3.07	49.2	0.00789	87.2	0.01399	147.8	0.02365
3.0	3.68	59.0	0.00945	104.8	0.01678	176.9	0.02835
3.5	4.29	68.7	0.01101	121.8	0.01950	206.0	0.03302
4.0	4.90	78.5	0.01260	139.2	0.02230	235.5	0.03775

TABLE 3. 27-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 95 Degrees F. and at 27 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 105 Degrees F., and at 27 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 109 Degrees F., and at 27 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	11.98	0.00192	21.0	0.00337	32.3	0.00518
1.0	1.226	23.95	0.00384	42.0	0.00674	64.6	0.01036
1.5	1.84	35.55	0.00570	63.0	0.01012	96.9	0.01556
2.0	2.45	47.90	0.00768	84.0	0.01348	129.2	0.02070
2.5	3.07	60.00	0.00962	105.0	0.01685	161.5	0.02595
3.0	3.68	72.00	0.01152	126.0	0.02022	193.8	0.03110
3.5	4.29	83.70	0.01342	147.0	0.02360	226.1	0.03620
4.0	4.90	95.80	0.01535	168.0	0.02695	258.4	0.04145

TABLE 4. 27.5-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 89 Degrees F., and at 27.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 99 Degrees F., and at 27.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 103 Degrees F., and at 27.5 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	13.91	0.00223	24.96	0.00400	40	0.00641
1.0	1.226	27.82	0.00447	49.92	0.00800	80	0.01282
1.5	1.84	41.73	0.00670	74.88	0.01200	120	0.01923
2.0	2.45	55.64	0.00894	99.84	0.01600	160	0.02564
2.5	3.07	69.55	0.01117	124.80	0.02000	200	0.03205
3.0	3.68	83.46	0.01341	149.76	0.02400	240	0.03846
3.5	4.29	97.37	0.01564	174.72	0.02800	280	0.04487
4.0	4.90	111.28	0.01788	199.68	0.03200	320	0.05128

TABLE 5. 28-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 81.5 Degrees F., and at 28 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 81.5 Degrees F., and at 28 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 95.5 Degrees F., and at 28 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	17.22	0.00276	30.6	0.00490	49	0.00785
1.0	1.226	34.44	0.00552	61.2	0.00980	98	0.01570
1.5	1.84	51.66	0.00828	91.8	0.01470	147	0.02355
2.0	2.45	68.88	0.01104	122.4	0.01960	196	0.03140
2.5	3.07	86.10	0.01380	153.0	0.02450	245	0.03925
3.0	3.68	103.32	0.01656	183.6	0.02940	294	0.04710
3.5	4.29	120.54	0.01932	214.2	0.03430	343	0.05495
4.0	4.90	137.76	0.02208	244.8	0.03920	392	0.06280

TABLE 6. 28.5-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 72 Degrees F., and at 28.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 82 Degrees F., and at 28.5 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 86 Degrees F., and at 28.5 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	21.15	0.00339	36.95	0.00592	60.5	0.00968
1.0	1.226	42.30	0.00678	73.90	0.01184	121.0	0.01936
1.5	1.84	63.45	0.01017	110.85	0.01776	181.5	0.02904
2.0	2.45	84.60	0.01356	147.80	0.02368	242.0	0.03872
2.5	3.07	105.75	0.01695	184.75	0.02960	302.5	0.04840
3.0	3.68	126.90	0.02034	221.70	0.03552	363.0	0.05808
3.5	4.29	148.05	0.02373	258.65	0.04144	423.5	0.06776
4.0	4.90	169.20	0.02712	295.60	0.04736	484.0	0.07744

TABLE 7. 28.8-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 65 Degrees F., and at 28.8 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 75 Degrees F., and at 28.8 Inches Vacuum.		Volume of Same Air Saturated with Vapor at 79 Degrees F., and at 28.8 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	26.6	0.00426	45	0.0072	71.8	0.0115
1.0	1.226	53.2	0.00852	90	0.0144	143.6	0.0230
1.5	1.84	79.8	0.01278	135	0.0216	215.4	0.0345
2.0	2.45	106.4	0.01704	180	0.0288	287.2	0.0460
2.5	3.07	133.0	0.02130	225	0.0360	359.0	0.0575
3.0	3.68	159.6	0.02556	270	0.0432	430.8	0.0690
3.5	4.29	186.2	0.02982	315	0.0504	502.6	0.0805
4.0	4.90	212.8	0.03408	360	0.0576	574.4	0.0920

TABLE 8. 29-INCH VACUUM.

1	2	3	4	5	6	7	8
Air in Water.		Volume of Same Air Saturated with Water Vapor at 59.5 Degrees F., and at 29 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 69.5 Degrees F., and at 29 Inches Vacuum.		Volume of Same Air Saturated with Water Vapor at 73.5 Degrees F., and at 29 Inches Vacuum.	
Cubic Feet of Free Air at 60 Degrees F. per 100 Cubic Feet of Water.	Pounds of Air per 100,000 Pounds of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.	Cubic Feet per 100 Cubic Feet of Water.	Cubic Feet per Pound of Water.
0.5	0.613	31.25	0.00501	57	0.00913	84.5	0.01355
1.0	1.226	62.50	0.01002	114	0.01826	169.0	0.02710
1.5	1.84	93.75	0.01503	171	0.02739	253.5	0.04065
2.0	2.45	125.00	0.02004	228	0.03652	338.0	0.05420
2.5	3.07	156.25	0.02505	285	0.04565	422.5	0.06775
3.0	3.68	187.50	0.03006	342	0.05478	507.0	0.08130
3.5	4.29	218.75	0.03507	399	0.06391	591.5	0.09485
4.0	4.90	250.00	0.04008	456	0.07304	676.0	0.10840

quired capacity of the air pump can be calculated. An allowance must be made for air entering the condenser with the steam. The amount of such air depends chiefly upon the tightness of the glands and joints. With a low vacuum and cold injection water, containing less than 2 per cent. air, the weight of air which enters the condenser with the injection water is relatively small and may be less in amount than that which enters with the steam, even if there is no excessive leakage; but, as a rule, and especially with high vacuum, the air which enters with the injection water constitutes the greater proportion of the total air; in the case of a steam-turbine condensing plant, this may constitute over 75 per cent. of the total air.

Where jet condensers are used the air and water are discharged by separate pumps and a certain amount of the air is discharged with the water. In the case of barometric jet condensers where the water is discharged through a tail pipe without the use of a pump, a certain proportion of the air is carried away by the water. The amount of work which the air pump is thus relieved of depends upon the design of the condenser.

The displacement per minute of the air-pump piston does not represent the effective capacity of the pump, which is the product of the piston displacement and the volumetric efficiency, the latter varying greatly.

In a surface condensing plant the greater part of the air handled by the air pump enters the system as leakage, consequently the weight of air per pound of steam varies greatly. If only the air contained in the feed water had to be dealt with by the air pump, the latter could be of much less capacity than is usually necessary. Therefore, columns 1 and 2 of the tables are of little use in connection with air pumps for surface condensers which usually discharge the water of condensation as well as the air and vapor. The tables are useful, however, in connection with a surface condensing plant chiefly to show how the volumes of air and vapor are affected by the vacuum and temperature in the condenser.

Although it is practically impossible to determine with great precision the necessary dimensions of an air pump, a rational treatment of the subject should precede the actual design; in this way many serious blunders may be avoided. It is no exaggeration to say that in some cases air pumps are made 100 per cent., and occasionally 200 per cent., too large, while in other cases they are much too small. A pump intended to maintain a certain vacuum, but of too small a capacity, will suck all the air as fast as it enters the condenser, but it will do so only when the vacuum in the condenser has fallen so much below the intended vacuum that the volume of the air and vapor is reduced to an amount corresponding to the effective capacity of the pump. By comparing the various tables with each other it will be apparent that the volume decreases very rapidly with the vacuum.

The Confession of an Engineer

By R. O. Warren

Most engineers have failed to make good at some time because they did not use their knowledge in a practical way. In this story the engineer never thought of keeping records and computing the coal consumption per horsepower-hour. The manager did.

After a man has spent twenty-seven years in an engine room it would be strange if he could not remember almost countless instances where mistakes had been made, either through ignorance or lack of judgment, and lost opportunities for making good because of lack of interest and enterprise.

It was only a few days ago, while sitting in my engine room that I began to recall some of the mistakes I have made during my engineering career. The longer I thought of these things the more I became impressed with the fact that the numerous errors I had committed and the greatest mistakes I had made had been due to failure in putting the knowledge I possessed to practical use to the advantage of both my employer and myself.

Such musings added greater weight to the conviction I already had that a man can know a whole lot about engineering, but if he fails to use that knowledge he is worth no more to his employer than a man who knows less, but uses what knowledge he does possess in a practical way.

I had had about three years' experience as an engineer at the time of which I speak, and, although I knew considerable about power-plant operation, a habit had been formed of letting well enough alone so long as things were moving along apparently in a satisfactory manner. No greater mistake is ever made by an engineer than when he gets the idea that conditions cannot be improved. No steam plant has ever reached such a point of perfection that it cannot be made better and more efficient. If every engineer would only fix this fact firmly in mind early in his career, he would be a much better engineer and of more value to his employer.

I did not see the matter in this light during my early engineering training, but got the lesson after passing through many humiliating experiences which, of course, lowered my standing in the eyes of my employer. About the first intimation I had that the manager was looking after his interests in the engine room occurred one day after I had been with the Holden Manufacturing Company about eight months. The firm had followed for years the practice of purchasing the coal supply from a local coal dealer. My past experiences have removed all doubt that most coal dealers fail to deliver the grade of coal ordered and seldom if ever duplicate an order with the same grade of coal, although it is sold under the same name and at the same price.

Evidently the new manager had had some experience with engineers who knew things and had acted upon their advice, for upon the day in question he dropped into the engine room and asked in a casual way how the coal was holding out. I answered that there was enough to last about four weeks, but that there was no need of getting anxious because if the new coal did not arrive on time, Miner & Hall, the coal dealers, would supply all we wanted and that it did not make much difference one way or the other as they furnished all the coal we used anyway.

Upon receiving this information, Wood, for that was the new manager's name, sat down and, after lighting a cigar, asked in an "I-am-not-interested" way how much coal I used in getting a horsepower out of the engine.

Now I knew how to figure the horsepower of an engine for I had worked out hundreds of such problems. Not only that, but I owned an indicator outfit and could take a card in rather an expert manner. So familiar was I with the operation and the figuring that the formula

$$\frac{PLAN}{33{,}000} = I.H.P.$$

was one of the easiest that could have been proposed.

Being energetic in some ways, I had expounded this very formula and covered the blackboard with figures down at the association time and time again, and yet the new manager had me stumped at the first question he had asked. For I did not know how many pounds of coal I was burning per horsepower developed by the engine. Can anyone imagine a more humiliating position for an engineer to occupy?

I was not man enough to own up to

my ignorance and tried to give the impression I knew by answering, "Oh, about four pounds." I had heard someone say that was a good average.

"Four pounds!" exclaimed Wood, "you don't mean that you are burning four pounds of coal for each horsepower developed by your engine, running condensing at that? Why, that is excessive."

"Ah, it might be a little less," I replied, feeling mighty uncomfortable, for dim recollections of having read of two pounds of coal for a condensing engine began to flit through my brain.

"But you are not sure," said he, "and now, honest, Warren, did you ever take enough diagrams from your engine to know the total average horsepower developed for the day's run?"

I had to admit that he had the situation sized up to perfection and, seeing I was caught "with the goods," admitted I did not know how much coal was required per horsepower-hour in our plant. I even went a little further and admitted I did not know how much coal was being burned each day.

Wood looked surprised, and I felt ashamed.

The manager recovered first and said, "Surely you are aware of the importance of knowing how much coal is being consumed. Coal is the most expensive item in power-plant operation, and the man who can save coal is the man who is appreciated."

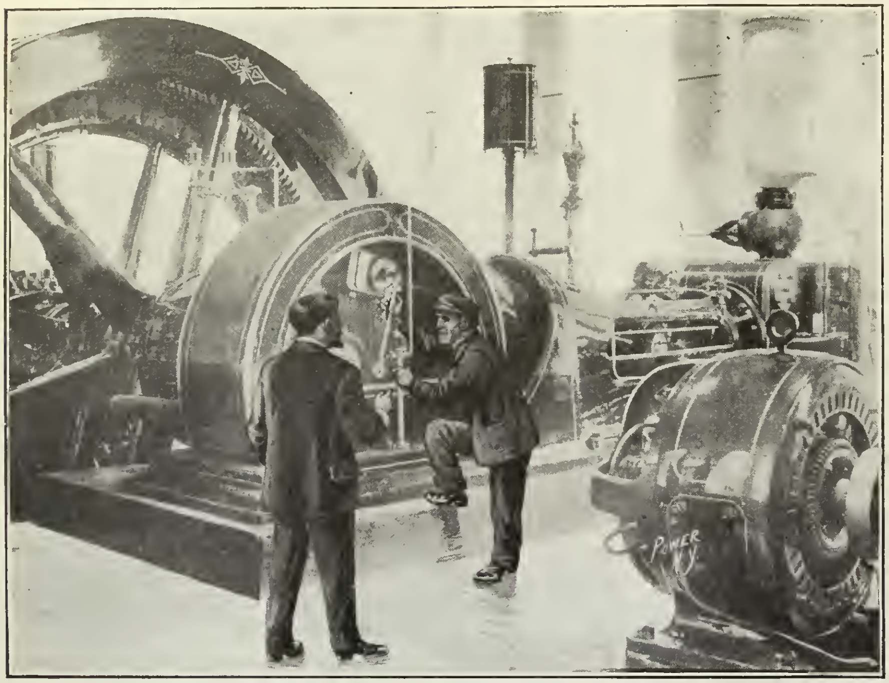
I could not very well say anything, seeing I was "in up to my eyes," and so wisely remained silent.

"Let us take this plant just as it is," he continued. "Here is an engine having a cylinder 48x72 and the engine runs at what speed?"

"Fifty-five," I almost yelled, glad that I could answer at least one question intelligently.

"Let's see, that would be about 2100 horsepower at one-quarter cutoff with your steam pressure of 110 pounds per square inch."

I nodded my head, although I did not know what the horsepower of the engine would figure at one-quarter cutoff. I had been told that the engine had a capacity of 2400, but had never taken the trouble to find out what horsepower was being developed at various ranges of cutoff.



The New Manager Wanted to Know How Much Coal I Burned Per Horsepower Developed

"Suppose that the engine required four pounds of coal for each horsepower developed, that would mean a coal consumption of 8400 pounds per hour, or 84,000 pounds each working day, not figuring in the coal necessary for starting and banking fires."

I admitted that he was right, but said no more, not knowing at what he was driving.

"Now, as coal costs $4.10 a ton, the coal bill would amount to $172.20 a day that the company must pay for that one item alone."

I nodded my head for I had figured the thing out, being mentally quick at figures when I wanted to be.

"I suppose you have heard the saying that a good fireman can more than save his wages. Well, that applies to an engineer as well. If an engineer can save a pound of coal, or even a fraction of a pound for each horsepower developed by the engine, it won't require much of a saving to more than equal his wages, in, say, a plant like this."

I nodded my head again, and as I began to see the drift of his talk, cold chills chased each other up my spine.

"Now," went on Wood, "if this plant requires four pounds of coal for each horsepower developed, which I very much doubt, and you reduce the coal consumption one pound for each horsepower per hour, there would be a saving of 2100 pounds of coal each day, over a ton, a saving of $4.30 a day. That's about as much as you get, isn't it?"

"More," I answered, "I get $3.50 a day," and to tell the truth I felt ashamed to think that I might possibly be wasting more coal value than the amount of my pay.

"Of course," Wood continued, "if the saving from four to three pounds amounts to so much, reducing the coal consumption from three to two pounds, which should be obtained from the equipment we have here, would save an equal amount. Of course, it is my business to run this company as economically as it is possible, and at the same time keep it in a strong operating condition. The steam end is one of the great expenses that shows no return for the money invested.

"Now, in order to get down to business, I want a report each week showing the indicated horsepower at some stated hour, about the middle of the forenoon of each day for one week and the same for the afternoon of each day of the following week. In that way a good average of the horsepower output can be had.

"It would not be a bad idea for you to find out just how much it is costing to develop a horsepower. Then the amount of the coal used each day should go on that report, and any other item that would be of value in checking the average daily performance of the plant."

Wood leisurely arose and, after stretching himself, walked out of the engine room. His manner indicated that he had no further concern about the plant, but I knew better. I had sense enough to know that his friendly talk on coal was a mild warning for me to look after that end of my work.

After Wood had disappeared I sat down and thought things over, and after due meditation began to cuss myself for a blank fool. I had allowed a good chance to make good go by. I knew how to indicate an engine, how to figure a diagram, how to set the valves, but had never had the gumption to take enough cards to get the average horsepower output of the engine for a week, and weigh the coal consumed in order to know how much coal was used in producing a horsepower.

If I had done these things and gone to the new manager before he came to me, and told him how much coal was being used per horsepower developed. I would have had a good start in making good.

If I had looked into this matter—had known that each horsepower demanded a consumption of three pounds of coal—knowing that the engine was in good condition, I would have been in a position to have said, "The coal we have is not as good as it should be. There is altogether too much ash and not enough heat units in it. We can make a saving if we insist on getting a better grade of coal."

But I had not done these things. I was as thousands of other engineers were then and are now, content to start and stop the engine and draw the weekly pay. I knew how to do these things, but did not know enough to realize that the management of any company appreciates the efforts anyone in its employ may make to reduce the running expenses.

Of course, after the talk with Wood I indicated the engine as suggested and weighed my coal, and, although I knew it was the proper thing to do, I felt a resentment against his coming into my engine room and suggesting methods of operation different from what I had been accustomed to follow.

But after a few days this feeling gradually wore away and I consoled myself with the thought that I was not the only engineer who did not know how much coal was being burned per horsepower developed.

The more I thought along that line the more satisfied I felt with myself, and, as a result, instead of profiting by my little experience with Manager Wood, I began to dose again instead of looking around for another opportunity to make good.

A Boiler Efficiency of 82.36 per Cent.

> *For construction purposes at the Panama canal there are installed twelve Keeler boilers aggregating a total of 5000 horsepower. During the government acceptance tests, one of these boilers showed an efficiency of 82.36 per cent. and an economic evaporation of 13.10 pounds of water from and at 212 degrees per pound of combustible.*

What are probably the largest plants ever built for construction purposes are two installations aggregating 5000 boiler horsepower which are at present in service at the Gatun and Miraflores handling plants at the Panama canal. These plants generate electricity for the operation of the cableways by which the concrete is handled and after the canal is completed it is planned to maintain the plants as an auxiliary to the permanent operation of the locks.

At each plant there are six Keeler water-tube boilers arranged in batteries of two each. They are erected in special steel casings entirely inclosing the front, sides and rear, these casings being lined with 2 inches of magnesia block and 9 inches of firebrick.

The Government specifications for these boilers required an evaporative efficiency of 65 per cent. when evaporating 17,000 pounds of water from a temperature of 210 degrees Fahrenheit at a gage pressure of 205 pounds absolute and 150 degrees of superheat. There was a further provision for a bonus of $1000 for each per cent. efficiency above the contract requirements of 65 per cent. The acceptance test on one of the boilers, the results of which are herein given, was conducted by Government engineers, without the assistance or presence of firemen, engineers or representatives of the manufacturers, and, therefore, may be considered as impartial. Extreme care was taken to have all the instruments standardized and to have every item of the test as nearly correct as possible. Two white firemen were employed throughout the test, relieving each other every thirty minutes. The coal used was weighed on platform scales at the front of the boiler and samples were taken from each wheelbarrow load. The feed water was pumped direct from the heater into weighing tanks, located over the heater, and discharged into two small tanks on the floor of the turbine room, from which it was pumped direct to the boiler under test. Owing to the condition of the Orsat apparatus it was impossible to determine the percentage of carbon monoxide and nitrogen present in the flue gases. The high percentage of CO_2 was obtained by close supervision of the draft, which was regulated according to the changes in load. The results of the test on the basis of contract requirements showed an efficiency of 82.36 per cent. Each plant was accepted and the efficiency bonus paid immediately after the completion of the official tests. The report of the test at the Miraflores plant is as follows:

BOILER TEST—MIRAFLORES POWER PLANT.

Kind of fuel	Pocahontas coal.
Method of starting and stopping test	Alternate
Duration of trial	8 hours

DIMENSIONS AND PROPORTIONS OF BOILER.

Grate surface, square feet	64.965
Height of furnace 45.25 front, 39 inches rear.	
Approximate width of air spaces in grate, inches	0.625
Proportion of air space to whole grate surface, per cent	46.87
Water-heating surface, square feet	4079.408
Superheating surface, sqare feet	1084.50
Ratio of water-heating surface to grate surface	62.7 to 1

AVERAGE PRESSURES.

Steam pressure at boiler, gage	188.06
Draft below damper, inches of water	0.412
Draft in furnace, inches of water	0.306
Draft in ashpit, inches of water	0.062

AVERAGE TEMPERATURES

Steam from superheater, degrees Fahrenheit	481.79
Degrees of superheat	98.79
Feed water entering boiler, degrees Fahrenheit	183.57
Escaping gases, degrees Fahrenheit	473.62

FUEL.

Size and condition	Run-of-mine.
Dry coal consumed per hour, pounds	1,540.21
Combustible consumed per hour, pounds	1,560.64
Dry coal per square foot of grate surface per hour, pounds	18.966
Combustible per square foot of water-heating surface per hour, pounds	2.942

CALORIFIC VALUE OF FUEL.

Calorific value, by Parr calorimeter, per pound of dry coal, B.t.u.	15,292
Calorific value, by Parr calorimeter, per pound of combustible, B.t.u.	15,683.4

ECONOMIC RESULTS.

Water apparently evaporated under actual conditions, per pound of coal as fired, pounds	10.30
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of coal as fired, pounds	12.53
Equivalent evaporation from and at 212 degrees Fahrenheit per pound of dry coal, pounds	12.76
Equivalent evaporation fromand at 212 degrees Fahrenheit per pound of combustible, pounds	13.10
Equivalent evaporation per hour from and at 212 degrees Fahrenheit per square foot of water-heating surface, pounds	4.82

HORSEPOWER.

Horsepower developed	569.97
Builders' rated horsepower	400
Percentage of builders' rated horsepower developed	142.49

EFFICIENCY.

Efficiency of the boiler, including the grate; heat absorbed by the boiler, per pound of dry coal, divided by the heat value of one pound of dry coal (feed water at 210 degrees Fahrenheit and steam pressure 190 pounds gage) per cent	82.36

COST OF EVAPORATION.

Cost of coal, per ton of 2240 pounds, delivered to boiler room	$4.25
Cost of fuel for evaporating 1000 pounds of water under observed conditions	$0.1841
Cost of fuel for evaporating 1000 pounds of water from and at 212 degrees Fahrenheit	$0.1513

METHOD OF FIRING.

Kind of firing	Spreading
Average thickness of fire, inches	7.666
Average interval between firings during time when fires are in normal condition, minutes	5.23
Average interval between leveling, minutes	26.38

ANALYSES OF DRY GASES.

Carbon dioxide (CO_2), per cent	12.458
Oxygen (O), per cent	2.747

An Engine Built Up of Sections

By A. B. Gould

To meet the demand for a steam engine in a locality which was accessible only by mule pack, a 14x42-inch engine was constructed of sections, none of which weighs over 350 pounds. The engine complete consists of over one thousand pieces.

Occasionally it is necessary to develop power at points remote from any railway, and to which it is not possible to build a wagon road for the purpose of transporting material. In such cases everything must be carried on the backs of mules, over hills and along trails where in some instances it is scarcely possible for even a mule to find a foothold. Machinery to be thus transported must be especially constructed so that when taken apart for shipment, no one piece will be of greater weight than can be easily carried by a mule. This often necessitates a departure from the usual types of construction, as the process of sectionalization sometimes introduces unusual conditions that must be met by unusual construction. Some kinds of machinery are adaptable to the cutting-up process, while others present difficulties that tax the ingenuity of the designer. As an illustration of how a Corliss engine was thus treated, the following description is given.

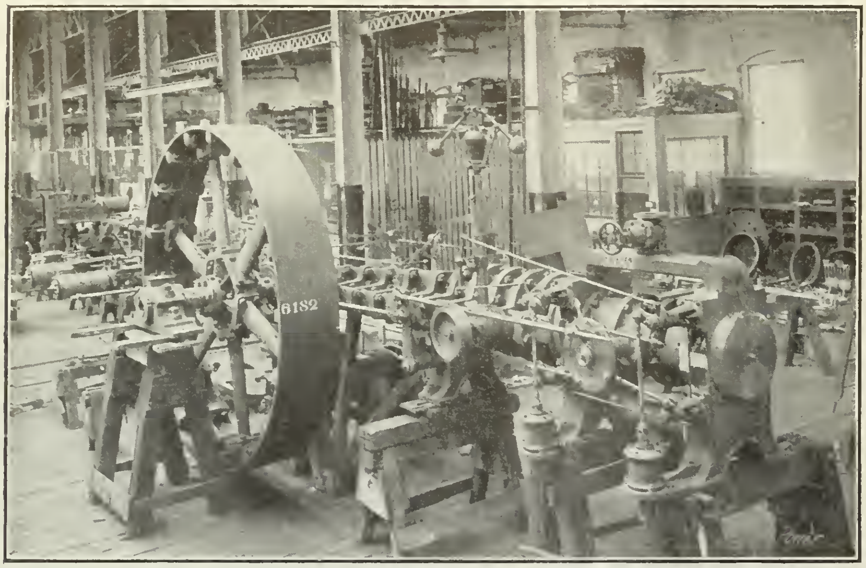


Fig. 1. Engine Assembled in Shop

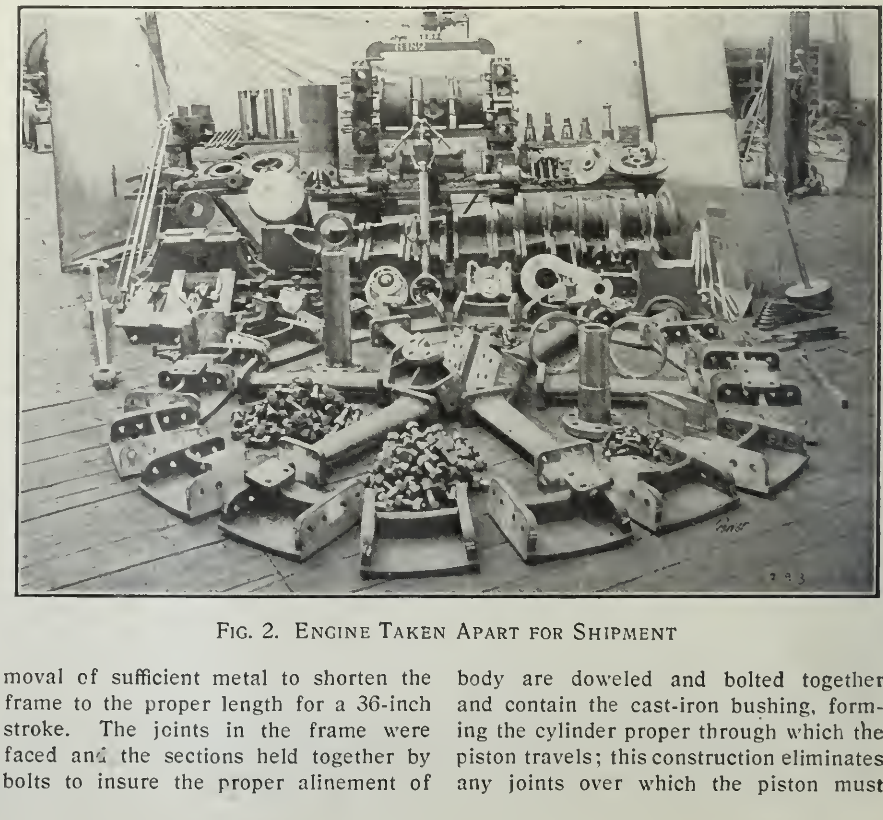


Fig. 2. Engine Taken Apart for Shipment

This engine, which is 14x36 inches and runs at 100 revolutions per minute, was built by the Wellman-Seaver-Morgan Company and was recently shipped from its Akron works to the Cia Minera Jesus Maria y Anexas in Mexico.

It may be seen from the illustrations that the engine is built with a girder frame. Owing to the class of work required, it was possible to use this type of frame, which was more easily sectionalized than any other. In this case a pattern for a solid 14x42-inch engine frame was changed by adding bolting flanges at the proper intervals and putting in splitting cores to break up the casting into pieces. These splitting cores were made thick enough to allow for the removal of sufficient metal to shorten the frame to the proper length for a 36-inch stroke. The joints in the frame were faced and the sections held together by bolts to insure the proper alinement of all the pieces. The pillow-block end is separate from the frame, and is divided into four pieces. The bearing is made up in the usual quarter-box manner, providing for all necessary adjustments, and the center of the frame rests on a leg which forms a support for the rocker arm.

The valve gear is of the standard releasing type, operated by a single wrist-plate attached to the side of the cylinder, and the steam valves are closed by noiselessly operating dashpots which contribute much to quiet running.

The flywheel is 10 feet in diameter and has a 20-inch crowned face. It is made in twenty-six pieces, securely bolted together with fitted bolts where necessary to preserve the proper relation of the various parts. The arms and rim are of cast iron, and the hub of cast steel, the latter being in two pieces bolted together, and still further secured by two steel rings shrunk on the outside of the flanges where bolted to the shaft. The shaft is in two pieces, one bolted to either side of the flywheel hub, making the hub part of the shaft. These two sections of the shaft are annealed-steel castings with a hole cored through each. This construction allows a considerable saving in weight, with but a small loss in strength.

The most interesting part of the entire engine is the cylinder. The barrel is in four pieces with the valve chests separate. The three outside pieces of the body are doweled and bolted together and contain the cast-iron bushing, forming the cylinder proper through which the piston travels; this construction eliminates any joints over which the piston must travel. To each end of the barrel are bolted the chests containing the steam and exhaust valves. These chests are divided horizontally at the center line of the cylinder. The steam chest is a separate casting and is bolted to the valve chest directly over the steam valve, while the exhaust from each end of the cylinder is carried down through the cylinder leg into the exhaust chest and from there to the exhaust pipe. The whole cylinder is lagged with magnesia and is covered by a steel jacket.

The other parts of the engine are of standard construction, and did not require to be made in sections. No part as shipped exceeded 350 pounds in weight, this being the limit imposed.

Fig. 1 shows the general appearance of the engine, minus the cylinder jacket, as it stood on the erecting floor, and Fig. 2 is a view taken after it had been dismantled and was ready for shipment. It is interesting to note that there are over one thousand pieces in this engine, all of which were successfully transported and erected with the loss of only a piston ring.

Charging a Refrigerating System

By F. E. Matthews

The author gives explicit directions for charging and a number of tables devoted to cubical contents of pipes and weights of gas at different pressures, are presented to assist in estimating the amount of charge required.

After it has been ascertained that the system is perfectly tight, the air and the ammonia should be allowed to escape, and the whole system should be pumped down to a vacuum. The expansion valve is first closed, then the valve on the drum of ammonia connected with the charging valve is opened. The liquid is allowed to flow through the slightly opened expansion valve into the system under the pressure of the gas above the liquid in the ammonia drum, just as water is forced out of a boiler by the pressure of the steam. It was suggested that the system be pumped out to a vacuum, not because a vacuum is necessary to draw the liquid around into the system, but rather to get rid of as much of the air as possible. While the production of a lower pressure within the refrigerating system than that of the atmosphere without undoubtedly hastens the operation of charging, there will be a tendency to draw air or water into the system, so that a vacuum should never be pumped until it has been demonstrated beyond doubt that the system contains no leaks. Small leaks into the system are not readily detected, and much more trouble can be made by what water a small leak will let into a system than by the amount of ammonia the same leak will let out.

Even pumping a vacuum does not insure the expulsion of all the air. The remainder may be allowed to stay in the system until displaced by purging at the condensers, or a large percentage of it can be driven out of the system during the process of purging by the judicious manipulation of the ammonia. If the ammonia be admitted very slowly at one end of a long run of pipe, it will drive the air before it without mixing with it to any great extent, and, if at the other end of the pipe line a valve be opened or a flange union be cracked after sufficient ammonia has been admitted to produce a pressure above that of the atmosphere, the air can be allowed to escape until it contains too large a percentage of ammonia, when the opening is closed. That the liquid ammonia will pass from the shipping drum into the system without the necessity of pumping is evident.

If the engine-room temperature be 80 degrees Fahrenheit, for example, the pressure in the drums will be 140 pounds. If the drums are exposed to the sun, the temperature may rise much higher than that of the surrounding air and dangerous pressures may result. It is accordingly advisable to store ammonia drums in a cool place. The shipping drums are designed to carry any reasonable pressures but there is a remote possibility that the drum may be filled too full. In this case, since there is not sufficient vapor space to take care of the expansion of the liquid as the temperature increases, and since liquids are practically incompressible, there is no limit to the amount of pressure that may be produced except that of the ultimate strength of the drum. There is the same danger in tightly closing the valves on all the outlets to the liquid receivers when it is not definitely known that they are not completely filled with liquid. Explosions due to such causes are second only to boiler explosions in their disastrous results.

It is obvious that the vapor generated in the drum will drive the liquid out into the system so long as the temperature of the liquid is such as to produce gas to keep the pressure higher than that in the system. If, for example, the system is operating under 15 pounds gage back pressure, 16 pounds vapor pressure in the drum would suffice to expel the liquid. The temperature of the liquid corresponding to a pressure of 16 pounds is about zero degree Fahrenheit. From this it will be seen that the only disadvantage of charging against back pressure is that the liquid will not flow so rapidly into the system because of the decreased difference in pressure. The slightest reduction in pressure within the drum, due to a removal of part of the liquid, causes the ammonia to boil more vigorously, generating more vapor to fill the increasing space above the liquid. The temperature at which the liquid boils gradually drops, however, until at a pressure of about 47 pounds, which corresponds to a temperature of a little less than 32 degrees Fahrenheit, the pipe leading from the drums will become sufficiently cold to precipitate and congeal atmospheric moisture, and is said to "frost." The melting of this frost indicates that there is no more ammonia passing through the pipe, and the valves can accordingly be closed, the empty drum removed and a full drum connected.

So far as the production of cold is concerned, there need be only sufficient liquid refrigerant to insure a solid stream at the expansion valves, so that no gas may enter the expansion coils at any time. This condition can be readily recognized by the intermittent hissing sound produced by the passage of quantities of liquid and gas. The condition in which there is just enough liquid to give a solid flow at the expansion valve is the minimum charge that can be economically employed without loss of both capacity and efficiency.

To provide for unforeseen contingencies, such as losses of ammonia through leaks, temporary trapping of liquid in low parts of the system, etc., it is always expedient to have the liquid charge somewhat in excess of this amount, a kind of credit balance in the bank to guard against the embarrassment of overdrawing one's account if collections do not come in from the freezer, expansion coils, condensers, etc., as expected.

Increasing the charge of anhydrous ammonia above that actually required to insure an uninterrupted flow at the expansion valve, will work no harm to the system up to the point where the liquid fills the receiver and begins to encroach upon the condensing surface. To be sure, the more liquid lying in the compression side of the system under the usual conditions of operation, the less space there will be for the storing of additional anhydrous ammonia should it become necessary to "pump out" any or all of the coils of pipe in the low-pressure side. The additional ammonia occasions an additional investment, but in the majority of plants it is expedient to carry a small stock of liquid to provide for contingencies; and aside from the rather remote possibility of an accident that would result in the loss of the entire charge, it is better to have the ammonia in use in the system than lying idle in shipping cylinders.

An overcharge of ammonia in a system can usually be detected in two different ways. Since the condensed liquid is usually several degrees colder than the uncondensed gas, the parts of the compression side that contain liquid ammonia, whether the liquid receiver connecting piping or pipes of the condenser, can usually be determined by their temperatures. Assuming that the plant is overcharged to such an extent that the liquid in the compression side begins to fill up the condenser, thus encroaching on the available condenser surface, a material rise in head pressure over that usually observed when running under similar conditions of speed, back pressure and water supply with only sufficient liquid in the system to insure a solid flow through the expansion valve would be expected, although a slight increase in head pressure observed on increasing the charge should not be accepted as positive proof of overcharging.

TABLE 1. RELATION OF CUBICAL CONTENTS TO RUNNING FEET IN PIPES OF VARIOUS SIZES.

Size of Pipe, Inches.	Running Foot per Cubic Foot of Contents.	Contents in Cubic Feet per 100 Running Feet.
¾	270.00	0.370
1	166.90	0.599
1¼	96.25	1.038
1½	70.65	1.415
2	42.36	2.360

TABLE 2. WEIGHTS OF AMMONIA VAPORS AT DIFFERENT GAGE PRESSURES.

Ammonia Gage Pressure.	Weight of 1 Cubic Foot of Vapor, Lb.	Ammonia Gage Pressure.	Weight of 1 Cubic Foot of Vapor, Lb.
0	0.0566	80	0.3304
10	0.0941	90	0.3617
20	0.1269	100	0.3939
30	0.1611	125	0.4766
40	0.1955	150	0.5566
50	0.2292	175	0.6340
60	0.2641	200	0.7188
70	0.2965	...	......

There can be no fixed rule by which to determine the amount of ammonia required for a direct-expansion refrigerating system. For systems not including sharp freezers, the only accurate means is to calculate the amount of the charge, taking as a starting point the amount of pipe to be filled with the refrigerant in both the high- and the low-pressure sides of the system. The accompanying tables, showing cubical contents of pipes and weights of gas at different pressures, will be found convenient when calculating the amount of ammonia required to charge the system.

The number of hundreds of running feet of pipe in the system having been determined, the cubic feet contained in it may be found from Table 1. The amount of ammonia necessary may be ascertained by multiplying the cubical contents by the weight of gas per cubic foot corresponding to the pressure to be carried on the pipes when the system is in operation. The weight of ammonia vapor required to fill both high- and low-pressure sides of the system may be determined in this way, in addition to which a liberal margin should be allowed for reserve liquid in the receiver, evaporating liquid in the expansion coils and condensing liquid in the condenser.

TABLE 3. ANHYDROUS AMMONIA REQUIRED FOR THE COMPRESSION SIDE OF REFRIGERATING PLANTS.

Tons of Refrigeration.	Pounds of Ammonia.	Tons of Refrigeration.	Pounds of Ammonia.
5	110	75	375
10	150	100	440
15	185	150	510
20	230	175	570
25	245	200	620
30	270	225	675
35	290	250	725
40	300	300	840
45	325	400	1040
50	350	500	1215

Where sharp freezers are in service a much larger amount of liquid will be required to charge the low-pressure side, the extra charge increasing very rapidly with decreasing pressures.

When the refrigerating machinery is to be operated under average conditions an ammonia charge figured according to the following table will be in line with commercial practice.

TABLE 4. ANHYDROUS AMMONIA REQUIRED PER 100 RUNNING FEET OF PIPE—EXPANSION SIDE.

Refrigerating Plants. Direct Expansion and Brine Cooling Coils.	Size of Pipe.	Ice Plants. Expansion Coils for Can and Plate Use.
14 pounds.	1 inch.	8 pounds.
18 pounds.	1¼ inches.	11 pounds.
20 pounds.	1½ inches.	12 pounds.
25 pounds.	2 inches.	15 pounds.

In ice plants the amount of expansion surface per ton is more nearly a constant than in direct-expansion refrigerating plants. Since different sizes of expansion piping are used by different builders, the expansion surface does not always bear a fixed relation to the space to be filled with ammonia vapor. The following ammonia charges for ice-making plants may be considered in line with average practice.

TABLE 5. AMMONIA REQUIRED FOR ICE MAKING PLANTS.

Tons of ice per 24 hours..........	5	10	15	25	50	100
Pounds ammonia .	100	250	500	1000	2000	4000

The amounts given in Table 5 are for the total number of pounds required to charge both high- and low-pressure sides of the ice-making systems in question, while those in Table 3, are those required for the compression side only.

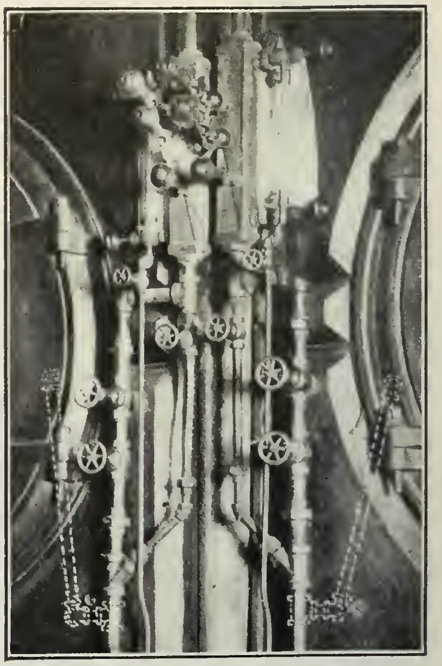
The amount of sodium chloride (NaCl) required to make brine for an ice tank of given capacity is also subject to wide variations on account of the diversity of designs used by the different builders.

A good general rule is to allow 15 pounds of salt per cubic foot of brine actually required to fill the tank when the cans are in place. Another rough rule is to allow two-thirds ton of salt per ton of ice-making capacity of tank per 24 hours. When calcium chloride (CaCl) is employed, some authorities estimate the amount required at one ton, per ton of ice-making capacity.

Trouble with Water Column

A trouble that many engineers have encountered in boiler-room practice is the failure of water to return to the water glass and column after they have been blown out.

In one instance the water columns on two boilers were originally piped as shown in the accompanying illustration by the dotted sections of piping. When the water column was blown down, the water failed to show at normal water level for about fifteen minutes.



Piping of Water Columns

When the top try cock on the water column was opened, the water would shoot to the top of the glass and stay there. When the bottom try cock was opened, the water would disappear for a period of from ten to fifteen minutes.

A boilermaker, several experts and some good engineers could give no explanation as to why the column should act in this way, and failed to suggest a remedy. The engineer finally made the connections between the boilers, as shown in the original lines of the photograph and the trouble disappeared.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Care of Rotary Converter Commutators

By W. C. Motley, Jr.

I have recently read several articles in Power dealing with rotary converters, but a few more general remarks on their care may be of interest to your readers. Many operators have the idea that the commutators in these machines cannot be kept in good order but, while I am aware that they require perhaps more attention than commutators on ordinary generators, the desired polish can be attained by close attention to details.

One very often finds commutators having very hard mica in them; the result is that when the copper wears away a little the brushes chatter and spark due to the poor contact caused by the high mica strips. The ordinary sandpapering is not effective in curing the trouble, and finally the superintendent orders the commutator turned down. This is wasteful and usually unnecessary, as the operator can keep the machine in order if he will follow the plan suggested.

Coarse and fine sandpaper are necessary and it is a good plan to rub the commutator with an oily rag, so that the dust will collect on the paper instead of in the machine. The commutator should be thoroughly rubbed with the coarse sandpaper until it is free of burnt spots. Then the fine paper should be used until the surface is bright and clean. In both of these scourings, the sandpaper block should be applied with considerable pressure. After the surface is well polished, apply the fine sandpaper again, *but do not press too hard this time.* This will just touch the high spots and will have the same effect as holding a lathe tool close enough to cut the mica without losing any copper.

Many operators advise cutting the mica away between bars, leaving a shallow groove between the faces of adjacent bars. No doubt this gives some good results, but there is a chance of dirt lodging in these spaces and making it possible for the machine to "flash over." It also puts the machine out of commission while the job is being done, which is objectionable in cases where the machines carry a heavy load.

For cleaning a commutator an oily rag is preferable to gasolene; gasolene sometimes leaves a sticky surface which causes the brushes to glaze and chatter. Be sure, however, to get all the oil off, as too much oil will cause trouble when the machine is running.

The direct-current brushes also require considerable attention and it is important that they be kept clean at all times; it is unwise to let them go until they become gummy. They should make perfect contact with the commutator and no cracked or "honeycombed" brushes should be tolerated. In sandpapering sets having a dozen or so brushes, it will be found much easier if the paper is moved *parallel* to the commutator bars and not in the usual direction, provided the design of the machine will allow it, as there is then no danger of rounding the toes and heels of the brushes. Of course, sandpapering each brush individually is much better than doing the whole set at once, but time is often too limited for that.

I have heard it said that commutators should be lubricated at stated times, but I do not see why this is so, as we all know that grease is a trouble maker. It is much better to let a machine alone as long as it runs quietly, greasing it only when absolutely necessary and then using as little as possible. The selection of a lubricant for commutators is also important. Some people use compounds and some use wax; others use dynamo oil or vaseline. The objection to most compounds and waxes is that if the machine runs cool they gum the brushes. Oil or vaseline will seldom do this and though they must be applied oftener, they usually give better results.

The alternating-current side of a rotary converter does not usually give much trouble. The only one worth mentioning is that the brushes sometimes cut due to stiff brushes, too much pressure or lack of lubricant. The cures for these troubles are obvious. I once operated some 400-kilowatt Stanley alternators the rings and brushes of which had regular saw teeth in them due to neglect. Rather than have them turned down in a lathe, I filed the teeth off the brushes, set them tight on the rings and did not use any lubricant. Naturally, the teeth in the rings began to wear away and as I kept the brushes filed smooth and square the rings soon got smooth also, and took on a fine polish when lubricated. The rings in this case were made of copper and lent themselves well to the treatment. Had they been of other material, I might not have been so successful.

Some superintendents are beginning to realize that better results can be obtained if all commutators are placed under the care of one man. In stations where shifts are changed from time to time and different men work on the machines, there is not the same interest taken as there would be by a man who thoroughly understands the work and is responsible for it. The old saying, "Too many cooks spoil the soup" is not out of date yet.

Direct Current Motors and Their Connections

By J. M. Row

A great many times when connecting up or making repairs on a motor, a little familiarity with the elementary principles and operating characteristics of the machines would save engineers lots of time and worry.

In dealing with motors there are three facts which should not be lost sight of; first, if an electric current is passed through a conductor which lies across a magnetic field, the conductor will try to move. Second, if a conductor moves across a magnetic field, "cutting" the lines of force, an electromotive force will be generated in the wire and this will tend to produce a current in such a direction as to oppose the motion of the conductor. Third, if an electric current be passed around a coil of wire having a soft iron core, the core becomes a magnet. Later on the significance of these three principles will appear.

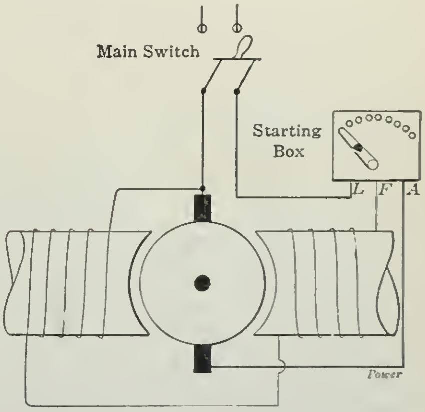


Fig. 1. Shunt-wound Motor

In referring to motors in this article only those which operate on direct-current circuits are meant. They are divided into four general classes: shunt-wound, series-wound, cumulative-compound and differential-compound. The shunt-wound machine is the one most commonly used where practically constant speed is desired. Fig. 1 shows how it is connected up. To put it in service, the main switch is first closed, then the resistance in the starting box is slowly cut out by turning the handle to the right, allowing the motor to come up to speed slowly. It will be noticed that the current divides at the positive brush, part flowing through the field winding and the other part, which is much the greater, passing through the armature. The armature conductors, being located in a magnetic field and carrying an electric current, will move, causing the armature to revolve, in accordance with the first of the principles previously stated. The rotation of the armature, moving the conductors across the field, will generate an electromotive force in them in such a direction as to oppose the flow of current. This is called the counter electromotive force of the machine, and it varies with the speed; that is to say, the greater the speed the greater the counter e.m.f. Also, since the counter e.m.f. opposes the flow of current, the greater it becomes the less will be the current. This makes the motor automatic in its operation, for if the load be increased the armature begins to slow down, thus decreasing the counter e.m.f. and allowing the current to increase sufficiently to pull the increased load. The speed of this motor may be varied between certain limits by inserting "dead" resistance in either the field or armature circuit, or both. The speed will be increased by increasing the resistance of the field circuit, and decreased by increasing the resistance of the armature circuit.*

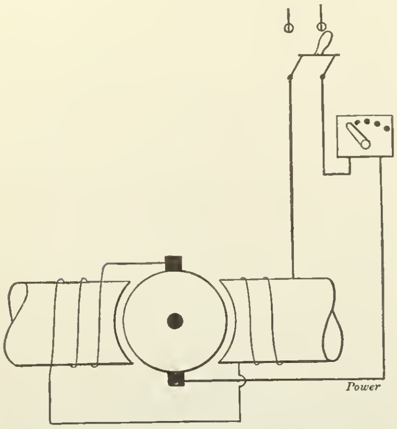


Fig. 2. Series-wound Motor

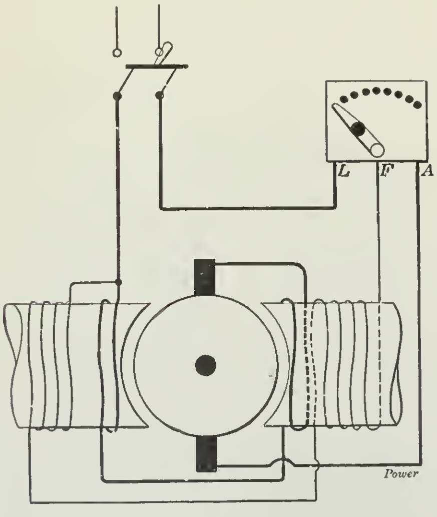


Fig. 3. Compound Motor

The series-wound motor, represented diagrammatically in Fig. 2, is used where the motor is to be repeatedly started and stopped and starts with a heavy load, such as driving freight elevators, street cars, etc. It is well adapted for this kind of work because of its large starting torque. But where constant or nearly constant speed is desired this type of motor is practically useless, for every slight change of load will make a very great change in the speed. Moreover, should the load be thrown off, the motor would immediately race to such a high speed as to completely destroy it. In this respect it acts just like an engine when the governor fails to operate.

It is often necessary to have a motor start with a large torque and at the same time retain the constant speed characteristic of the shunt-wound machine. To meet this condition the ordinary compound-wound, or cumulative-compound motor has been designed. As indicated in Fig. 3, it has two field windings, one in series and one in shunt to the armature; these give it a characteristic combining those of both the series and shunt-wound motors. A great deal more care must be exercised in connecting up this machine. First make sure that the two windings are not interchanged. They can easily be identified by examining the wires leading into the coils; the one composed of the larger wire is the series winding. In connecting up it is best to connect the series field winding first, and test the machine to see if the armature runs in the right direction; if it does not, the connections of the field winding must be reversed. Then connect up the shunt field winding and, with a temporary "jumper" across the terminals of the series windings, test again for direction of rotation, reversing the terminals of the shunt winding this time if the armature runs the wrong way. Remove the jumper and the motor is ready for regular work.

It often happens, usually by carelessness, that in connecting up a compound-wound motor the series winding is connected in opposition to the shunt winding. This makes the machine a differential compound motor. This fault, when present, will make itself known by the motor stopping and reversing its direction of rotation (providing the circuit-breaker does not open) every time the load is increased to a certain amount; or even sometimes while the starting resistance is being cut out.

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*See article on "Varying the Speed of an Electric Motor," by Cecil P. Poole, page 735, November 2, 1909, for fuller information on this point.

Central Station Growth

That the central-station industry is growing rapidly is made evident by a recent special report of the Census Bureau at Washington. The report covers the period between 1902 and 1907, during which the central-station capacity in this country was more than doubled.

In 1902 there were 3620 plants in operation, representing the investment of $504,740,000 and having a total capacity of nearly 2,000,000 horsepower. In 1907 there were 4714 plants, representing the investment of about $1,100,000,000 and having a capacity of slightly more than 4,000,000 horsepower. During the five years, therefore, the capacity increased about 120 per cent. and the total cost about 117 per cent., while the increase in the number of plants was only 30 per cent. The total output in 1907 was approximately 5,862,000,000 kilowatt-hours, and in 1902 it was slightly over 2,507,000,000; this is a gain of about 134 per cent.

These figures serve to illustrate the influence of the continual increase in the size of the units and the accompanying decrease in the space required per unit of capacity. Had the increase in the number of the stations or even the increase in the sizes of the stations kept pace with that of capacity, the increase in cost would in all probability have far exceeded the gain in capacity.

The report divides the central-station employees into two classes, the one composed of salaried officials, clerks, etc., and the other of wage earners. The number of those in the first class increased by 86 per cent., while that of the second group gained only 49 per cent. Here, too, the increase in the size of the unit has had an influence. The average salary rose about 11 per cent. while the average wage rate rose about 4½ per cent.

The number of reciprocating steam engines and steam turbines increased from about 6000 to approximately 7200, or 20 per cent., while their total capacity rose from 1,379,941 to 2,627,450 horsepower, an increase of 90 per cent.

The number of gas-engine units increased from 165 to 463, or 181 per cent., while their total capacity rose from about 12,000 to about 56,000 horsepower, or 343 per cent.

In 1902 there were 1390 water wheels in service, while in 1907 there were 2481. Their capacity increased from about 440,000 to nearly 1,350,000 horsepower, which is a gain of about 208 per cent.

On account of the greater adaptability

of the alternating-current generator to central-station requirements, the number of direct-current generators in use diminished from about 7400 to about 5400, or about 27 per cent., while that of the alternating-current machines rose from 5122 to 6808, or 33 per cent.

The foregoing figures do not represent the entire production of electrical energy. To arrive at the aggregate it would be necessary to consider also the electric railways and the isolated electric-light and power plants, not to mention the relatively small-power telephone and telegraph lines, electric police-patrol and fire-alarm systems.

The report states that in 1907, exclusive of the isolated plants, there were upward of thirty thousand individuals, companies, corporations and municipalities which reported the generation or utilization of electric current in what may be termed "commercial enterprise." These industries represent an outstanding capitalization of over $6,000,000,000, of which about $1,333,000,000 is credited to central stations, $3,667,000,000 to electric railways, $800,000,000 to commercial or mutual telephone companies and about $250,000,000 to telegraph companies, the latter item including nearly $33,000,000 of capital stock of wireless companies.

Although central-station statistics of the comparatively few street railways that sold current and that were able to prepare complete separate reports have been included with those for central electric stations, in order that that branch of the electrical industry might be reported as distinctly as possible, the full measure of the growth of the central station is not made evident by the figures given because of the fact that the sale of current for light and power is also largely carried on by numerous street-railway companies which combine central-station business so completely with other activities as to make complete separate reports impossible.

As would naturally be supposed, central stations are concentrated in the most populous States and at points within these States from which the largest portion of the population can be served economically. New York, Pennsylvania, Illinois and Ohio, the four States having the largest populations, contain nearly 30 per cent. of the total population of the United States. In 1907 these four States reported 1296 electric stations, or 27.5 per cent. of the total number in operation during that year. The annual output of these stations amounted, roughly, to 2,500,000,000 kilowatt-hours during 1907, which is 43.6 per cent. of the total. While the proportionate number of central stations and the proportionate population of this group of States were very nearly the same, and less than one-third of the respective totals, their output was nearly one-half of the total.

The Burke Induction Motor

The Burke induction type of alternating-current motor is a very rugged, substantial machine, as may be inferred from an inspection of the accompanying illustrations. The stator or primary member, shown in Fig. 1, comprises the usual slotted core of laminated steel mounted in a massive cast-iron housing and equipped with form-wound and individually insulated coils. The core laminations are keyed to the housing, in order to avoid the use of bolts through the core, which entail eddy-current losses. The slots in the core are partly closed, unless otherwise ordered, because this construction increases the effective airgap area, thereby decreasing the gap reluctance and therefore the magnetizing current; this improves the efficiency slightly and the power factor appreciably. The construction of the stator housing is obvious from the picture.

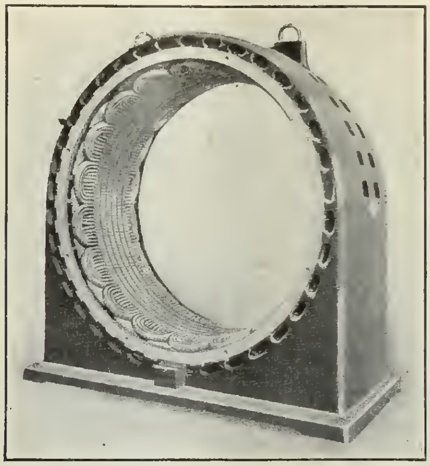


FIG. 1. STATOR OF BURKE MOTOR

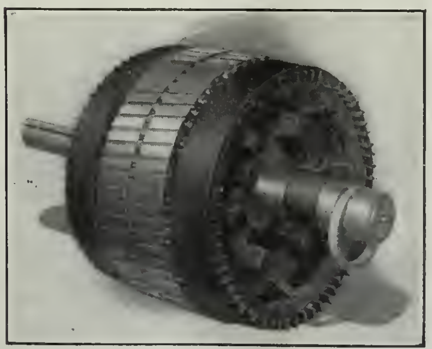


FIG. 2. ROTOR OF BURKE MOTOR

The rotor core disks are assembled with the conductor slots progressively twisted out of line just enough to make the completed slot diagonal, as illustrated in Fig. 2. This reduces the flux pulsations in the airgap, which tends to reduce iron losses and magnetic humming. The rotor conductors of the squirrel-cage type consist of one-piece closed loops punched out of copper sheet in the form shown at the top of Fig. 3 and afterward pulled out into the form shown at the bottom of the same engraving. The two straight "sides" or parallel parts of the loop are separated to a distance approximately equal to the polepitch of the stator, and it is therefore unnecessary to connect the ends of the various coils together; each one forms its own closed circuit. The slip-ring type of rotor is, of course, equipped with a "polar" winding, the terminals of which are connected to collector rings. This form is used for variable-speed service and for conditions requiring high starting torque with moderate starting current.

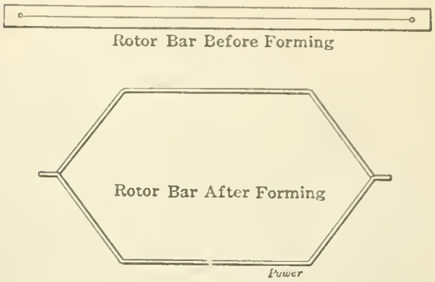


FIG. 3. A ROTOR BAR

The bearings are mounted in the usual circular bonnets or brackets, fitted to the face and bore of the stator housing. The journal sleeves are cast-iron shells with babbitt linings, and the oil rings consist of a number of thin metal disks, as indicated in Fig. 4. This arrangement is very unusual, but it is said to give excellent results. In the bearing from which this picture was taken there were ten of these disks.

Burke motors of 5 horsepower and under are not supplied with starters; those of larger output are regularly supplied with either resistor-type or autotransformer starters. With the latter type a separate oil switch is supplied.

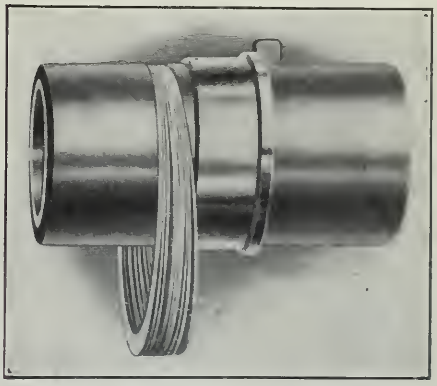


FIG. 4. LAMINATED OIL RING

These machines are built by the Burke Electric Company, Erie, Penn.

Sid Diggles ast Hal ef he knowed why his bilers wuz like a herd uv melk cows. Hal sed he didn't, an' didn't keer a cuss nohow. Sid told 'im 'twas cause they hed so dumd many bags on 'em.

Gas Power Department

Flying Machine Motors

The recent aviation "meet" at Belmont park, near New York City, afforded an excellent opportunity for close inspection of the various types of aëroplanes and their power plants. As our readers doubtless know, an aëroplane consists of huge planes or wings of light material mounted on a framework which also carries a two-blade propeller and a gasolene engine to drive it. While flying machines do not come within the regular field of POWER, the engines that propel them are of interest to all of us by reason of their kinship to stationary and marine engines driven by gasolene.

All of the engines used in the Belmont Park contests operated on the four-stroke cycle, and all were of the multiple-cylinder type with single-acting pistons, following rather closely the general lines of automobile practice. The machine used by the Wright brothers departs from this practice less than any

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

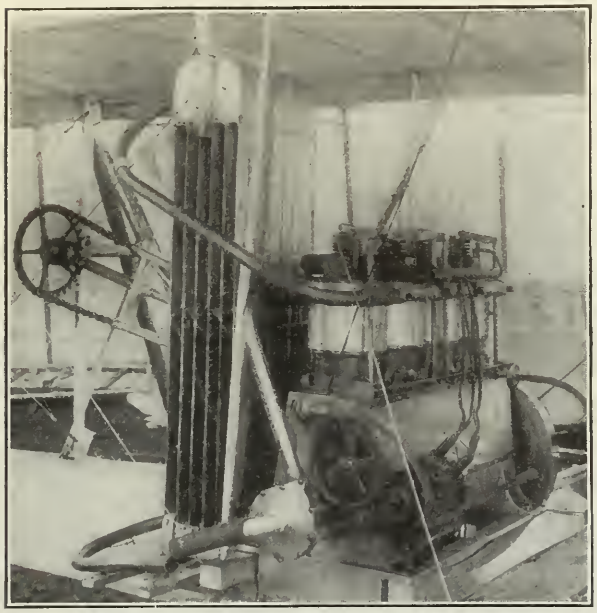


FIG. 1. THE WRIGHT MOTOR

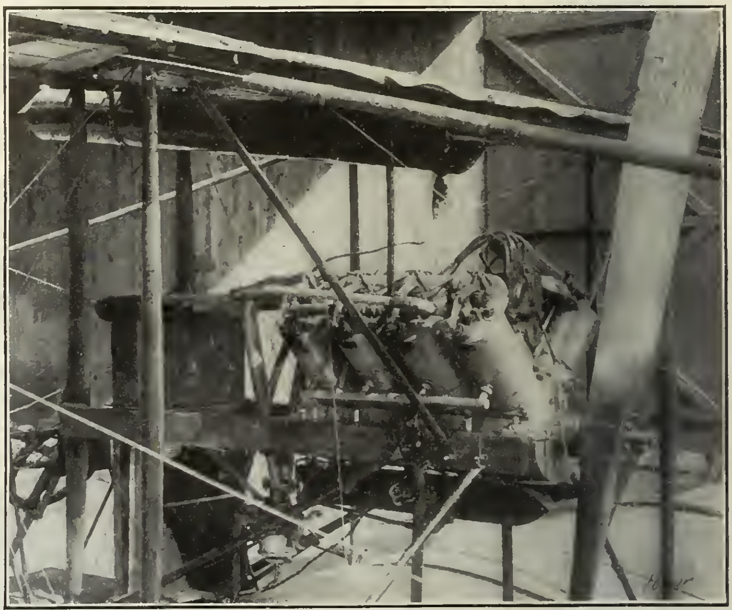


FIG. 2. CURTISS' REGULAR MOTOR

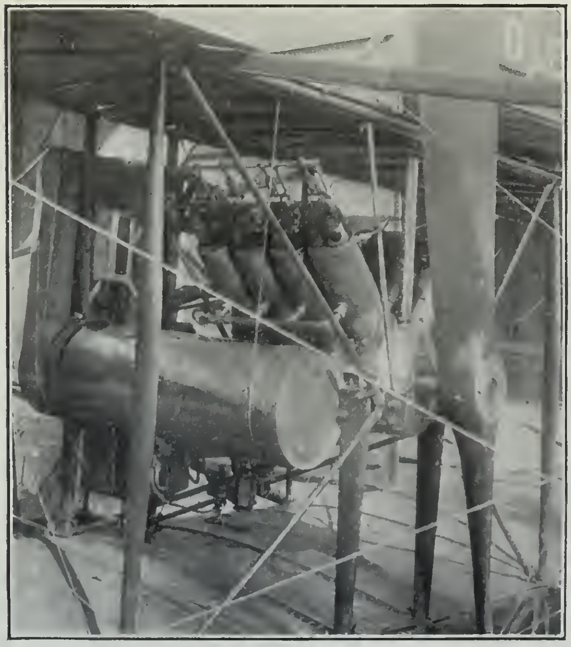


FIG. 3. HAMILTON'S MOTOR

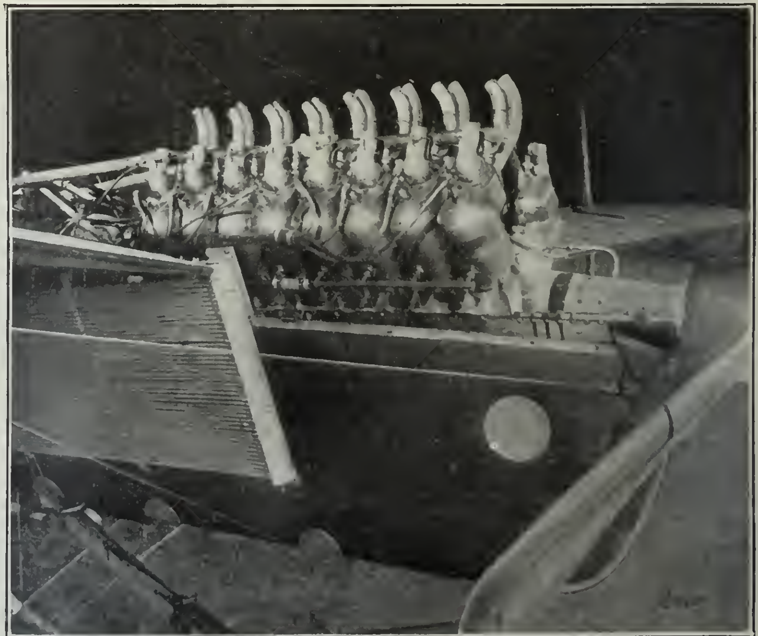


FIG. 4. THE 100-HORSEPOWER ANTOINETTE MOTOR

of the others. This is a four-cylinder engine of the appearance shown in Fig. 1. It embodies no features radically different from the ordinary motor-car engine; its weight has been reduced to the lowest practical point and there are some special details in the arrangement of the ignition magneto; that's about all.

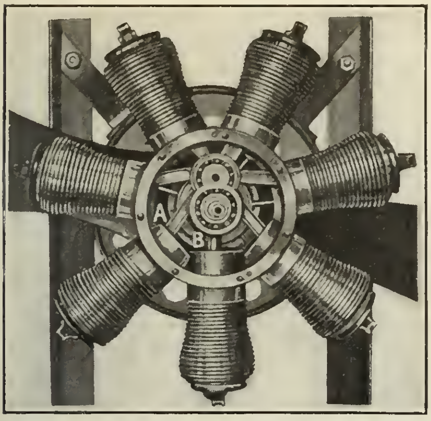


Fig. 5. Gnome Rotating Cylinder Motor

All of the other engines, excepting the Gnome, were of the V type; that is, the cylinders are divided into two rows set at an acute angle with each other, as shown in Fig. 2, which illustrates the Curtiss motor. The valve gear of the Curtiss engine is unusual in that both the inlet and exhaust valves of each cylinder are opened by one rocker arm. The chief object of this arrangement is to reduce the weight. Only one rocker arm and one push rod are needed for each cylinder. The openings in the cylinders near the heads are exhaust outlets; no exhaust pipe is used—another weight-reducing expedient. The long, slender cylinder with pointed ends, suspended above the engine, is the gasolene tank, from which the fuel flows to the carbureter, located below the rear end of the engine. The ignition magneto is located between the two rows of cylinders at the front end of the engine.

The construction of the engine used by Hamilton (Fig. 3) is similar to that used on the regular Curtiss machine. The differences between them are not of fundamental importance.

The Antoinette engine, illustrated by Fig. 4, is also of the V type, but it embodies the radical feature of injecting the gasolene directly into the intake pipe instead of atomizing it and mixing it with air in a carbureter. One result of this arrangement is that it is extremely difficult to start the motor; another is that the speed cannot be regulated by varying the admission of fuel, except to a very limited extent, because decreasing the quantity of gasolene injected makes the mixture in the cylinder too lean for prompt ignition and increasing it makes the mixture too rich. Most of the speed control is effected by changing the timing of the ignition, which overheats the engine at the lower speeds. The short pipe extending upward from each valve chamber is the intake pipe; the fuel nozzle enters the pipe on the side in a direction almost parallel to the axis of the pipe. The longer pipes projecting upward in pairs and curving backward are the exhaust pipes. The spark plugs are located in the centers of the cylinder heads. The manifold extending alongside the body of the craft, from the sixth cylinder rearward, is one of the radiators for cooling the jacket water; the other one is in the corresponding position on the other side of the machine. The tubes and headers are made of aluminum, to reduce weight.

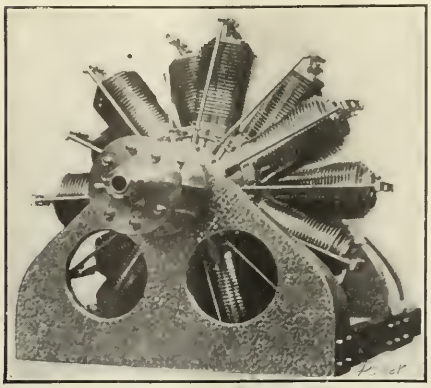


Fig. 6. Double Row Gnome Motor

The Gnome engine differs from all of the others that were at this meeting in two important particulars: the cylinders revolve about a stationary crank shaft and they are air cooled, instead of being water cooled. Fig. 5 is a view of a seven-cylinder Gnome engine with the crank-case cover removed to show the arrangement of the connecting rods and crank. There is only one crank and that is located above the shaft; consequently, the lowermost piston always projects beyond the end of its cylinder, as shown at *B*. The engine is built with single and double ranks of cylinders; the double-rank engine has fourteen cylinders, seven in each rank, as shown in Fig. 6, and two cranks located 180 degrees apart. This type was used by Grahame-White at Belmont park. Fig. 7 is a view of the motor in the Blériot machine, and Fig. 8 is a side view, showing that the cylinders in the two ranks are staggered with respect to each other. This makes the engine shorter, and therefore the weight less, than it would be with the cylinders in line fore-and-aft, and it also allows the air to get to the rear cylinders more effectively.

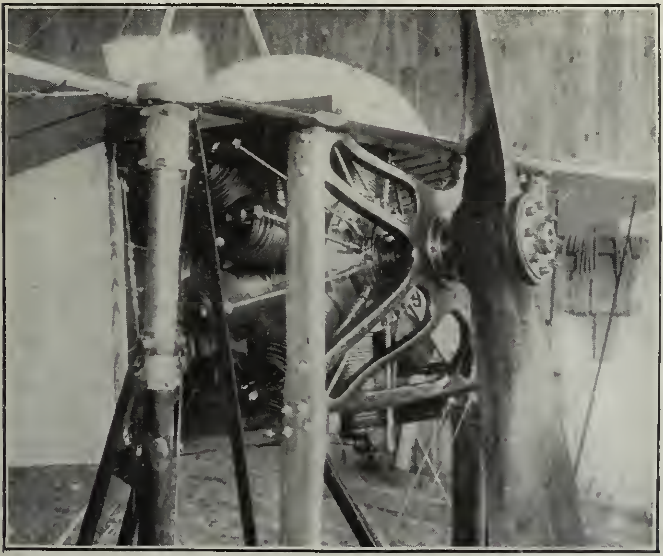


Fig. 7. Grahame-White's 100-horsepower Gnome

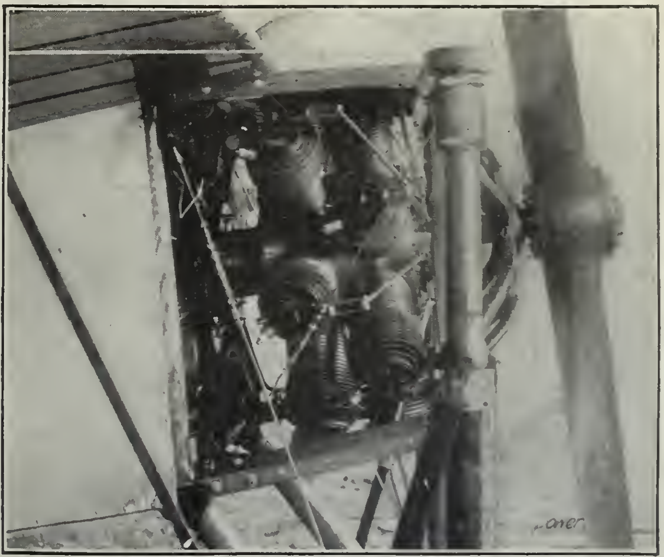


Fig. 8. Side View of 100-horsepower Gnome

The inlet-valve arrangement is unusual, even for a revolving-cylinder motor. It is located in the piston head, and is of the "automatic" type—that is, it is opened by the pressure of the air in the crank case when the piston is pulled outward during the suction stroke. This arrangement makes it easy to deliver the working mixture to the engine; the carbureter delivery pipe merely discharges into the crank case through the stationary shaft, which is bored for that purpose, and the cylinders take their charges from the mixture in the crank case.

The exhaust valve of each cylinder is located in the head and opened by a radial push rod, as shown in Fig. 8. The push rods are moved successively by a cam on a short shaft inside the crank case, this shaft being turned by a gear which is driven at half speed by a gear mounted on the stationary shaft. No exhaust piping is used; the burned gases are discharged directly into the atmosphere from the outlets in the cylinder heads.

If the reader will first imagine the cylinders to be stationary and the crank shaft to revolve, the mechanical relations of the various parts will perhaps be clearer. Then remember that the cylinders and crank case revolve around the crank shaft instead of the shaft revolving within the crank case, and it will be obvious that the mechanical reactions and relative motions are exactly the same in the actual engine as they would be in the imaginary one.

A Puzzling Case of Premature Ignition

By S. Kirlin

In a certain power plant the load is carried on one unit, consisting of a tandem double-acting gas engine operating on natural gas. A great deal of difficulty was experienced at times due to premature ignition. The engine is rated at 300 horsepower; during the day time the load carried is only about 100 horsepower, but in the evening when the lighting load begins to come on the load rises to about the full capacity of the engine. As long as the engine was carrying the light load no trouble was apparent, but as soon as the load began to increase premature ignition would take place so frequently for about half an hour during the time that the heavier load was coming on that the engine would slow down to such an extent that the lights were very unsatisfactory; after about this length of time the trouble would gradually cease and would not occur again until the same time the next evening.

All of the valves and igniters were in perfect condition and the engineer was at his wits' end to find the cause of the trouble. One evening a short time before the load began to rise, some of the admission valves showed signs of sticking, and in order to clean out the deposit around the stems which was supposed to be the cause of the trouble, the engineer poured a small quantity of kerosene into the air inlet, allowing it to be drawn into the cylinders through the admission valves. The engine made several premature ignitions during the next few minutes but as the load was light they did not interfere with the speed, and the valves soon began working freely again. When the heavy load came on a few minutes later the engine gave no trouble, but picked up the full load and carried it through the run without any premature ignition whatever.

This gave the engineer a clue to the cause of the trouble, and each evening thereafter a small quantity of kerosene was poured into the air intake a few minutes before the heavier load was due to come on, and no more preignition trouble was ever experienced. The explanation seems to be that during the light-load period the temperature of the cylinders was not high enough to burn the deposits of carbonized oil which gradually collected on the heads of the cylinders and pistons, but when the heavier load came on the temperature rose to a point which ignited the deposits and produced premature explosions until all of the carbon was burned out of the cylinders. By admitting the kerosene, the carbon deposits were loosened and consumed in a few minutes instead of burning gradually as the load increased.

CORRESPONDENCE

Corrosion of Exhaust Pipe Drips

M. W. Utz in the November 8 issue tells of some troubles he had with water-cooled exhaust pipes and their drain pipes. I have had much the same experience with the exhaust-pipe drains of Diesel engines, and I think the exhaust gases are of similar composition in both cases. In one plant it was our custom to run as little water into the exhaust as possible, or just enough to keep the fuel valve casing cool, as the water from there was piped into the exhaust. Ordinary 1-inch black pipe would last about three weeks when running 24 hours per day and the nipples even a shorter period. I also tried galvanized pipe but the water could attack this where the threads were cut and, although it was longer-lived than black pipe, it did not solve the difficulty. I then tried extra-heavy iron pipe which endured for about three months. The exhaust pipes were of cast iron and gave no trouble.

Upon taking charge of another plant I found it had been the practice with the former engineers to run a much larger quantity of water into the exhausts than I had, which kept them much cooler and increased the life of the drain pipes far beyond anything I had experienced. In this plant there was one engine equipped with brass drains which ran for over a year without showing signs of failure.

My theory as to the cause of the corrosion is that the gases combine in a lesser or greater extent with the water to form a mixture similar to sulphuric acid, which contains two parts of hydrogen, four of oxygen and one of sulphur, as is shown by the formula, $H_2 SO_4$. The one part of sulphur would be easy to obtain, as would the two parts of hydrogen, although there might be a scarcity of oxygen but not enough to make much difference. This acid would attack wrought iron very readily, but increasing the supply of water, since it is composed of two parts of hydrogen and one of oxygen, would change the nature of the combination and lessen the strength of the acid. This is further borne out by the action of the cooling water on any polished steel surface, which immediately becomes dull and blackened.

If I were in Mr. Utz's place I would have some of the water analyzed and find just what is formed. In any event, should the expense be permissible, I think lead-lined pipe such as is used for the conveying of acids would do for drain pipes and cast iron for exhaust piping.

G. H. Kimball.

East Dedham, Mass.

Cracked Flywheel Hubs

In the issue of Power dated September 20, there was printed on page 1692 a letter by Frank E. Booth, entitled "Flywheels Broken by Preignition." It has been my experience that the breakage of flywheel hubs, in the location shown in the cut, is not caused by preignition but by improper methods in putting the flywheel on the shaft. The taper key is driven into the keyway when the bolts in the hub are loose, causing one-half of the flywheel hub to ride on the key instead of on the shaft. The bolts are then pulled down, throwing a tremendous strain on that section of the hub which Mr. Booth's drawing shows broken. This method of securing the flywheel will almost invariably crack the hub. The hub may not necessarily break at the time the bolts are pulled down, but the strain can be thrown into the hub of the wheel so that this breakage occurs at some later time. The proper method is, of course, to pull down the bolts perfectly snug on the shaft before the key is driven home.

A. Y. Edwards,

Chief mechanical engineer, Foos Gas Engine Company.

Springfield, O.

Quite recently, within the state of Pernambuco, Brazil, was discovered what promises to be a most valuable coalfield. The area embraces about 22 square leagues.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Power Plant Design and the Operating Engineer

Every day one sees mistakes in the layout of power plants, caused by the designer being thick headed, or perchance it may have been a consulting engineer or the chief engineer in charge of operation or it may have been the fault of the superintendent or the ideas of the president of the company.

I believe, emphatically, that in the majority of cases blunders in power-plant design are due more to self-inflation than any other cause.

Operating engineers frequently say, "The engineer in charge is best qualified to pass on contemplated improvements"; again, the consulting engineers laugh up their sleeves over mistakes made by operating engineers, which mistakes were due solely to ignorance of the laws of philosophy, simple laws which every man should know thoroughly before attempting to design or be a critic in laying out a power plant. How can the engineer in charge secure uninterrupted and satisfactory service while watching the hundred and one things around a plant, continually looking where improvements can be made and at the same time be a designing engineer? Is not the old adage, "He is a jack of all trades and master of none," quite true? This is a day of specialists, and any man who desires to aspire to any degree of success must specialize along one particular line of work.

I do not wish to infer that the operating engineer should not be consulted when changes are to be made, but that others who have made power-plant designing a study should be consulted, thus securing the cardinal ideas of both. The consulting engineer is not expected to come in offhand and operate a power plant with any degree of satisfaction; then why should the operating engineer expect to be eligible for designing? To be a success along any particular line a person must devote his efforts without reserve along that line.

The secret of success in any concern at the present time is organization and system; this can be obtained only by securing men of ability for each individual line of work, such that every phase of the organization may secure faultless results. This factor of strong organization and system is responsible in a greater degree than any other thing for the monstrous success of the central stations forcing so many isolated plants to close down. The only course left for the isolated plants to pursue in order that they may withstand the onslaught of the central station is to be under the supervision of an able consulting engineer, thoroughly uptodate in operation and design, but not just an operating engineer, by any means.

If the operating engineer would make himself eligible as a designer, he must aspire to a higher plane of education and social standing, such as is gained through constant study and observance, to which should be added the constant association with men of brains and achievements.

WALDO WEAVER.

Middletown, O.

Gas in the Pump

I had a duplex pump which was used to pump water from the river through a lift of about 15 feet, and which gave entire satisfaction in this capacity. However, as the river water was very poor, we had two wells driven and connected the suction of the pump to these wells, with no misgivings, as the lift was only about 8 feet.

The pump worked all right for a few minutes and then started to buck, as if it were drawing air. I measured the hight of the water in the wells, and finding the lift to be less than 15 feet, I took a candle and carefully looked over the suction line, trying every joint, but did not succeed in locating a leak. I then looked over the valves and packed the water end, but everything seemed to be in good order; still it continued to buck when run at normal speed, which was considerably below 100 feet per minute.

The pump was handling water all the time, but not enough to supply us, and every attempt to make it do so resulted in failure.

That evening on passing one of the filter tanks, I noticed that the pressure was above normal, and proceeded to cut it out and flush the mud into the sewer. I had just started, when there was a slight explosion, filling the cellar with flames. In less than a minute it was all over and I went back, although there was still a flame about two feet long coming from the flush pipe of the filter. This was in an oil section, and the water wells were in the vicinity of many old oil wells, which accounted for the trouble with the pump and the explosion. It developed that the pump had been handling water and natural gas mixed, the filters had formed a pocket for the gas, and when I liberated it in flushing them, a gas jet on the wall ignited it, and the explosion occurred.

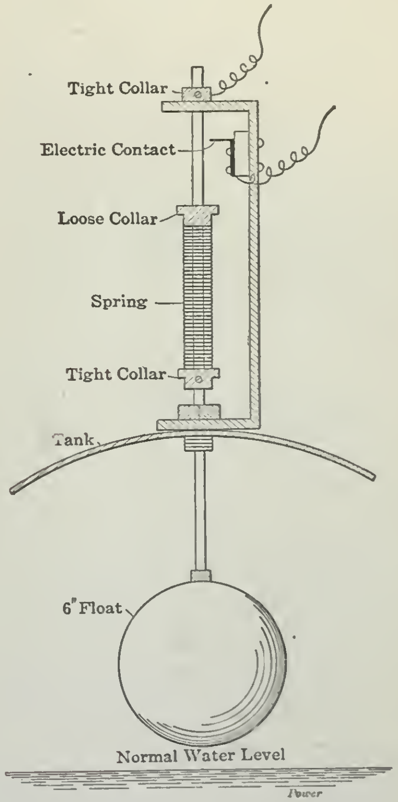
I put a pet cock on each one of the filters, and lighted that part of the cellar from them, but I am still unable to pump the water through a lift greater than 15 feet.

A. STEVENS.

Renfrew, Penn.

High Water Alarm

In the boiler room of a large industrial plant were located several vented return tanks into which the condensation resulting from the manufacturing processes and from the heating system was returned. Due to the unsteadiness with which the water returned and the fact that the pump room was not under constant supervision, the tanks occasionally filled up, the water backing into the return system with serious results. Therefore, a ¾-inch hole was drilled and tapped in the top of one of the tanks and a brass bushing with a hexagon head and



FLOAT AND ELECTRICAL CONNECTIONS

a 5/16-inch hole was fitted to the tank. A piece of ⅜x1½-inch flat iron 18 inches long was bent, as shown, and at one end was drilled a ¾-inch hole, by means of which it was secured to the tank under the head of the bushing, and at the other end was drilled a 5/16-inch hole alining with the one in the bushing. A 6-inch float attached to a 5/16-inch brass rod was provided and held in place by a small collar at its upper end, this being adjusted so that when the water was at its normal level the float was above it. A brass collar with a wide flange was screwed to the lower part of the rod and a similar collar was fitted to slide freely on the rod with a light coiled-brass spring between the collars. At the side of the frame was fastened a pair of contacts which were attached in such a position as to close a bell circuit when the sliding collar came in contact with them. As the level of the water rises the float lifts the loose collar against the contacts and rings the bell; further upward movement of the water compresses the spring and leaves the bell in operation. The apparatus as described has answered the purpose admirably and can be made with material which can be readily picked up around most any plant.

LEWIS C. REYNOLDS.

Willard, N. Y.

A Convenient Flue Cutter

The accompanying illustrations show a device for cutting out flues from a locomotive type of boiler. The tool holder is first inserted in the flue, the cutting tool is placed in the holder and the point is driven through the tube. The socket *A* is then attached to the tool holder at *A*, the 1⅛-inch steel rod with a square on one end to fit the spindle *Y*, and an eye on the other end which is welded into socket *A* at *B*. This rod drives the cutter to be inserted into any flue without changing the device. One revolution cuts off the flue.

Fig. 1 shows the assembled device and the method of hanging it up with a block and fall. The two chains shown on the lower part of the device are very important. These are anchored to each frame to steady the device and to keep it from swinging around when cutting off a flue. The large gear wheel is mounted on the spindle *Y*, which drives the cutter, and the power of the motor is transmitted through the gears *V* and *W*. The ease with which this device can be adjusted from one flue to another is a feature that should recommend it to anyone.

W. H. SNYDER.

Stroudsburg, Penn.

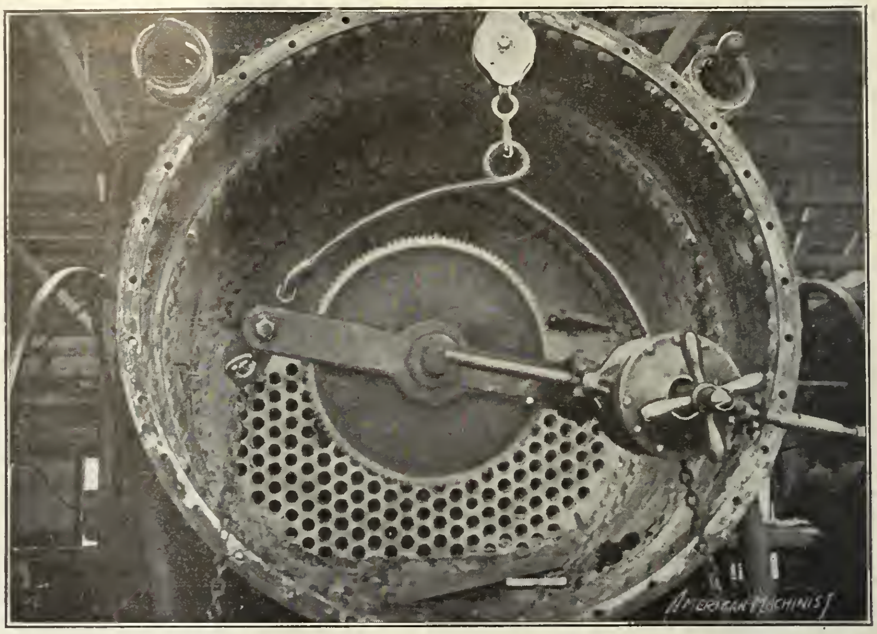


FIG. 1. FLUE CUTTER IN POSITION

Color Scheme for Piping

Any color scheme for general adoption must be simple in form and broad in scope. Both of these essentials are entirely neglected by the majority of proposals for color schemes. The trouble lies in the consideration of details to the neglect of fundamentals, resulting in schemes not only devoid of method, but much too cumbersome for general utility.

A standard color scheme for piping must be capable of consistent application to any and every plant. Plants are not

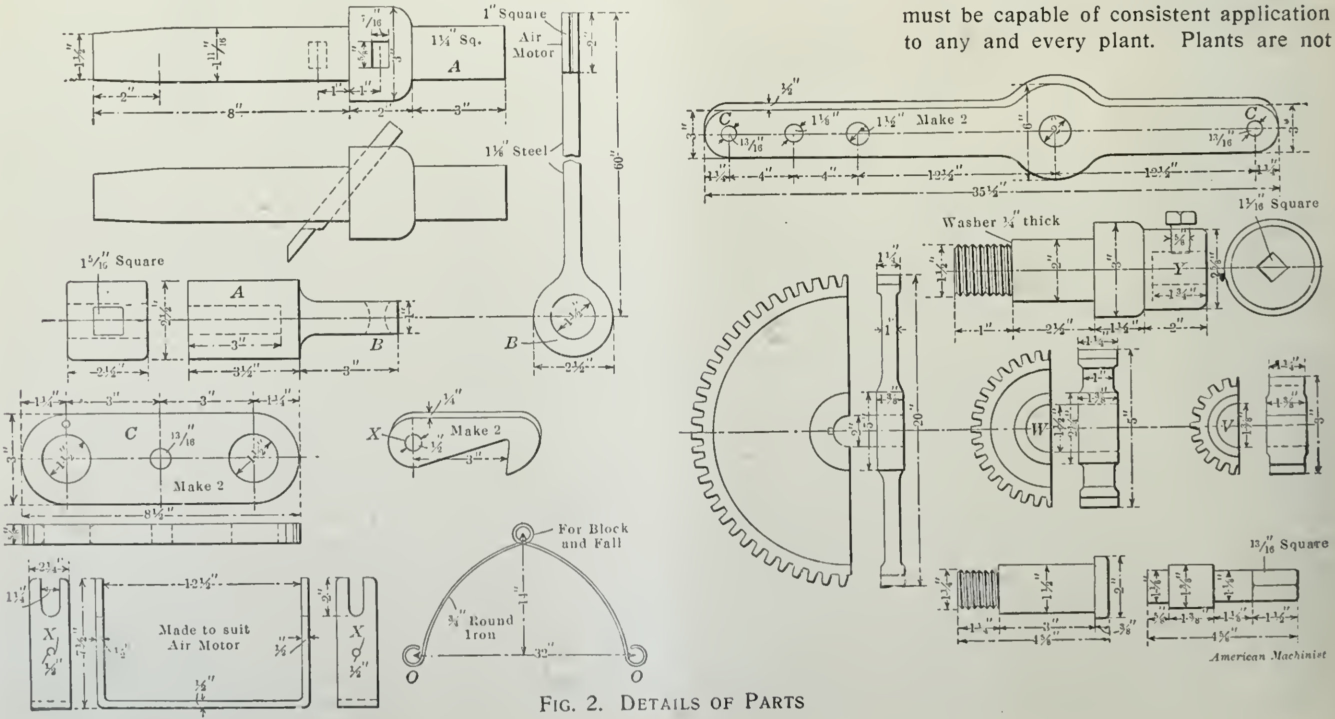


FIG. 2. DETAILS OF PARTS

all steam nor power plants and by no means should industrial plants be neglected. Industrial plants are extensive and complicated and in far greater need of some system of pipe identification than are the majority of power plants. The list of materials conveyed by piping in the various industrial plants would exhaust any possible color scheme, unaided by Roman letters and Arabic numerals. Therefore, little thought should convince one that in such an extensive system, colors can only aid and not supplant language.

Materials conveyed through pipes are in one of two physical conditions: either liquid or gaseous. Solids are occasionally conveyed but they are usually held in suspension by either a gas or a liquid. The liquid or gaseous state is either stable or unstable within the range of the temperature and pressure employed at the plant. The terms stable and unstable are used here not in an absolute sense but in a relative sense, the condition within the pipe to that in the atmosphere, subject to the change that would occur at the bursting of the pipe or opening of a valve. Even a gas is not absolutely stable, though within ordinary working pressure and temperature it is decidedly so. A vapor, though not a gas, is in a gaseous state and is unstable. A liquid that would remain a liquid at the bursting of a pipe would here be considered a stable liquid, while a liquid that would flash into vapor at the release of pressure would be unstable. This gives the four principal conditions of materials carried by piping and suggests a color scheme of the following character:

Gas	Black
Vapor	White
Stable liquid	Olive
Unstable liquid	Maroon

Some method of distinguishing relatively high and low pressures can be added by representing generator outlets and motor inlets by plain-colored valves and fittings, and by coloring yellow the valves and fittings of generator inlets and motor outlets. The term generator is used here to designate any apparatus doing work upon the fluid and boosting the pressure, and the term motor as any apparatus operated by the fluid and through which the pressure drops.

This color outline is adaptable to any plant. Any addition would but limit its scope. The further working out of details should be left to the managements of individual plants and this can be carried out either with colors or preferably with stencils. The stencil requires no more maintenance than does the plain color and is certainly simpler and more comprehensive. A stencil can show the material, name, pressure, temperature and direction of flow if advisable. It is frequently found necessary to post directions and cautions at important valves.

On lines that are difficult or impossible to trace, some identification should be permanently stamped on the valve body or upon a metal tag secured to the valve or line.

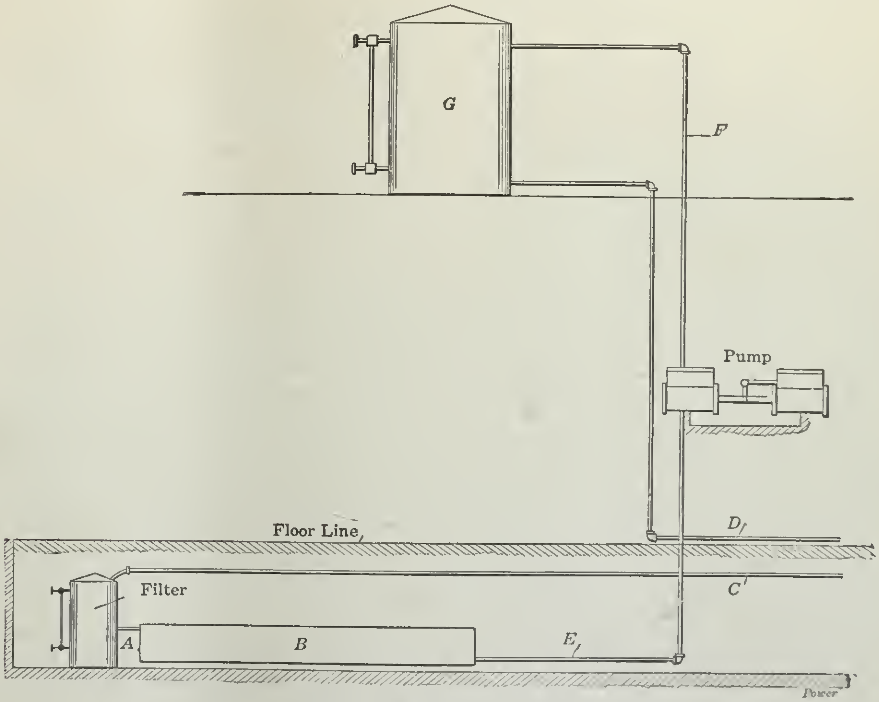
In addition to the foregoing methods of identification there should be an accurate diagram of the piping layout. However, with most industrial plants, especially with the older ones, the drafting of such a diagram would be next to impossible. The necessity of an accurate and convenient identification of piping cannot be overestimated and the standardizing of this identification as far as possible would be welcome.

J. W. TAYLOR.

Massillon, O.

A Homemade Oiling System

I have never been able to understand why some engineers will slop around day after day filling oil cups by means of a squirt can, when in most plants an automatic oiling system can be installed with little or no expense. In most plants there is enough scrap of various kinds lying around with which to install the system, and there is usually some man on the job who can do the work during his spare time.



LAYOUT OF OILING SYSTEM

The illustration shows the system which was installed in a plant of which the writer had charge. There were two 14x14-inch high-speed engines, each equipped with a large oil pot, the oil being piped to the various bearings from this pot. After lubricating the bearings the oil ran down into the base of the engine, from which it had to be drained out into a bucket, poured into the filter, drained through the filter into another bucket, and again poured into the oil pot. This arrangement not only required considerable time and careful watching, but was sloppy, wasteful and unsightly. With the new arrangement, however, the oil instead of being caught in a bucket was led through pipe *C* to the filter, which was placed in a pit under the engine-room floor; after being filtered, it drained through the pipe *A* into the storage tank *B*. It was then pumped through the pipes *E* and *F* to the storage tank *G*, from which it was led back by pipe *D* to the oil pots on the engines. These oil pots were fitted with tight covers so that the pressure due to the head of tank *G* was on the needle valves, which regulated the feed to the bearings. A stream of oil was kept running over all the bearings and much better lubrication was secured with less trouble and cost. It was necessary only to run the pump about five minutes to fill the tank *G*, and after being filled it lasted from six to eight hours, the oil running over the bearings and returning through the filter into the storage tank *B*.

Nearly all the material used was taken from the scrap pile, tank *G* and a few pipe fittings being all that had to be purchased. Tank *B* was made from a piece of ventilating duct, picked from the scrap pile. If there is no pump about the plant that can be used for the purpose, a small plunger pump can be made easily from pipe fittings and be attached to the end of the engine shaft, rocker arm or valve stem. Even though the whole apparatus had to be purchased, it would soon pay for itself in the saving of oil. The first month the system was in operation, only seven gallons of oil were used on both engines.

R. L. RAYBURN.

Kansas City, Mo.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

The Theory of the Cooling Tower

Joseph H. Hart's "Theory of the Cooling Tower" as expounded in the October 4 issue contains some statements which, if true, are of great importance to all owners and operators of cooling towers. The most important of these statements and at the same time the one which I believe is based more upon theory than upon actual tests is that in which he recommends the warming of the air before its entrance into the cooling tower. His claim is that by first passing it through the boiler or engine-room loft before blowing it into the cooling tower the evaporation is increased and the efficiency of the tower is improved. He reasons that since more water is evaporated, more heat becomes latent and the water is cooled to a lower temperature. Here is where I believe he is in error. What really occurs when the temperature of the air is raised, is that the heat which becomes latent owing to the increased evaporation comes largely from the air, and the final result is that the water will not be cooled to as low a temperature as it would be, had the air been left in its natural state.

That this is true can probably best be shown by an example. Assume the temperature of the air to be 60 degrees, the barometer 30 inches and the psychrometer or hygrometer readings to be, dry bulb, 60 degrees, and wet bulb, 55 degrees. According to the psychrometric tables as published and used by the United States Weather Bureau, the temperature of the dew point would then be 51 degrees, and the absolute humidity, 4.222 grains of water vapor per cubic foot. Now, suppose that this air were warmed in some way to a temperature of 96 degrees. According to the experiments of Regnault, Herwig and others, the coefficient of expansion of a mixture of air and water vapor is very nearly the same as for dry air. Then, one cubic foot of the air at 60 degrees would expand to

$$1 + [(96 - 60) \times 0.00203252] = 1.073$$

cubic feet at 96 degrees and, each cubic foot at 96 degrees would contain

$$\frac{4.222}{1.073} = 3.935 \textit{ grains}$$

of water vapor. According to the psychrometric tables, 3.935 grains per cubic foot corresponds to the dew point of 49 degrees, nearly, and when the air temperature is 96 degrees and the dew point 49 degrees, the difference in temperature between the dry- and the wet-bulb thermometers is 28.7 degrees, or the wet-bulb thermometer shows a temperature of

$$96 - 28.7 = 67.3 \textit{ degrees.}$$

This is the temperature of evaporation and the lowest temperature to which water can be cooled by passing air over it whose temperature is 96 degrees, and whose humidity is 3.935 grains of water vapor per cubic foot. But, the temperature of the wet bulb or the temperature of evaporation was 55 degrees before the air was warmed. So, it will be seen that by raising the temperature of the air from 60 to 96 degrees, the moisture per pound of air remaining the same, the temperature of evaporation is raised from 55 to 67.3 degrees or the possible cooling effect lessened by

$$67.3 - 55 = 12.3 \textit{ degrees.}$$

Of course, in an actual cooling tower the water is never cooled to as low a point as the temperature of evaporation.

Now, by means of a somewhat lengthy calculation it can be shown that this loss in cooling effect on the water, in spite of the increased evaporation, is caused by the fact that the air must also be cooled. Mr. Hart seems to think that he can avoid this cooling of the air by speeding up the fan and by shortening the path of the air through the tower. It is true that this may not give the air time to cool, but neither will there be time for it to become saturated and, if the heating effect by conduction of the air on the water is lessened, the cooling effect due to evaporation is also lessened, and to the same extent. This can be shown to be true by a very simple experiment. Take a sling psychrometer and whirl it with just sufficient rapidity to obtain an accurate reading and then whirl it very rapidly. It will be noticed that the temperature of the wet bulb is not materially changed. In fact, owing to the increased friction of the air it may show a slight rise. The only thing to be gained by increasing the velocity of the air through the tower is a nearer approach of the temperature of the water leaving the tower to the temperature of evaporation, though the former will never equal the latter as long as the water is flowing.

Mr. Hart also makes the statement that after the air has risen the first 2 to 4 feet in the tower and has become saturated, it then does its cooling solely by conduction, convection and radiation. This is not strictly true because, as the air rises in the tower, it comes in contact with warmer and warmer water, cools the water by conduction and itself becomes warmer; the vapor it carries is then no longer saturated and more water is evaporated. In fact, in the upper part of the tower the cooling effect due to evaporation greatly outweighs that due to conduction.

Mr. Hart uses the unscientific expression "the absorptive power of air for water vapor." If air has an absorptive power for water vapor it is so small that no experimenter has been able to determine it. When water exposed to the air evaporates, it does not evaporate because of the air, but in spite of it. Air is always a hindrance to evaporation because the water vapor must diffuse through the air as fast as it is formed and unless the air is in motion this is a slow process.

A. E. MUELLER.

Mayville, Wis.

Right Angled Triangular Weir

In the article on "The Right Angled Triangular Weir," by Charles N. Cross, which appeared in the September 20 number, the author, in framing a formula for weirs of this kind, assumes

h = *the head of water.*

The illustration and description show this to be his assumed head of water above the lowest part of the notch. And, having a right-angled triangular weir, h^2 = the area of the notch filled up to the level of the assumed greatest head acting on the weir, and equal to the area of cross-section of the theoretical discharge.

With these conditions he concludes that because the theoretical velocity at the lowest part of the notch would be $\sqrt{2gh}$, "therefore, the theoretical discharge is,

$$q = h^2 \sqrt{2gh}\text{"}$$

i.e., he concludes that "the theoretical discharge" of a weir is to be taken as equal to the assumed cross-sectional area of the discharge, multiplied by the theoretical velocity due to the greatest head acting on any part of the cross-sectional area.

Theoretical discharge, like actual discharge, is equal to the product obtained from multiplying the assumed cross-sectional area by the assumed mean velocity of water traversing the cross-section. Hence, in order for the formula to be consistent for representing the theoretical discharge of a weir wherein h^2 represents the cross-sectional area, the other factor, $\sqrt{2gh}$, would have to represent the mean velocity. But, when h is the total head, the assumption of $\sqrt{2gh}$ as the value of the factor representing the mean theoretical velocity is erroneous, whether used with reference to triangular or any other form of weir, because the heads which create velocity at different parts of the cross-section have values which vary between h and zero. The velocity at different parts of the cross-sectional area must, therefore, vary between $\sqrt{2gh}$ and zero, and the mean velocity can be only a fractional part of $\sqrt{2gh}$.

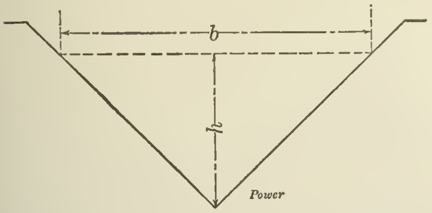


FIG. 1. RIGHT-ANGLED TRIANGULAR WEIR

The theoretic discharge of a notch, whether triangular, semi-circular, or other form of open-sided figure, may be assumed as made up of a number of narrow rectangular notches of uniform width placed side by side and filling the cross-sectional area under consideration. Referring to Fig. 1, which is like the right-angled triangular weir illustrated in the article of Mr. Cross, wherein h is the depth and

$$b = 2h$$

is the breadth of the notch filled, the cross-sectional area may be supposed to be composed of narrow rectangular weirs of uniform width placed side by side as illustrated in Fig. 2.

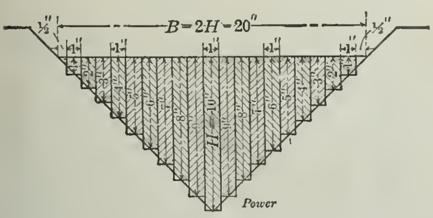


FIG. 2. DIAGRAM OF WEIR DIVIDED INTO A NUMBER OF NARROW RECTANGLES

The discrepancy between the total cross-sectional areas due to stepping downward the narrow weirs to different depths need be very little and resulting discrepancies due to head may be almost entirely eliminated, though the width of each section is taken as a considerable proportionate part of the width of the notch. For a right-angled triangular weir, if the width of sections is taken as equal to 1/20 of the total width b, Fig. 1, as indicated in Fig. 2, then the summation of discharges of the different sections will be found to be within a small fraction of 1 per cent. of the discharge of the notch computed by the refined methods of calculus.

The theoretical discharge of a rectangular notch or weir wherein

L = Width in feet, and
H = Total head in feet

is given in cubic feet per second by the formula:

$$Q = LH\tfrac{2}{3}\sqrt{2gh} \qquad (1)^*$$

In Fig. 2, assuming the width of each section to be $L = 1$, the total depth of the notch as 10 and the total width as 20, then we would have a central section in which $L = 1$ and $H = 10$, and nine weirs each side of the central section having $L = 1$ and values of H from 9 to 1, and two small triangles at the extreme right and left sides which may be neglected.

The sum total of discharges of these weir sections, computed from equation (1), will be found to equal 1354.79 + cubic feet per second. Computed by the equation

$$q = h^2\sqrt{2gh}$$

in which h, for a right-angled triangular weir, is taken to be 10, we would obtain a theoretical discharge of 2532.76 + cubic feet per second, from which we see that the quantity obtained by the summation of rectangular weirs is about 53.45 per cent. of the quantity computed by the last named equation, which is the formula for the theoretical discharge given in the article referred to.

A similar result is obtainable by dividing the triangular weir into horizontal sections and taking the sum of their discharges, considering each section to discharge with a velocity due to the head above it. A precise formula for the theoretic discharge of a triangular weir which is founded on the summation of discharges of horizontal sections reduced by calculus gives

$$Q = \tfrac{4}{15} B H \sqrt{2gH} \qquad (2)$$

wherein

Q = Discharge in cubic feet per second;
B = Breadth of notch in feet;
H = Total head (or "depth of notch filled") in feet, and
g = Acceleration due to gravity (= 32.16).

For a rectangular triangular weir, in which $B = 2H$, the formula becomes

$$Q = \tfrac{4}{15} \times 2H \times H\sqrt{2gH} = \tfrac{8}{15} H^2\sqrt{2gH} \qquad (3)$$

or representing the coefficient for actual discharge by K we have the usual formula for the actual discharge of a 90-degree notch, viz.,

$$Q = \tfrac{8}{15} K H^2\sqrt{2gH} \qquad (3a)$$

or its equivalent,

$$Q = \tfrac{8}{15} K H^{\frac{5}{2}}\sqrt{2g}$$

It is to be observed that the fraction 8/15 in the second member of (3) and in the formulas following indicates that the theoretical discharge = 53⅓ per cent. of $H^2\sqrt{2gH}$. The discharge obtained by computing vertical sections as illustrated by Fig. 2 being 53.45 per cent. of the discharge computed as $q = h^2\sqrt{2gh}$, H in one case being the same as h in the other, it is of interest to note that the result obtained by dividing the notch into narrow rectangular weirs, and taking the sum of their theoretic discharges, gives a discharge which is within 0.03 of one per cent. of the theoretic discharge obtained by the employment of formula (3), which is based upon the most refined methods of mathematical analysis.

FRANKLIN VAN WINKLE.

Paterson, N J.

Indicator Reducing Motion

Having had some experience with the indicator reducing motion described in the issue of October 11, I write to point out some of the disadvantages of it which led to its being replaced by the well known reciprocating lever type.

The attractive features of the reciprocating wheel type of reducing motion are its neatness, its correctness and the fact that it can be set in motion at any time, there being no need to stop the engine to attach the gear to the crosshead.

It was proposed to adopt the gear as the standard for a line of horizontal engines which would be run at any speed up to 180 revolutions per minute. It was recognized from the first that there might perhaps be some risk of trouble due to the effect of the inertia of the wheels in prolonging the travel of the indicator drum when the crosshead came to the end of its stroke, and this was guarded against by having aluminum wheels as light as possible.

When this gear was tried, it was found that with an engine running at about 120 revolutions per minute, there were unmistakable signs of this inertia effect shown on the indicator diagrams, and at speeds above 120 revolutions per minute the gear was quite unreliable from this cause.

It was seen that when moving in one direction the indicator-drum spring added its pull to the inertia of the wheel and that when moving in the other direction the drum spring acted against the inertia of the wheel. Various refinements were tried to make this gear satisfactory, but they were unsuccessful and the gear was abandoned in favor of the reciprocating-lever type, which has the advantage of being cheaper.

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*For demonstration see POWER AND THE ENGINEER, October 5, 1909, page 567.

In the reducing motion supplied by indicator makers and attached to the bottom of the indicator, there is no inertia effect of the wheels to affect the length of the diagram, as during the outward stroke of the crosshead the indicator cord winds up a spring and this spring overcomes the inertia of the wheels, while on the inward stroke of the crosshead the cord is held taut by the tension of the spring and by the nonstretching of the cord there is no overrun of the indicator drum when the crosshead reaches the end of its stroke.

James Cannell.

Stanford le Hope, Eng.

Leakage through a Piston Valve

In the October 11 issue appeared an article under the above title. That leakage occurs past any valve, whether piston, flat, Corliss or other type, is, of course, recognized, and for years the efforts of designers and builders have been constantly put forth to reduce this loss to a minimum, while at the same time considering the other factors that enter into the building of a successful engine.

To measure accurately the amount of leakage past the valve of an engine which is in motion is a difficult problem, and it is doubtful if the results as determined by Mr. Mitchell's laboratory method can be relied upon as properly representing actual performance under normal conditions. Even assuming the method employed in the tests reported as representing actual working conditions, as nearly as such can be approximated, what reason is there for assuming that similar results would not be obtained if this same method were applied to a flat valve, by making one end of the valve wide enough to lap the port, entirely preventing its opening, or to a Corliss valve by special construction?

This method of testing the valve by allowing it to move over an unported surface and measuring the water escaping by the "blind end," is in effect duplicating the condition which exists in certain forms of single-acting engines. The steam is on one side of the piston only, and with the constant changing of temperature, due to the movement of the piston, there is of necessity leakage of water of condensation no matter how well the piston is fitted. Much depends, of course, upon the fit of any valve, and the leakage test reported by Mr. Mitchell is very incomplete, in that it does not report the total steam consumption of the engine, for this would be a guide as to its condition, which the facts presented do not directly yield.

It is possible, of course, from the curve in Fig. 6, of Mr. Mitchell's article, to draw some conclusions, but results so obtained are so ridiculous as to create uncertainty that the curve is understood. If, for example, this curve represents leakage per hour, then at 15 horsepower, the normal load of the engine, there was approximately 200 pounds leakage, or 13⅓ pounds per horsepower per hour.

This leakage is represented as being 22 per cent., approximately, of the total steam consumption, which would mean that the steam consumption of the engine was about 60 pounds per horsepower per hour. This is nearly twice what an engine of this type, even of so small a size, would consume if in reasonably good condition.

There are on record many authentic tests of engines with solid piston valves, which show results considerably better than 30 pounds, and it is safe to assert that these results have not been surpassed by any other form of single-valve engine.

From another viewpoint, if it should be accepted that the conclusions drawn by Mr. Mitchell from his tests, that 22 per cent. of the total steam consumption of the engine is attributed to valve leakage, and that this amount is to be looked for in all engines of this type, then it would be proper to assume that the results of recorded tests could be bettered by that amount if some other form of valve had been used.

To demonstrate the absurdity of such an assumption, there follows a brief summary of two such records. These tests are selected as fairly representative of the type of engine under discussion, and because the engines were of small size, although not so small as that upon which Mr. Mitchell's tests were conducted.

The first test was conducted by Professor Diederichs, of Cornell University. The engine had been in operation prior to the test for four years, running much of the time day and night.

Type of engine, single-cylinder automatic with solid piston valve.

Size of engine, inches	13x12
Revolutions per minute (average)	262.17
Steam pressure (average), pounds	118.3
Quality of steam, per cent	98.65
Indicated horsepower	106.7
Steam per i.n.p. per hour, pounds	27.05

The second was a test made by the engineers of the War Department, United States of America. It was an acceptance test, the engine being new.

Type of engine, single-cylinder automatic with solid piston valve.

Size of engine, inches	9½x11
Revolutions per minute (average)	304.2
Steam pressure (average), pounds	102.5
Quality of steam, per cent	97.54
Indicated horsepower	56.58
Steam per i. h. p. per hour, pounds	28.63

This second test, it will be observed, was of an engine considerably smaller than the first, and the steam pressure was lower, which factors probably account for the difference of approximately 1½ pounds in the results obtained. Both engines were manufactured by the Harrisburg Foundry and Machine Works.

That the results given could not have been bettered to the extent of 22 per cent. is evident, for this would mean 21 pounds in one case and 22⅓ pounds in the other, which are results never approached with single-cylinder engines equipped with any type of single valve using low-pressure saturated steam and operating noncondensing.

It is possible that there may be some advantage in the use of rings in the valves of larger engines, where the relative cost of renewal is great, but the test by Professor Diederichs given above is evidence that the wear of a piston valve cannot be very great if such results as obtained in this case can be had after four years of hard service. If rings are used, they should be of the self-adjusting type.

Outside the question of economy, the piston valve has many notable advantages over other forms of single valves and, as far as the economy is concerned, the results have not been surpassed. Its very general and growing use in marine and locomotive practice is strong evidence of the favor in which it is held by leading engineers of this and other countries and not without reason.

The piston valve is the lightest form of valve and can readily be perfectly balanced, thus reducing wear to a minimum and causing the smallest amount of friction and strain on the governing mechanism.

Changes of temperature do not distort it, as is the case with square or rectangular forms of valves, working between the seat and a heavy pressure plate.

High steam pressure and superheat can be safely used.

The valve-chest covers and stuffing boxes are only subject to the pressure of the exhaust.

Greater port openings are afforded than with any other form of single valve, resulting in less wire drawing and better steam distribution.

Repairs and replacement can always be quickly and cheaply effected.

These and other arguments can be summed up in the statement that the piston valve is the most perfectly balanced, quietest running and simplest form of single valve, and whatever the leakage may be past a properly fitted valve of this type, it is certainly not any greater, if as great, as past other forms of single valves, as is attested by the comparative results obtained from engines using this and other forms.

Investigations such as those made by Mr. Mitchell are always interesting, but in reporting them there should be given the full data of the tests, and care should be exercised in drawing conclusions, for otherwise injustice may be unwittingly done and progress rather retarded than advanced.

Benjamin T. Allen,

Chief Engineer, Harrisburg Foundry and Machine Works.

Harrisburg, Penn.

One Valve Performing the Functions of Two

Under the above caption in the issue of October 18, B. U. Potter asks for opinions as to the advisability of the application of his device to a compound pump.

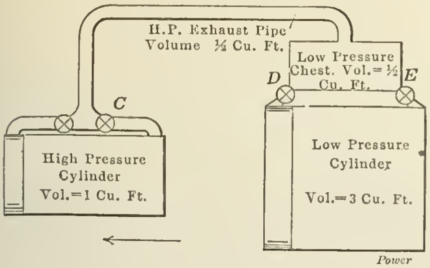

FIG. 1. DIAGRAM OF PUMP WITH USUAL VALVE ARRANGEMENT

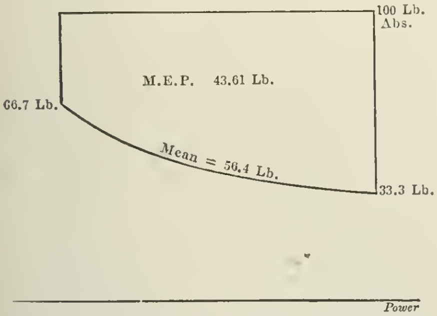

FIG. 2. INDICATOR DIAGRAM FOR HIGH-PRESSURE CYLINDER

If the high- and low-pressure cylinders could be placed side by side close together, his arrangement would probably work out good, but with the usual arrangement of a compound pump, that is, with the high- and low-pressure cylinders placed tandem the scheme involves an increased steam consumption owing to the volume of the pipe connecting the cylinders. The following explanation based on assumed volumes for cylinders and connections shows the comparative weight of steam used for the usual construction, and for that proposed by Mr. Potter.

The usual construction is diagrammatically represented by Fig. 1. Assume steam at 85 pounds gage pressure. The pistons having just completed the stroke in the direction of the arrow, the high-pressure cylinder is full of steam at 100 pounds absolute pressure; the high-pressure exhaust pipe, low-pressure steam chest and low-pressure cylinder are full of steam at terminal pressure due to expansion. Since the ratio of the cylinders is 3 to 1, this terminal pressure should be

$$\frac{100}{3} = 33\ 3 \textit{ pounds}$$

absolute. The next thing that happens is the shifting of all of the valves, E closing and C and D opening. At this time the pressure in the high-pressure cylinder, the high-pressure exhaust pipe and the low-pressure chest equalize and one volume at 100 pounds plus one volume at 33.3 pounds results in two volumes at 66.7 pounds. Then, the pistons travel in the opposite direction and expansion occurs. At the end of expansion, the exhaust pipe, chest and low-pressure cylinder are full, this being equivalent to four volumes, thus the two volumes at 66.7 pounds have expanded to four volumes at 33.3 pounds. The theoretical diagrams from the cylinders are shown in Figs. 2 and 3.

For two expansions the ratio of mean to initial pressure is 0.846; therefore, the mean pressure of the high-pressure exhaust and the low-pressure admission is

$$66.7 \times 0.846 = 56.4 \textit{ pounds.}$$

The mean effective pressure in the high-pressure cylinder is

$$100 - 56.4 = 43.6 \textit{ pounds.}$$

The mean effective pressure in the low-pressure cylinder is

$$56.4 - 15 = 41.4 \textit{ pounds.}$$

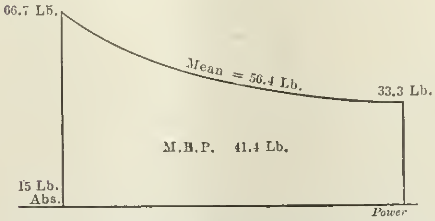

FIG. 3. INDICATOR DIAGRAM FOR LOW-PRESSURE CYLINDER

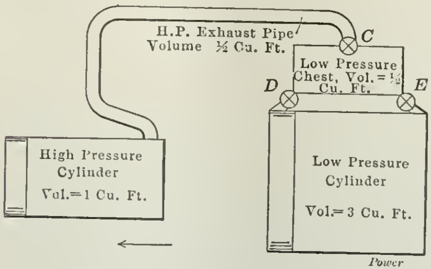

FIG. 4. DIAGRAMMATIC ARRANGEMENT OF PUMP WITH SINGLE VALVE

The areas of the cylinders are proportional to their volumes. Assume the area of the high-pressure cylinder to be 100 square inches and that of the low-pressure cylinder, 300 square inches. Then, the total pressure is found as follows:

$$100 \times 43.6 = 4{,}360 \textit{ pounds}$$
$$300 \times 41.4 = 12{,}420 \textit{ pounds}$$
$$16{,}780 \textit{ pounds}$$

total pressure on the pistons. At each stroke, the high-pressure cylinder takes in 1 cubic foot of steam, which at 100 pounds absolute pressure weighs 0.23 pound.. Therefore, each pound of steam represents a total combined pressure on the pistons of

$$\frac{16{,}780}{0.23} = 73{,}000 \textit{ pounds.}$$

Now, making the same assumptions as to volumes and pressures, the Potter proposed construction is indicated diagrammatically in Fig. 4.

On the opening of the valves the high-pressure cylinder and the pipe are full of steam at initial pressure, hence there are 1½ volumes at 100 pounds absolute pressure. Also, the low-pressure chest contains steam at the terminal pressure of the low-pressure cylinder. From this, the high-pressure diagram works out as per Fig. 5, and the low-pressure as per Fig. 6.

The mean effective pressure in the high-pressure cylinder is

$$100 - 72.7 = 27.3 \textit{ pounds.}$$

The mean effective pressure in the low-pressure cylinder is

$$72.7 - 15 = 57.7 \textit{ pounds.}$$

Then, the total pressure is:

$$100 \times 27.3 = 2{,}730 \textit{ pounds}$$
$$300 \times 57.7 = 17{,}310 \textit{ pounds}$$
$$20{,}040 \textit{ pounds}$$

This total pressure is larger than in the previous case but in order to secure it there has been required both the high-pressure cylinder and the pipe full of live steam or 1.5 cubic feet at 0.23 pound per cubic foot or 0.345 pound of steam.

Therefore, each pound of steam represents a total pressure on the pistons of

$$\frac{20{,}040}{0.345} = 58{,}072 \textit{ pounds.}$$

Thus, under the conditions assumed above, a pound of steam will do about 12.5 per cent. more work with the regular construction than with the construction proposed by Mr. Potter. The principal idea in compounding a steam pump is to save steam and therefore the proposed scheme is a step in the wrong direction and any mechanical advantages are more than offset by the loss of economy in operation.

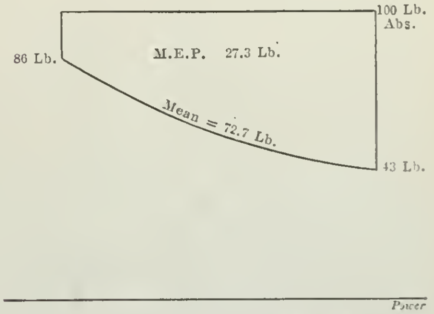

FIG. 5. HIGH-PRESSURE DIAGRAM FROM SINGLE-VALVE PUMP

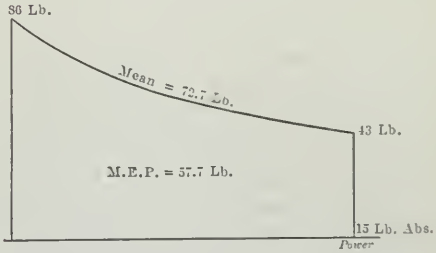

FIG. 6. LOW-PRESSURE DIAGRAM FROM SINGLE-VALVE PUMP

C. H. MAYO.

South Hadley, Mass.

Firing with Natural Gas

In the October 18 issue, Mr. Wilson asks, among other things, the amount of natural gas required per horsepower for return-tubular boilers, and the amount of water that 100 cubic feet of gas will evaporate from and at 212 degrees Fahrenheit. It would be difficult to give a specific answer with the limited amount of information given, but it is quite easy to arrive at an approximate answer.

The amount of heat contained in natural gas is a rather variable quantity, but 1000 B.t.u. per cubic foot would probably represent a fair average. Assuming gas of this quality to be used and that the efficiency of the boilers is 75 per cent., we can readily compute the amount of gas required per horsepower. The accepted equivalent of a boiler horsepower is 33,305 B.t.u. To deliver this amount of heat to the boiler

$$\frac{33{,}305}{0.75} = 44{,}406 \text{ } B.t.u.$$

will be required. This amount of heat will be supplied by 44.4 cubic feet of gas.

Each pound of water will require 970 B.t.u. to evaporate it at 212 degrees Fahrenheit. To supply this amount of heat to the boiler

$$\frac{970}{0.75} = 1293 \text{ } B.t.u.$$

must be supplied by the gas. If each pound of water requires 1293 B.t.u. it is evident that 100 cubic feet of gas, which contains 100,000 B.t.u., will evaporate 77.3 pounds of water.

W. L. DURAND.

Washington, D. C.

Irrigation Problem

In the "Irrigation Problem" presented by L. J. Robertson in the issue of October 18, he makes no mention of the diameter of the pipe line or of the horsepower required to drive the pump; hence, no definite figures can be given in the explanation. However, simple formulas can be given in which substitution will give the desired results.

Nothing would be gained by the second proposed change as it makes no difference in the operation of the pump as long as its position is below the maximum working atmospheric lift, that is, 26 feet above the water level for small pipes and less with increase in the diameter. The fact that the combined length of the suction and discharge pipes remains the same, the same friction losses are present, provided there is no change in diameter or in the number of bends.

The first proposed change is the one that would result in a decrease in friction and consequently a decrease in the power required to drive the pump.

Friction in pipe lines is made up of several elements. First, a loss due to entrance. This is dependent on the shape of the end of the suction pipe and the velocity, and runs from

$$\frac{0.02 \, v^2}{2 \, g}$$

for a bell mouth without strainer, to

$$\frac{0.095 \, v^2}{2 \, g}$$

for a straight mouth without strainer, v being the velocity of the water in feet per second, found by dividing the flow in cubic feet per second by the area of the opening, and g being the value of acceleration due to gravity, 32.2 feet per second. A strainer would materially increase the loss due to increased velocity through the mesh. Second, a loss due to the friction in the straight runs. This depends on the velocity and the diameter of the pipe. It would equal

$$\frac{f\,l}{d} \times \frac{v^2}{2\,g}$$

where

l = Length of pipe in feet;
d = Diameter of pipe in feet;
f = Friction coefficient depending on the nature of the inside surface of the pipe. A common value taken is 0.02.

Third, a loss due to curvature. This would equal

$$\frac{f_c l}{d} \times \frac{v^2}{2\,g}$$

where f_c is a coefficient of curvature depending on the radius of the bend and the diameter of the pipe. If R equals the radius of the bend and d equals the diameter of the pipe, as $R \div d$ decreases f_c increases.

$\frac{R}{d}=$	24	16	10	6	4	2.5
$f_c =$	0.036	0.037	0.047	0.060	0.062	0.072

In sharp-angle fittings the loss factor is greatly in excess of those just given.

The advantages to be gained by a change in line in accordance with the sketch would be 50 per cent. shorter suction pipe and consequently 50 per cent. less friction loss in the runs, provided the ditch to take the place of this length of pipe be made of such a size that the inflowing water will have a velocity less than one foot per second; disposal of three 45-degree elbows and one 90-degree bend and, consequently, their friction loss less the small loss due to the new 30-degree bend. The entrance conditions are considered to remain the same, hence no added loss or gain at this point.

The total head pumped against would be 22 plus 18½ feet, plus the sum of the different losses given above, plus the loss in the pump, all reduced to feet of head. Since the entrance loss is considered as being constant and the loss in the pump is so, these can be omitted from the discussion. The resultant horsepowers, both under the present system and under the proposed system, are

$$\textit{Present horsepower} = \frac{q \times w \times H}{33{,}000}$$

$$\textit{Proposed horsepower} = \frac{q \times w \times H'}{33{,}000}$$

In the two formulas just given

q = Cubic feet per minute $= \frac{16{,}500}{7.48}$
w = 62.5 pounds;
H = Sum of all heads in present system;
H' = Sum of all heads in proposed system.

The saving in horsepower equals

$$\frac{q \times w \times (H - H')}{33{,}000}$$

This can actually be saved as power or can be used to give a greater output.

D. B. TAYLOR.

Troy, N. Y.

Writers among Engineers

B. P. Gage's article in the October 25 issue, regarding writers among engineers, touches upon a subject that should be given more attention by the practical man. There is rarely an issue of POWER from which I cannot get some useful information. Whenever a subject comes up on which I am not posted, a letter asking for aid will bring me a store of information from practical men that could not possibly be found in a dozen volumes treating the subject technically.

Perhaps a great many men think that what they would write could not interest others. This is a wrong view to take of the matter. It not only interests others but aids the writer to a certain extent, since he will study his subject more closely that he may express himself more clearly. There are very few of the calling who have not directly or indirectly had some aid from others, regardless of their assertions to the contrary. The plan of helping those who help themselves should be followed among engineers. The columns of POWER are a most appropriate place in which to execute this plan.

CARL E. WEBSTER.

Higley, Ariz.

Quarter Turn Drive

Referring to "A Quarter Turn Drive" in the November 1 issue, by Walter J. Bitterlich, a distance of twice the diameter of the larger pulley between centers would be too close with a 6-inch belt for economy. However, a 4-inch belt would give entire satisfaction. If I were to use a 6-inch belt for a drive of this nature, I would at least make the distance between the centers of the pulleys two and one-half times the diameter of the largest pulley.

Regarding the maximum belt width it is well known that a thick, narrow belt is better than a thin, broad one, being more durable and working more satisfactorily.

The belt speed for maximum economy should be from 4000 to 4500 feet per minute.

T. J. HAMMERSLEY.

Milwaukee, Wis.

The Problem of Smoke Abatement

Discussion of a paper delivered by D. T. Randall at the Boston meeting of the American Society of Mechanical Engineers on November 10, an abstract of which appeared in our November 15 issue.

After the presentation of Mr. Randall's paper on "Smoke Abatement" at the joint meeting of Boston engineers, on November 10, an interesting discussion took place. Mr. Bailey was first called upon and his remarks in abstract follow:

E. G. Bailey—The essential factors which enter into the prevention of smoke have been known for many years, yet the economical smokeless combustion of coal in everyday practice is still far from being attained. Although the Ringelmann smoke chart has been used for many years, it has only recently received the attention which it deserves. Its use has been greatly stimulated through its adoption and reproduction in convenient form by the United States Geological Survey.

The personal factor of the observer enters into the readings with the Ringelmann chart to a certain extent, but this causes a much smaller error than is generally believed. The greatest difference between the readings of different observers is due to the smoke being sometimes of a brownish cast as compared with the jet black ink used in making the charts. If the charts were made the same shade as the smoke, the chance for personal error would be lessened. However, when using the black chart one should remember that smoke corresponding to No. 1 must allow 80 per cent. of light to pass through it; No. 4, 20 per cent.; and No. 5 is opaque, regardless of the exact matching of color shades. When comparing smoke with the chart, the smoke should be as dense as, or denser than the number assigned it, yet not so dense as the next higher number. For instance, if smoke is denser than No. 3, and not so dense as No. 4, it should be recorded as No. 3. Likewise, if some smoke is being emitted from a stack, yet it is not so dense as No. 1, it should be recorded as — 1 or + 0.

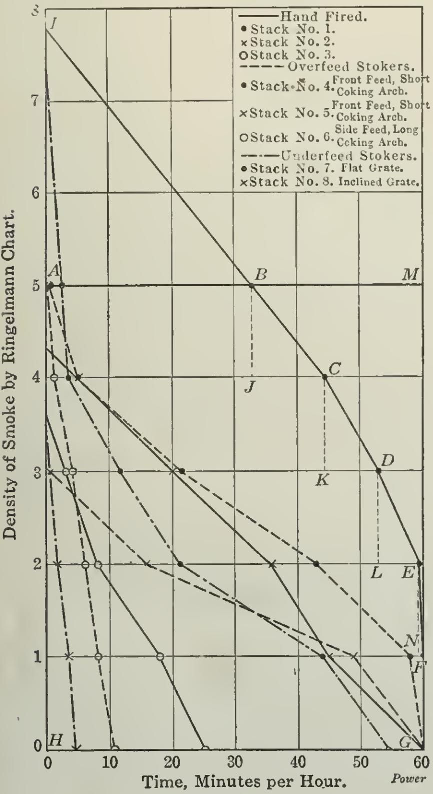

Fig. 1.

Ringelmann chart readings are usually taken at half-minute or minute intervals during the period of an hour or more in the case of stationary stacks, and the readings are plotted in the form of a log, or the density of the smoke is obtained by averaging all readings and expressing the result in percentage of smoke on a basis of No. 5 being 100 per cent. The plotted continuous log gives a good general idea of the manner and regularity of the smoke emission, but is very unsatisfactory for comparing one stack with another, or the same stack from time to time. The average of the readings alone gives but little idea of the nature of the smoke, whether continuous and of low density, or in intermittent puffs of denser smoke. Another method which I have used with good results for several years is to plot a series of readings as one characteristic curve. This is done by adding the total number of readings of each different density, reducing them to a basis of minutes per hour, adding to the number of minutes corresponding to each density all of the minutes corresponding to the greater densities, and plotting Ringelmann chart numbers as ordinates and minutes as abscissas. For example, with stack No. 7, Fig. 1, the readings of a two-hour period reduced to minutes per hour are given in Table 1.

The values in the last column are plotted as abscissas on the lines corresponding to the different density numbers, and the curve drawn through these points indicates the general character of the smoke being emitted.

The curves in Fig. 1 represent a variety of plants located in different places with different equipments, but all burning semi-bituminous coal containing between 17 and 21 per cent. of volatile matter. Most of this data was obtained in connection with the drafting of the smoke bill recently enacted for Boston through the efforts of the fuel-supply committee of the Boston Chamber of Commerce.

No. 5 on the Ringelmann chart includes all smoke that is opaque, and, for a stack of a given diameter, a certain amount of carbon particles per cubic foot is necessary to make this smoke opaque, but twice as much carbon or smoke could be carried per cubic foot of gas without changing the reading. This point is graphically brought out in these characteristic smoke curves. For instance, stack No. 1 would not be apt to produce smoke of No. 5 density, no more nor no less, for thirty-three minutes out of an hour; but during a part of this time it was undoubtedly much denser, as indicated by carrying the curve up until it intersects the zero line at a density of about 8. The exact nature of the extended part of this curve is problematical, but it illustrates the fact that the average density of smoke may be much greater than is indicated by merely averaging the readings in the usual method. The average density of the smoke, as compared with No. 5 as a basis, is the area *H I B C D E F G H* divided by *A M G H A*, or 103 per cent.

TABLE 1.

Density, Ringelmann Chart Number.	Time during which smoke of each density was emitted. Minutes.	Time during which smoke of each density and darker was emitted. Minutes.
5	2.5	2.5
4	1	3.5
3	8	11.5
2	9.5	21.0
1	23	44.0
−1 + 0	10.5	54.5
0	5.5	60.0
Total..........	60.0	....

If the area representing smoke denser than No. 5 were neglected, the result would be area *A B C D E F G H A* divided by *A M G H A*, or 87.8 per cent. If the usual method of averaging smoke readings, such as was apparently used in the United States Geological Survey bulletin No. 373, be applied to this stack, the density referred to No. 5 is only 83.3 per cent. This error is due to the omission of the triangular areas *B J C B*, *C K D C*, *D L E D* and *E N F E*, which properly belong in the area representing the total smoke emitted. All of the No. 4 readings, representing the eleven and one-half minutes from *J* to *C*, were as dense or denser than No. 4, and less than No. 5, so they would naturally be distributed along the diagonal line *B C*, and the same holds true with respect to the other parts of the curve. Applying the averaging method to the curve representing stack No. 5, Fig. 1, gives 21.8 per cent., although the true average from the area is 31.8 per cent. The flatter the curve, the greater is the error caused by the use of the averaging method as usually applied, and the result is always too low.

Stacks Nos. 4, 5 and 6 were connected to stoker furnaces of the over-feed type. Nos. 4 and 5 had short coking arches with front feed, and the plants were similar with respect to the number of boilers and the size of stack, but the relative rates of combustion, volatile matter in the coal, etc., are not known. Stack No. 6 had only one furnace of the side-feed type with a long coking arch connected to it. Stacks Nos. 7 and 8 were connected to under-feed stokers, the former having five stokers with flat grates, and the latter three stokers of the inclined-grate type.

TABLE 2.

Stack Number.	Diameter of Stack, Feet.	Number of Boilers Connected.	Type of Boilers.	Rated Boiler Horsepower.	Grate Area, Square Feet.	Method of Firing.
1	8	6	Horizontal Return Tubular.	900	204	Hand spreading
2	7½	8	Horizontal Return Tubular.	1200	272	Hand spreading
3	4	1	Porcupine.	300	56	Hand spreading
4	7	1	Horizontal Return Tubular.	1050	238	Hand spreading

No specific conclusions as to the value of the different methods of firing should be drawn from the curves of Fig. 1, as they are merely given to show the great variation that exists in actual practice between plants with similar equipment.

TABLE 3.

	C	D
Analysis of coal as fired:		
Moisture, per cent	4.81	5.40
Volatile, per cent	16.22	18.81
Fixed carbon, per cent	70.40	69.00
Ash, per cent	8.57	6.79
Sulphur, per cent	1.20	0.70
B.t.u	13,639	13,838
Average draft in firebox, inches of water:		
Boilers of stack No. 1	0.45	0.32
Boilers of stack No. 2	0.25	0.31
Boilers of stack No. 3	0.30	0.32
Boilers of stack No. 4	0.21	0.27
Coal fired per square foot of grate per hour, pounds:		
Boilers of stack No. 1	15.1	13.8
Boilers of stack No. 2	11.9	11.8
Boilers of stack No. 3	19.6	16.5
Boilers of stack No. 4	9.2	11.7
Flue-gas analysis, percentage of CO_2 by volume:		
Stack No. 1	8.1	7.7
Stack No. 2	6.9	7.4
Stack No. 4	7.4	9.5
Flue-gas analysis, percentage of CO by volume:		
Stack No. 1	0.0	0.0
Stack No. 2	0.0	0.05
Stack No. 4	0.0	0.20
Excess air from flue-gas analysis, per cent:		
Stack No. 1	137.00	146.00
Stack No. 2	180.00	149.00
Stack No. 4	155.00	87.00
Percentage of smoke with No. 5 on Ringelmann chart as 100 per cent., from Fig. No. 3:		
Stack No. 1	12.8	22.8
Stack No. 2	18.2	35.8
Stack No. 3	34.7	56.0
Stack No. 4	4.3	17.2

The curves shown in Fig. 2 were all taken from the same stack connected to one internally fired boiler of the locomotive type. The same kind of coal was fired in each test by the same fireman, using the spreading method. The only changes made on the different tests were the amount and the temperature of secondary air supply. The coal used on these tests contained about 30 per cent. volatile matter, and over 50 pounds of it were burned per square foot of grate surface per hour.

The agreement between different observers is shown in tests Nos. 1 and 3. Observers *A* and *C* had had very little experience in the use of the Ringelmann chart, and did not know in either case that observer *B* was taking readings.

The curves of Fig. 3 represent the smoke emitted from four different stacks at one plant on two eleven-hour tests made on consecutive days. The tests were primarily made to determine the comparative values of the different coals from the evaporation on the entire plant, and the smoke readings were taken as a side issue, as the location of the plant was such that the prevention of smoke was given no consideration. There were nine firemen, and their methods of firing were not changed in any way from their usual practice.

Table 2 gives the general dimensions and equipment connected with each stack, and Table 3 contains the general data which have direct bearing upon the amount of smoke produced. All conditions were maintained as nearly uniform on the two tests as the daily operation of the plant would permit, the only intended change being in the coal burned. The principal difference between the coals so far as smoke was concerned was in the percentage of volatile matter, which was 16.22 and 18.81 in coals *C* and *D*, respectively, as fired, or 17.04 per cent. and 19.88 per cent. on the dry basis. Owing to changes in the draft and the amount of clinker formed, the rate of combustion was not quite the same for the two tests. There was also some change in the flue-gas analysis and air excess. The gas samples were taken from the main flues, and included considerable leakage of air through the boiler settings.

Slight differences in the percentage of volatile matter, rate of combustion, excess air, method of firing, etc., are found to have a marked effect upon the amount of smoke produced when the characteristic curves are drawn. By the use of such data and curves, it is possible to determine the relative importance of the different factors affecting this problem, and to take intelligent steps to reduce the density of the smoke to the desired limits.

FREDERICK H. KEYES—The power plants of New England are so situated with reference to the source of supply that, as a rule, it is impracticable to use anything except bituminous coal; and unfortunately the price is usually the controlling factor rather than quality, and I believe that the quality is largely responsible for the amount of smoke. Some of the largest users of bituminous coal have already found it advantageous to buy their coal under specifications especially drawn up to meet their requirements; and this would seem to be the first step in the right direction, so far as smoke is concerned.

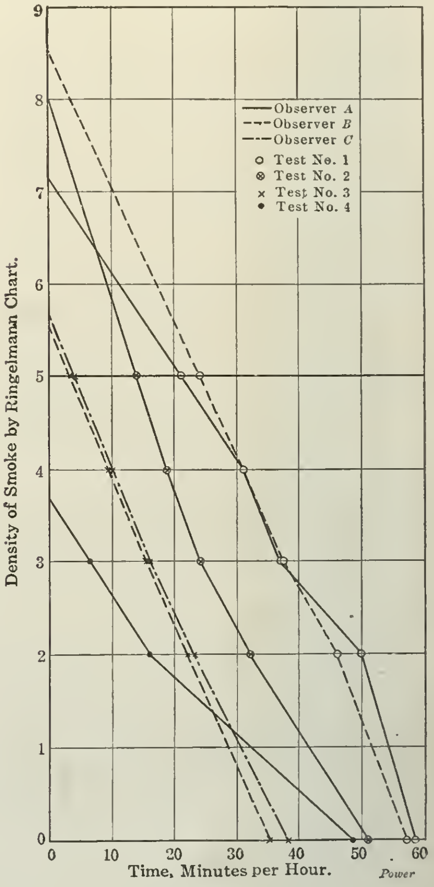

FIG. 2.

Having obtained coal with the minimum of smoke-producing qualities, the next step is to fire and burn it properly. Here is where it is absolutely necessary to have the coöperation of the fireman.

Next to the fireman, the quantity of air, together with the manner and conditions under which it is mixed with the products of combustion, is of the greatest importance; but this point must be determined for each individual case according to the kind of coal used and the general conditions governing the operation of the plant.

I believe that special devices, such as steam jets and firebrick arches, with few exceptions, usually cost more to operate and maintain than can be saved through the use of low-grade fuel for which such special devices are invariably required to obtain even approximately satisfactory results.

Henry Bartlett—I have read with much interest Mr. Randall's paper and note that he deals with smoke abatement from a stationary standpoint. I shall confine my remarks to the problem of smoke prevention with locomotives. Bituminous coal is the principal fuel supply for locomotives, and the subject of smoke abatement has been a live one, especially within our cities, where the situation has become very serious.

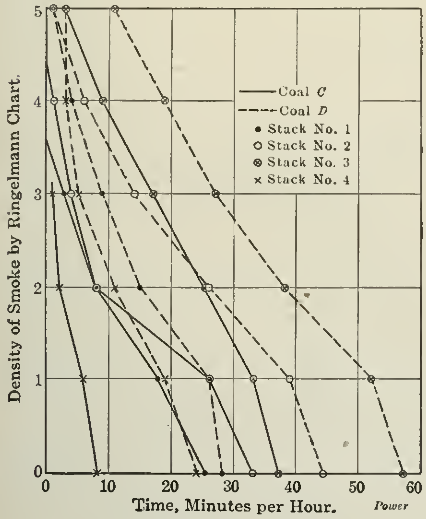

Fig. 3.

The locomotive boiler is called upon for ranges of work greater than that of any other service. One moment the locomotive is standing quietly in the roundhouse with banked fire and only 40 or 50 pounds of steam on the boiler, and within the hour it may be running on the road at 200 pounds of steam and exerting its maximum power. It may be moving along pulling a heavy train and emitting little smoke, when the next moment it is suddenly stopped or required to take a siding. Under such changes it is impossible to eliminate the smoke as the large body of coal necessary to perform the work keeps on emitting the volatile gases until these have disappeared. The terminals of runs are necessarily in large cities and communities where locomotives are housed and have their fires cleaned and built up for their runs.

To avoid this the question might be asked—why not burn smokeless fuel? I only wish this solution of the problem were possible, but the only smokeless fuels available are anthracite coal, coke and oil. Anthracite coal is found principally in Pennsylvania, where statistics show that 70,000,000 tons are mined annually, or less than the amount of coal yearly consumed by the railroads. Considering the demands for this fuel in domestic purposes, where no other fuel seems available as a substitute, should the railroads resort to its use in general, the supply would be insufficient and the price would so advance as to render the cost prohibitive.

As regards coke the Boston & Maine railroad is more fortunately situated in having on its line a large producer, and for the last few years has availed itself accordingly to the fullest practical extent.

Oil properly fired is a smokeless fuel, found extensively in a region where it is largely used by the neighboring railroads, but it also is limited in supply and the oilfields are so located that the cost of its transportation would make it prohibitive for general use.

Smoke-consuming devices have been largely employed by different railroads in trying to solve this momentous smoke problem and large amounts of money have been spent in seeking the ideal in this direction, but so far without success. The only device of this kind which is retained in common use is the brick arch.

George H. Barrus—When a charge of bituminous coal is thrown into a hand-fired furnace, the first thing which occurs is the heating of the coal to the temperature of ignition. During this process the volatile gases in the coal are driven off. These escape with great rapidity in the early stages, after which the quantity of gas gradually diminishes, and finally nothing but incandescent coke remains.

The gases first evolved from a charge of coal are in their original cold state. Contact with the radiant heat of the furnace increases the temperature of the exposed portions of the gases sufficiently to ignite them. Those portions of the gas which are not favorably exposed receive an insufficient quantity of heat to become thoroughly ignited, and these pass out of the furnace in an unconsumed state. It is the underheated gas thus formed which causes smoke, and the whole problem of smoke prevention consists in heating sufficiently these gases before leaving the combustion chamber. When they receive the proper amount of heat before entering the boiler, they are burned without smoke and the degree of smokelessness obtained depends upon the degree with which the heating of the gases has been effected.

This feature of the subject may be illustrated by referring to an experiment I made on a battery of two 300-horsepower vertical-pass horizontal water-tube boilers. The result aimed at was to ascertain the effect of firing the two boilers alternately and employing the incandescent coke of one furnace as a medium for overheating the gases evolved from fresh charges of coal in the other furnace.

To prepare for the experiment an opening was cut through the intermediate wall between the two furnaces. This opening, which extended horizontally the whole length of the grate and vertically from the grate to the lower row of tubes, put the two furnaces into free communication, and the closing of either of the flue dampers was sufficient to cause the products of combustion from the furnace of one boiler to pass through the intermediate opening and over the fuel bed of the other furnace, and thereby secure the object sought.

The method of operation was as follows: Designating the boilers as Nos. 1 and 2, the two furnaces were fired alternately at equal intervals, and whenever a boiler was fired its damper was closed. The products of combustion from No. 1 furnace, just fired, passed through the intermediate opening, over the fuel bed of No. 2 furnace, mixed with the products of combustion from that furnace and passed on through No. 2 boiler and No. 2 damper. When the volatile gases had distilled off from the fresh coal, the positions of the two dampers were reversed, No. 1 being opened and No. 2 closed. Then No. 2 furnace was fired and the volatile gases from this furnace passed through the intermediate opening into No. 1 furnace, thence over the incandescent fuel bed in that furnace and on into No. 1 boiler along with the products from No. 1 furnace.

To compare the amount of smoke produced in this system with the ordinary system of operation, the boilers were first run in their usual manner. New River coal was used and was charged through the two outside doors first, there being three doors in all for each boiler, and then after waiting three to five minutes, the middle door was charged, one boiler being fired at a time. Smoke observations were made every minute for a continuous period of two hours, the estimated percentage of dense black smoke being judged as it appeared to the eye on escaping from the top of the chimney.

The average density of smoke observed for the entire period of two hours, including the time when there was no smoke, was 13.6 per cent., while the maximum was 75 per cent.; and there was an entire absence of smoke, or only a trace, for fifty-four minutes.

When the alternate system was brought into use, the three doors of each boiler were fired in rotation, there being no waiting between doors. In general, 12 shovels of coal were fired at each round.

The average density of smoke observed during the entire period of two hours, including the time of no smoke, was 2.3 per cent. The maximum amount was 10 per cent., and there was an entire absence of smoke, or only a trace, for one hour and twenty-two minutes out of the two hours.

Inquiries of General Interest

Number of Expansions

How may I find the number of expansions taking place in the cylinder of a steam engine?

N. O. E.

The number of expansions taking place in a steam-engine cylinder is always the reciprocal of the cutoff; that is, one divided by the cutoff. If the cutoff is one-fifth, the number of expansions will be five. The number of expansions may also be found by dividing the absolute initial pressure by the absolute terminal pressure. If the initial pressure is 160 pounds gage, 175 pounds absolute, and the terminal pressure is 20 pounds gage or 35 absolute, dividing the former by the latter will give five as the number of expansions.

Weight of Safety Valve Lever

In safety-valve calculations, why is the distance of the center of gravity of the lever from the fulcrum taken?

W. S. L.

The weight of the lever acts on the valve, tending to keep it closed against the pressure of the steam, and the effect is the same as it would be if its total weight were concentrated at the center of gravity; therefore, the distance of the center of gravity of the lever from the fulcrum is multiplied by its weight.

Pulley Dimensions for a Given Horsepower

What determines the size of a pulley needed to transmit a given horsepower?

P. D. H.

The power to be transmitted and the speed at which it revolves. For a single belt choose a diameter and face that for the number of revolutions will give from 60 to 70 square feet of belt surface passing over the pulley per minute for each horsepower. For a double belt the surface may be reduced one-fourth.

Setting Engine Valves

What is the correct rule for setting the valves of Corliss engines?

S. E. V.

The elements of valve setting, whether for Corliss or other engines, may be stated in a very few words. But each step in the process may be enlarged upon almost without limit. No rule, however elaborate, will furnish anything beyond directions to perform certain operations. The briefest will do as much. To set the valve or valves of any engine, make the eccentric and valve rods of the right length, then with the engine on the center turn the eccentric in the direction the shaft is to revolve until the proper amount of lead is secured; fasten the eccentric and the job is done.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Steam Chest Pressure

What should the steam pressure be in the steam chest of an engine with a throttling governor, the engine running with a normal load and the boiler pressure being 100 pounds?

H. B.

Assuming a mean effective pressure of 40 pounds in the cylinder and a cutoff at five-eighths of the stroke and that 80 per cent. of the theoretical pressure is realized, then

$$40 \div 0.80 = 50 \textit{ pounds},$$

the theoretical mean effective pressure. At five-eighths cutoff the mean effective pressure will be nine-tenths of the initial pressure. If the mean effective pressure is 50 pounds the initial will be 55.55 and as the probable fall between the steam chest and the cylinder will not exceed two pounds the steam-chest pressure may be taken to be

$$55.55 + 2 = 57.55 \textit{ pounds}.$$

Width of Exhaust Port

In laying out a valve seat, how can I determine the correct width of the exhaust port?

W. E. P.

The width of the exhaust port for a plain slide-valve seat should be the width of the steam port, plus one-half the travel of the valve, minus the width of the bridge between the steam and the exhaust ports.

Pressure in Feed Pipe

In order to feed water into a boiler carrying 80 pounds pressure, must the pressure in the feed pipe be above that in the boiler?

P. F. P.

Yes. In order to produce a flow in any system there must be a difference in pressure. If the pressure is the same in the pipe as in the boiler there will be no flow. If the pressure in the boiler is above that in the pipe, the flow will be from the boiler into the pipe, and if greater in the pipe than in the boiler, the flow will be from the pipe to the boiler.

Temperature of Steam at Reduced Pressure

I have a vulcanizer using steam reduced to 40 pounds from a pressure of 100 pounds. What is its temperature?

T. R. P.

The total heat in steam at 100 pounds pressure is 1188.8 B.t.u.; at 40 pounds, 1175.4

$$1188.8 - 1175.4 = 13.4 \textit{ B.t.u.}$$

The specific heat of steam at 40 pounds pressure is approximately 0.52; therefore, the additional 13.4 B.t.u. in the steam at the reduced pressure would superheat it 25.7 degrees.

The temperature of steam at 40 pounds pressure is 287.1 degrees and the 25.7 degrees added will make it 312.8. This is assuming that the steam at 100 pounds pressure is dry. It may contain so much moisture that when reduced to 40 pounds its temperature will not be above that due to its pressure.

Efficiency of Boilers

What is the difference in efficiency of a boiler when expressed in pounds of coal and pounds of combustible?

E. O. B.

The heat absorbed by the boiler is divided in both cases by the calorific value of the fuel.

In one case the efficiency is expressed by the formula,

$$\textit{Efficiency} = \frac{\textit{Heat absorbed per pound of combustible}}{\textit{Calorific value of one pound of combustible}}$$

and in the other by

$$\textit{Efficiency} = \frac{\textit{Heat absorbed per pound of coal}}{\textit{Calorific value of one pound of coal}}$$

Advantages of Compound Engines

What are some of the advantages in using a compound engine rather than a simple one?

A. C. E.

A higher initial pressure, a greater number of expansions and less cylinder condensation, owing to a shorter temperature range in each cylinder than would obtain if the whole number of expansions took place in one cylinder.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for October, 1910, *was* 132,000.

November 1	33,000
November 8	33,000
November 15	33,000
November 22	33,000
November 29	34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

Accumulators for Furnace and Boiler Capacity

It is a well recognized fact that a steam engine of given displacement will operate most efficiently upon a particular load under given conditions of initial and back pressure. The engine which drives a vessel, a pump or a cotton mill has a decided advantage, if it is at all well proportioned to its work, over the engine which supplies the varying demands of an electric railroad, and which can be running at its maximum efficiency only a small part of the time.

This is equally true, if not more true, of the boiler. A boiler can be designed to furnish a given amount of steam per hour in which both the furnace and the boiler efficiency will be high; but, if this boiler has to work at one moment fifty per cent. above the capacity to which it is best adapted and at another at fifty per cent. less, the result must be less favorable.

It is sought to ameliorate the engine condition by straightening out the load curve as much as possible. A storage battery of sufficient capacity would do this completely, but would cost more than the saving would come to. The capacity is also adjusted to the load, when the variations are slow enough and of sufficient amplitude, by cutting in and out engine units.

The boiler condition may be ameliorated in the same way either by cutting in or out more or less units or by providing some sort of an elastic takeup or accumulator between the boiler and the engine.

The steam space, including that of the piping, furnishes such a takeup to a limited extent. When the demand exceeds the rate at which the boilers are working the steam space becomes partially depleted and the pressure falls. If, on the other hand, the rate of demand is less than that at which the boiler is working, the pressure will rise.

It has been one of the first precepts of a boiler room to keep the pressure constant, but it is a question if the boiler pressure cannot be allowed to vary through a considerable range with less damage to the over-all efficiency than would result from the constant manipulation of the damper and the slice bar necessary to hold it constant. At the pressures ordinarily carried, a considerable pressure drop will produce a comparatively insignificant change in the initial temperature, and it is the temperature range which affects the efficiency.

Some years ago there was introduced in England a system of thermal storage. Great tanks in the boiler room were filled with water which was allowed to circulate in the boilers and become heated when the load was light, and which in its heated condition was drawn upon when the load was heavy. It is doubtful if the possibilities of such a system, combined with economizers, have been exhausted.

The boiler furnace is at the same time a gas producer and a gas burner. If it gets to producing gas faster than it is needed to be burned, there is trouble. Sometimes there is trouble to burn, in the constricted combustion chamber, all the gas which it can produce. The dutch oven or extension furnace is a move in the direction of a separate producer and a gas-fired furnace. Something like a gasometer between the producer and the furnace would allow the producer to work always at its maximum efficiency and the combustion to be controlled in accordance with the load, but such an outfit would probably bear the same relation to the boiler that the storage battery does to the engine, and if gas were available it would be better to burn it in the engine itself rather than in the boiler.

The limit of boiler capacity seems to lie not in how much heat the surfaces can absorb, but in how much coal can be burned in the furnace. It is largely a question of how much firebrick one can afford to burn up as against buying and running more boilers.

Proximate Analysis of Coals

The present tendency in power-plant operation is to pay special attention to the quality of fuel purchased, not only as to its total heat contents, but also for the purpose of determining the proportions of its various constituents so that it may be adaptable to the service required. Most of the large plants have facilities for making accurate analyses of the coal received, such tests forming part of the regular operating routine; but this necessitates a somewhat elaborate apparatus and the employment of an expert to make the tests, the expense of which, in the large plants, is more than compensated for by the benefits derived.

In the small plants, however, it is not always expedient to go into such refinements; at the same time, it is desirable that a check should be made upon the quality of coal being supplied. For such purposes the proximate method of analysis may be resorted to. This gives the approximate proportions of fixed carbon, volatile matter, moisture and ash. It is made by subjecting a sample of coal to a temperature of 250 to 300 degrees Fahrenheit to drive off the moisture; then to a red heat to drive off the volatile matter; then to a white heat which causes the carbon to pass off as carbon dioxide, leaving the ash. By weighing the coal in the beginning and the residue after each operation the various percentages can be computed. By exercising a little care most anyone can make such an analysis; and, although it does not afford an accurate basis for computing the heat value, it does afford information as to the general characteristics of the coal and also a means for comparing the relative heat values of various coals.

The Human Element

Not long ago an analysis of the cause of all of the accidents reported from power plants for one year disclosed the fact that over seventy per cent. were due to the carelessness of attendants. Human machines are, according to the result of this investigation, less than one-half as reliable as those made of wood and metal.

To what extent the mental and physical condition of the human element influenced the result will probably not be known; but it would be interesting to learn for a certainty whether the man with tasks well within the limits of his mental and physical strength is more prone to neglect obvious duties than the one with hours and toil beyond the ability of the average man.

Is the overworked fireman more liable to allow an erratic water level in the boiler than the one with time for smokes and naps between firings?

Is the oiler watching a single engine more attentive to the hight of the oil in a few cups or more solicitous as to the temperature of the bearings than he who can scarcely replenish the last cup and touch the swiftly passing crank pin before the first one is empty?

There are engineers, firemen and oilers the routine of whose uneventful daily work is never varied by an emergency, and there are others whose utmost energy is taxed in replacing broken follower bolts just in time for the heavy run while a blown-out joint in the main steam line shrieks for attention.

In the great majority of power-plant accidents is it the man or the material that is primarily at fault?

To the personal equation the analyst has ascribed seventy per cent. of the failures. But in the remaining thirty per cent. due to faulty design or construction, how many of the accidents could have been avoided if the mind of the man had been on his work? An engineer who is alert and resourceful is a treasure, but one with that forethought and attention to operating conditions which renders exhibitions of spectacular resourcefulness unnecessary is priceless.

Gas Engine Valves

It may not occur to many operators of gas engines that the heads of the inlet and exhaust valves have to withstand exactly the same pressure per square inch that is applied to the other interior surfaces of the combustion chamber when the charge is ignited. Of course, it is obvious when one is reminded of it, but a good many inexperienced designers have made the mistake of regarding the valves merely as hole-closers and have made the disks thin and the heads flat to reduce the weight and the consequent tendency to hammer the seats. The unsupported area of a three-inch valve disk is about seven square inches, and when the explosion pressure goes up much beyond three hundred pounds the total pressure on such a valve is more than a long ton. That is why most valve heads are heavily crowned.

The proper way to prevent the valves from hammering their seats is to make the retreating curve of the lifting cams such that the cams won't "run away" from the rollers and allow the springs to bang the valves down on their seats. This sounds like the essence of simplicity, but it isn't. The proper layout of gas-engine valve cams comes pretty near being a fine art.

Enlightening the Engineer

Not long ago the subscription representative of a leading engineering paper called at the office of a large manufacturing concern, and upon stating his business to the superintendent he was told that the firm had no use for such a thing as a technical paper. Thereupon, he requested permission to talk the matter over with the engineer, to which the superintendent replied: "Positively no, I do not allow my engineer to fool with such stuff; and, furthermore, if I hear of him doing so, I will fire him."

Thus launching into the subject he went on to say that he always knew what was needed and how the plant should be run without requiring any suggestions from his engineer; besides, his own time was taken up with the commercial side of the business and he had no time to devote to the study of engineering subjects. He claimed that such matter as is contained in an engineering paper only tends to ruin an engineer, giving him big ideas and making him less useful to his employer.

Here is the case of a man condemning himself by his own words. He is under the delusion that the interests of the firm will be served best by keeping his engineer in ignorance of the latest engineering developments, admitting at the same time that he, himself, has no time for such matters. Fortunately for the industrial interests, this type of superintendent is fast disappearing, and is giving way to the wide-awake management which realizes that the engineer plays an important part in the cost of production.

To meet increasing competition in the manufacturing industries, improved machinery has been brought into use and the methods of production have been standardized to such an extent that the zenith of economy has almost been reached in this line. To attain further economy, attention has been directed toward the power plant; the consulting engineer has been called in and new equipment has been introduced which is designed to meet the special requirements of the service. But having gone this far is not enough; the engineer is a factor to be reckoned with and upon his efforts and ability largely depends the efficiency of the plant. For this reason the management should see to it that he is given every opportunity to improve his knowledge and to keep abreast of the times.

Boiler Tubes

News comes of another boiler-tube failure. They are becoming woefully frequent. What is the matter?

Charles S. Blake, of the Hartford Steam Boiler Inspection and Insurance Company, called attention to this condition at the last meeting of the American Boiler Manufacturers' Association. The tubes which are furnished now, he said, for pressures of one hundred and fifty and one hundred and seventy-five pounds are identical in thickness with those that they used to furnish twenty-five years ago, when pressures were from eighty to one hundred pounds. The average quality has not improved. The standard commercial tube is no better than, if as good as, the tubes of those days. We know of two cases where pieces the size of one's hand have come bodily out of tubes as clean as though they had been taken out with a punch. There is something wrong with the making of a tube which fails in that way.

There is a wave motor running at Atlantic City which actually generates electric current and compresses air. Whether it will compete with present methods depends upon the required investment per horsepower. The projectors claim that this cost, including the cribwork, compares favorably with that of a steam plant.

The Climax boiler which exploded at the works of the American Manufacturing Company in Brooklyn, an account of which appeared in our November 1 issue, is claimed to be the first of this type that has ever exploded.

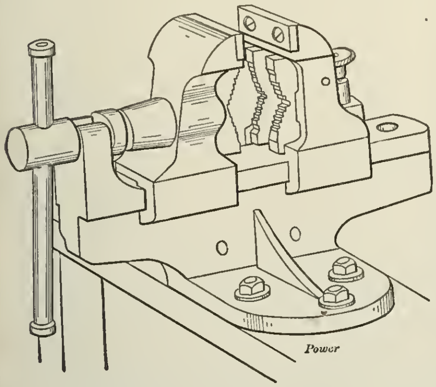
How many books do you know well?

New Power House Equipment

Combination Pipe and Bench Vise

The accompanying illustration represents an improvement in a quick-adjusting combination pipe and bench vise. It is provided with an adjusting pin attached to the back jaw of the vise in such a way that it engages in the base of the vise.

This base has three holes in which the pin fits. When the pin is in the first hole, the vise holds from ½- to 1-inch pipe; when in the second, it holds up to 2 inches, and when in the third hole, from 2½- to 3-inch pipe.



Pipe and Bench Vise

The vise proper has steel-faced jaws. Sockets for 1¼-inch pipe legs are also provided.

This vise is manufactured by the Armstrong Manufacturing Company, of Bridgeport, Conn.

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

The Improved Taylor Stoker

In the article on "Under-feed Automatic Stokers" which appeared in the November 15 number, some of our readers may not have realized that the gravity under-feed stoker described was the Taylor, made by the American Ship Windlass Company, of Providence, R. I. Such is the case and since this article has been published, some improvements in the stoker have been perfected.

These changes relate principally to the operation of the lower line of plungers, which push the burned-out coal onto the dumping plates. The travel, or length of stroke, of the upper row of rams which assist gravity by pushing the coal outward into the fire is constant, but the travel of the lower row can be increased to push the refuse out more rapidly just after the dumping plates have been dropped, or whenever it is desired to more quickly get rid of the refuse which has collected in the lower part of the furnace.

As previously built, each lower ram was connected directly to the bell crank which operated the upper plunger. Adjustment of the stroke of the lower plungers was made for each separate retort. With the improved mechanism, the travel of all the lower rams may be lengthened or shortened simultaneously. This is accomplished by means of a sort of pantograph lever connection having a cam and balance weight on the upper member, as shown in the sectional view of the furnace. Pivoted to the plunger connecting rods these cams are connected by means of chains, lever arms and shaft to the operating handle at the center of the stoker. To alter the travel of the lower rams the adjusting lever is dropped, releasing the cams and permitting them to become horizontal, which action lengthens the stroke of the plunger. In case the vertical connection of the bell-crank lever is in such position that the cam cannot drop, it merely remains at rest until the connection moves backward and the counterbalance weight falls and carries the cam into position.

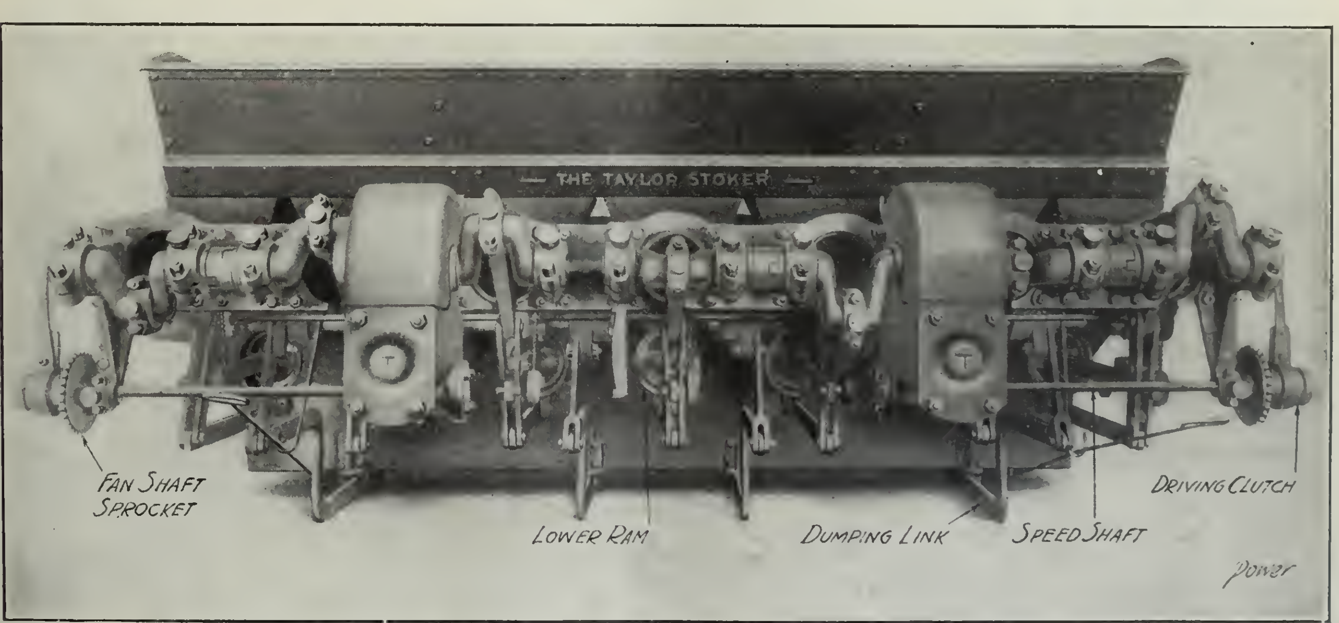


Fig. 1. Front View of Taylor Stoker

Another change which simplifies the operation of the stoker is the relocation of the speed-shaft sprocket. It is now placed at the end of the stoker with the clutch and faceplates which hold the shearing pin on the outside. This greatly facilitates the replacing of the shearing pins when required and also the throwing of the stoker on or off the line. It also improves the connection to the fan shaft.

A third improvement is the alteration of the dumping plates. In early forms, the dumping plate was cast in one piece but the present design calls for a dumping plate in sections which are independently renewable, facilitating and lessening the cost of repairs. Improvements of minor importance are the use of lock nuts in all places where possible jar would tend to lessen ordinary nuts, the fitting of the crank-shaft brackets to the ram boxes with body-bound bolts, and an increase in the size of the gears in the gear boxes and the shafts and bearings. To insure more certain connection between the cast-iron ram boxes and the tuyere boxes they are now carefully doweled together.

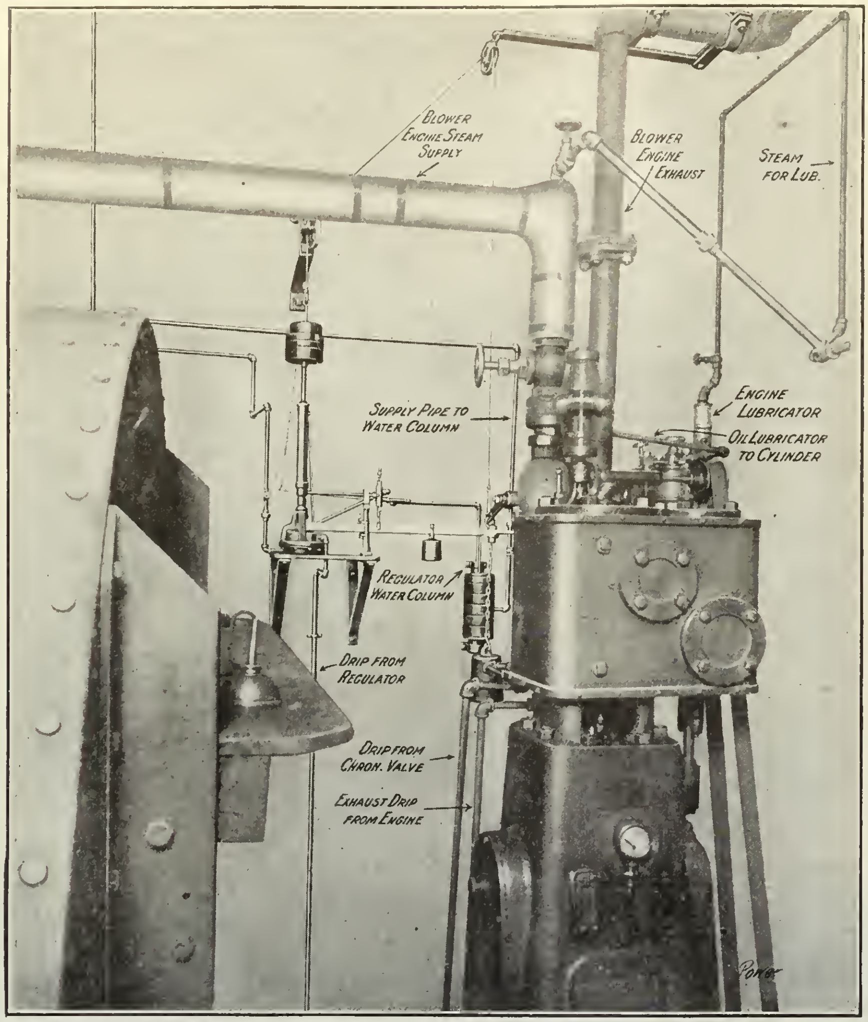


Fig. 2. Stoker Engine and Fan Draft Regulator

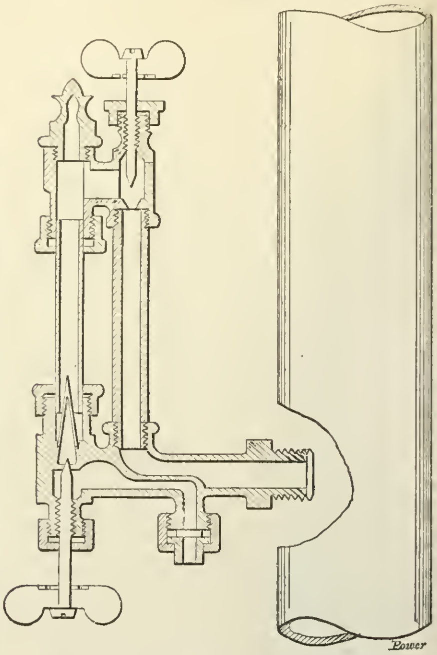
The air for combustion is supplied by a volume blower driven by the same engine which operates the stokers. A change in the rate of feeding coal is therefore accompanied by a change in the rate of supplying air. After the stoker is installed, tests are made to determine the ratio of air and coal to give perfect combustion. The relative speed of the fan and stoker is then fixed; thereafter no further change is necessary. The fireman does not need to worry as to the air supply. In this way a high efficiency is obtained throughout a wide range of load, and errors of judgment are eliminated.

The lever arm of the regulator is connected to the lever arm of a butterfly or chronometer valve in the steam line of the fan engine by means of cables and pulleys. Whenever the steam-pressure fluctuations cause the damper-regulator lever arm to move up or down the fan engine speeds up or slows down accordingly.

The Ashton Sight Feed Lubricating System

This system consists of a central receptacle, or reservoir, from which the cylinder oil is distributed to each cylinder and fed through a sight-feed device shown in section in the illustration. The feeder is screwed into the steam pipe by the nipple. The oil connection from the reservoir is made at the union *E; C* is a valve to be closed should the glass break, and *D* is the needle valve for regulating the flow of oil.

Any number of cylinders may be lubricated by this system, the tank supplied being of a size corresponding with the amount of oil to be fed. This system is manufactured by Walter L. Flower & Co., 1023 Chemical building, St. Louis, Mo.



Ashton Lubricating System

Opening American Museum of Safety

The formal exercises in connection with the permanent exhibition of safety devices of the American Museum of Safety were held on Monday evening, November 21, at the auditorium of the Engineering Societies' building, Philip T. Dodge, presiding. The Museum of Safety is the fourteenth similar organization of which there are twelve in Europe and one in Canada. The object of the museum is the conservation of human life, by means of a permanent exhibit of the best and most practicable safety devices for making safe the dangerous parts of machines and processes.

A number of gold medals were presented and a greeting from President Taft was read to the gathering.

Dr. W. H. Tolman, director of the museum, who had spent the summer in Germany, studying its system of accident prevention, showed by means of lantern slides how the fatherland is saving 50 per cent. of the accidents to labor and told how in the last 25 years Germany had spent 2½ billion dollars in preventing accidents.

T. C. Martin, chairman of the executive committee, made an earnest plea for additional funds to enlarge the exhibition space as the present exhibit hall is already outgrown. Other speakers of prominence responded to the call of Mr. Dodge with a few brief remarks on the work of the museum.

Springfield Convention of Ohio Engineers

While the attendance at the twenty-second meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers, which was held at Springfield, O., November 18 and 19, was not as flattering as at some of the previous gatherings, the amount of interest shown by those present and the entertainment provided by the citizens of Springfield were fully up to the standard of former years. A number of new members were taken in during the session, indicating a healthy growth of the organization.

M. C. Huyette, of Buffalo, presented an illustrated paper on "Automatic Smokeless Furnaces and Smoke Repression," in which a number of cases were cited showing the increased economy which resulted when proper means had been provided to eliminate smoke. Furnaces were classified as:

First. Overfeed, including all hand-fired furnaces and the "sprinkler" and "flipper" mechanical coal-feeding devices on the market.

Second. Front-feed, including all mechanical furnaces with grates inclined from the front toward the bridgewall, and all chain-grate and traveling-grate furnaces.

Third. Under-feed furnaces, where coal is forced upward, the coking of the coal and combustion of fixed carbon and volatile combustible taking place abóve the green coal.

Fourth. Side-feed, in which the coal is automatically fed at the sides of the furnace and provision is made for mechanically breaking up the coke and removing the ash and refuse to the ashpit.

It was pointed out that the cost of power is the first and important factor in the cost of any manufactured product, and is seldom known with any degree of accuracy. As an illustration of how power production may be brought down to a question of dollars and cents, the performance at a plant in the Union stock yards, Chicago, was given, where a battery of boilers, rated at 400 horsepower each, were worked at 11½ to 40 per cent. above rating, using Illinois screenings costing $1.10 per ton. The boilers gave an average evaporation from and at 212 degrees of 7.58 pounds of water per pound of coal, the cost of steam being $0.07 per 1000 pounds.

The financial damage caused by smoke was also taken up in detail, showing from the reports of smoke inspectors and others interested throughout the Middle West to what enormous figures this damage will run. In the city of Chicago it is estimated that this amounts to $50,000,000 a year or $20 *per capita.*

Another paper, entitled "The Smoke Situation in General," by George R. Bott, was presented on this important subject. The author spoke of the work now being done to prevent smoke and referred to the difficulties encountered in the small one-man plant. A remarkable case was then cited as to what can be done to improve the furnace conditions of a steam-boiler plant. The plant, consisting of three boilers, was located in New York State, and being unable to carry the load, those in charge decided to improve the conditions. A new 150-horsepower boiler and 20-foot additions to the stacks of the boilers were first thought of; however, the improvements were started by equipping the old boilers with shaker grates and an improved smokeless furnace using superheated steam. After some weeks of operation, the changes noted as having taken place were as follows: Coal at $1.50 per ton was used in place of coal costing $2.10 per ton; two men, firing the boilers, working twelve-hour shifts, had taken the place of six men, working two at a time on eight-hour shifts; the average steam pressure had risen from 50 to 80 pounds; steam blowers in the stacks were discontinued and no extensions were necessary; there was also a marked decrease in the amount of coal used.

"Vapor Heating Systems" was the subject of a paper by T. G. Monat, of Cleveland. Such a system was defined as a steam-heating system operating at a pressure of two to three ounces per square inch, with either a single- or double-pipe installation. No vacuum is employed, the air being expelled to the atmosphere by the pressure alone. The system is essentially one for residences or smaller-sized buildings, the advantages claimed for it being lower cost than a hot-water system, no danger from overpressure because of direct connection to atmosphere, no power required to remove the air, no noise in the system, greater economy because of the low temperature of the returns which are consequently in a condition to readily absorb a large amount of heat from the furnace.

W. E. Haswell, chief examiner of engineers for the State of Ohio, addressed the society on the necessity for a State boiler-inspection law and asked for its coöperation along this line. A resolution was passed to appoint a committee of five to confer with the State examiner's office, boiler inspectors and others interested and to lend the aid of the society toward the drafting and passing of a satisfactory law.

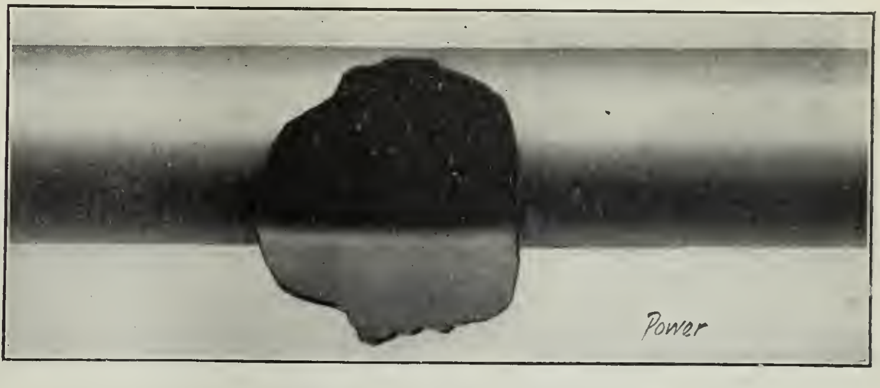
Oscar F. Rabbe, of Toledo, O., was re-elected president; H. L. Patterson, of Youngstown, was elected vice-president and J. J. Hoppes, of Springfield, and Dan Delaney, of Cincinnati, were elected new members of the board of managers. Charles P. Crowe, Ohio State University, Columbus, was retained as acting secretary.

The business sessions were held in the rooms of the Springfield Commercial Club, where everything necessary was provided for the comfort of the visitors. The entertainment program included a number of interesting automobile trips to the various industries of Springfield, the plants visited being the American Seeding Machine Company, Foos Gas Engine Company, Kelley-Springfield Road Roller Company, Kelley Motor Truck Company, Foos Manufacturing Company, Lagonda Manufacturing Company and Hoppes Manufacturing Company. On Saturday the members and visitors were given a luncheon at the Lagonda Club, as the guests of J. J. Hoppes, of the Hoppes Manufacturing Company.

Youngstown was chosen as the place of next meeting, the date of which will be May 18 and 19, 1911.

Boiler Tube Failure

At about 9.15 Monday evening, November 6, a tube in one of the boilers at the Gold street station of the Brooklyn Edison Electric Illuminating Company exploded, painfully, though neither seriously nor dangerously, scalding two men. The tube which failed was the second from the wall in the second row. This boiler had been "let down" during the afternoon for the purpose of putting a new gasket on one of the manhole covers in one of the drums, and at 5 o'clock a fresh fire was started. When the steam pressure had reached 160 pounds, or 20 pounds below the regular working pressure, the failure occurred.



Ruptured Section of Tube

The opening in the tube is of a half-oval shape, 4 inches long and 2 wide.

A part of the heated water flashed into steam as it poured out of the hole in the tube. Striking the grates the steam was deflected and driven directly across the 16-foot alleyway between the two rows of boilers, through the open door of the furnace of the boiler opposite. There were several men in this furnace, repairing the brickwork, and it was in attempting to escape by the door through which the steam and water were entering that they were scalded.

The tube was made of charcoal iron No. 10 gage, though at the point of initial rupture it was barely 1/32 inch thick. The boiler has been in use a few months over four years.

Annual Meeting of American Society of Mechanical Engineers

The thirty-second annual meeting of the American Society of Mechanical Engineers will be held in the Engineering Societies building, 29 West Thirty-ninth street, New York, December 6 to 9. Tuesday evening will be the occasion of the annual presidential address and of a reception by the president and the president-elect. The professional papers to be presented are unusual in variety and merit. On Wednesday morning, following the transaction of business, an account of the joint meeting in England will be given by the secretary, Calvin W. Rice; and there will also be a paper by George A. Orrok, of the New York Edison Company, on the transmission of heat in surface condensation. On Wednesday afternoon a session on steam engineering will be held, with a paper on carbon dioxide as an index to combustion, by E. A. Uehling, of the Uehling Instrument Company; two accounts of tests on steam turbines in the locality of San Francisco, one by S. L. Naphtaly, of the City Electric Company, and the other by F. H. Varney, of the Pacific Gas and Electric Company. Other related topics will also be presented. The reception by the local membership will be held on Wednesday evening instead of Thursday evening as heretofore.

On Thursday there will be three sessions: one in the morning upon miscellaneous topics; and in the afternoon two simultaneous sessions, a machine-shop meeting devoted to the subject of grinding with papers by C. H. Norton, of the Norton Grinding Company, W. A. Viall, of the Brown & Sharpe Manufacturing Company, and B. M. W. Hanson, of the Pratt & Whitney Company; and the meeting of the gas-power section, beginning with a paper by E. P. Coleman upon the blast-furnace gas-power installation of the Lackawanna Steel Company.

On Thursday evening there will be an address by Dr. Georg Kerschensteiner, superintendent of schools in Munich, on "Industrial Continuation Schools of Munich." Doctor Kerschensteiner has been foremost among educators in Europe in bringing industrial establishments into coöperative relations with the public-school system. So much importance is attached to this address that it is proposed to make the session a joint meeting with the National Society for the Promotion of Industrial Education, the American Institute of Mining Engineers and the American Institute of Electrical Engineers.

Friday will be devoted to excursions to points of engineering interest, thus making it possible for the out-of-town members to become more familiar with New York, as well as concluding the meeting in an informal way which should give opportunity for increased acquaintanceship in the society.

Correction

In the article on "Exhaust Steam Regenerators," page 1893, November 1 number, the author overlooked a clerical error in the statement that the first Rateau installation was made in 1890. The first installation was put in commercial use in 1902.

PERSONAL

C. W. Clarke has resigned his present position as steam engineer with the New York Central & Hudson River Railroad Company in order to accept an appointment with the Stone & Webster Engineering Corporation, of Boston, the change to take effect December 1, 1910.

NEW PUBLICATION

"The Effect of Keyways on the Strength of Shafts," by Herbert F. Moore, has just been issued as Bulletin No. 42 of the engineering-experiment station of the University of Illinois. This bulletin records the results of tests made to determine the relative strength of solid shafts and shafts with keyways. Various sizes of shafts were tested, and for each size of shaft the weakening effect of keyways of several proportions was determined. Tests were made on shafts subjected to twisting only and on shafts subjected to twisting and bending at the same time. The results show that the weakening effect of keyways of the usual proportions upon the strength of shafts is considerable, a square keyway of the usual size causing a reduction of about one-sixth in the strength of the shaft. Formulas and a diagram are given for determining the effect of keyways of various proportions. The effect of keyways for the Woodruff system of keying is discussed. The results of the tests are summarized in a table showing the power which can be transmitted by various sizes of shafts with keyways of usual proportions.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power and The Engineer

Vol. 32 NEW YORK, DECEMBER 6, 1910 No. 49

PRACTICAL ability undoubtedly is a grand thing and very valuable.

A man could read his eyes out on the subject, "How to be a Successful Swimmer," and yet drown like a rat in a submerged trap if he should attempt to put his theoretical or "book" knowledge into practice in any but shoal water.

The sane man does not live who would be willing to try to "turn the trick" in deep water.

Further, if a man could be induced to try, beyond the shadow of a doubt he would fail miserably no matter how hard he had studied to succeed. He would not have the knack and self-confidence necessary to make good.

The same thing applies, in variable degree, to practically every operation involving manual skill.

Take, for instance, the fairly simple "stunt" of stopping an engine, the novice—nervous and overanxious—would botch the job in the majority of cases. Yet, what little there is that could be written about the task could be learned in a very short time.

On the other hand, there are some things which, though easy to learn, are of little value to the performer unless he possesses an amount of theoretical or "reason-why" knowledge, such as is required to indicate an engine intelligently.

The average man could learn to take a diagram perfectly in a few hours. But—he could indicate engines "till the bench broke", and be but very little better off and no more valuable to himself or others unless he knew what the diagrams showed and could adjust the engines to operate more satisfactorily.

The theoretical part of the equipment which a first-class engineer must have is by far the more difficult to acquire.

There is no denying that systematic study is something of a hardship for most mortals.

There is no royal road to knowledge. The expenditure of money alone will not purchase the "right of way". Mental effort is required and perseverance as well.

That the efforts so directed secure handsome rewards is corroborated by the brilliant careers of countless men made possible simply because these men were given to the study habit.

Methodical study serves to discipline the mind and assist it to acquire the ability to reason. And reasoning is the highest type of mental activity and probably the rarest.

Have you the study habit?

If not, ask yourself, Why not?

It is never too late to learn!

Some Important Furnace Alterations

By A. D. Blake

By extending the furnace 30 inches the maximum capacity of each boiler was increased from 750 to 1000 horsepower. The changes necessitated the substitution of a suspended arch in place of the usual form of sprung arch.

Some interesting developments in furnace practice have taken place recently at the Long Island City power house of the Pennsylvania railroad. This plant is the central station which furnishes power for the electric trains of the Long Island railroad and for those of the Pennsylvania railroad's recent extension into New York City, which includes the tubes under both the Hudson and the East rivers.

The original installation included thirty-two Babcock & Wilcox boilers each, including superheaters, having a rated capacity of 650 horsepower and a maximum of about 750 horsepower, the rated capacity being based upon 10 square feet of heating surface per boiler horsepower. These furnished steam to three 5500-kilowatt Westinghouse-Parsons turbines which carried the train load for the Long Island railroad. To meet the increased demands for power subsequent to the extension of the Pennsylvania railroad it was decided to install four additional units, two of 8000 and two of 3000 kilowatts capacity. This required more boiler capacity, but instead of increasing the number of boilers, the capacities of those already installed were increased. Accordingly, the grate area of each boiler was enlarged 20 per cent. by extending the furnace 30 inches in front and giving it a dutch-oven effect. Also, the Roney stokers were replaced by larger ones of the same kind, but having the improved form of grate. By this means the working capacity of each boiler was increased to about 1000 horsepower.

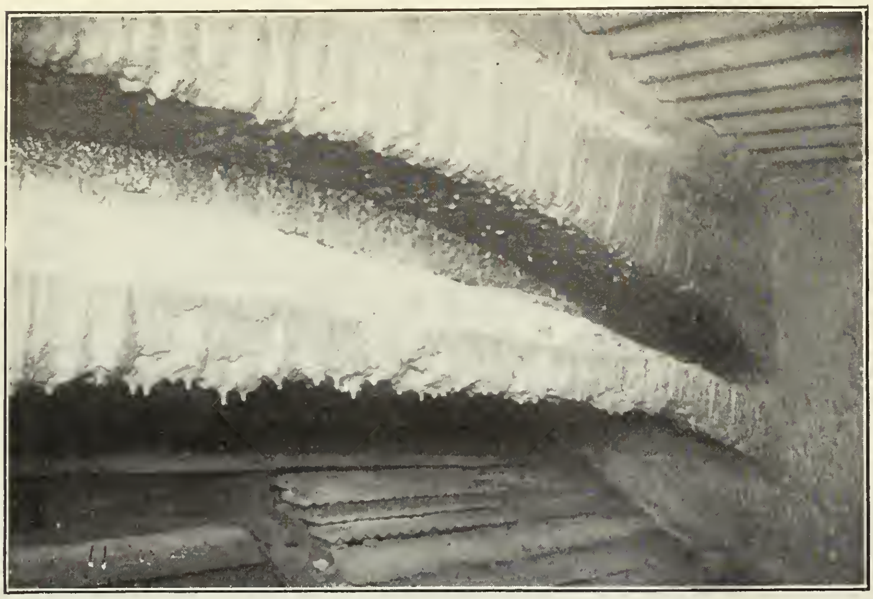


FIG. 1. FAILURE OF SPRUNG ARCH UNDER EXCESSIVE TEMPERATURE

These furnace changes, however, involved certain difficulties. By extending the furnace fronts, in many cases a steel building column was brought directly within the fire space. To prevent overheating, each column was inclosed within a cast-iron casing, between which there is considerable air space, and the latter was sheathed with asbestos board, which, in turn, was protected by firebrick. Measurements taken in the air space surrounding the columns have shown that this method of protecting the columns from the heat of the fire is entirely adequate.

The second difficulty encountered was due, more directly to the new type of grate. In this the slots between adjacent fuel plates allow much more air to pass through the fire than with the old type, hence a much hotter fire results. As a consequence, it was soon discovered that the sprung firing arches would not hold up for any length of time under the increased temperatures but began to fuse and fall as shown in Fig. 1.

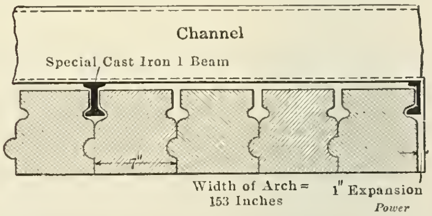


FIG. 3. METHOD OF SUSPENDED FLAT ARCH

The engineers accordingly set about to devise a means for overcoming this trouble and finally decided upon installing a suspended arch in place of the regular

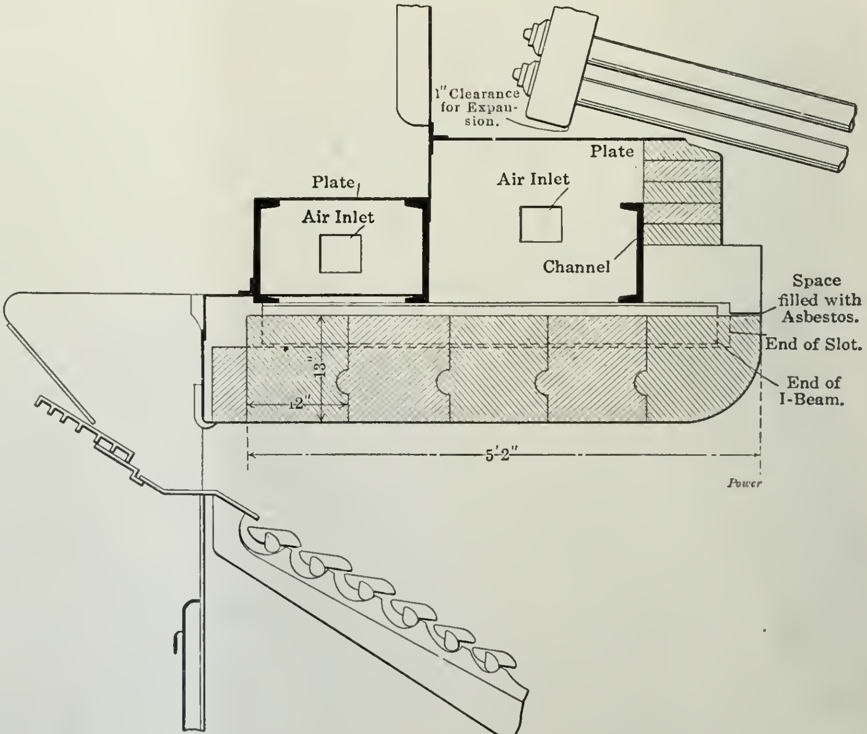


FIG. 2. SECTION THROUGH FURNACE

sprung arch. This is a straight arch made up of especially formed tile hung from special cast-iron I-beams. The latter are prevented from becoming overheated by the circulation of air through spaces over the arch, the air entering through rectangular holes in the sides of the boiler setting. It has been found that, even when forcing, the temperature of these supporting beams is not excessive. Some of these new arches have been in service since last February and have held up remarkably well. Fig. 2 is a cross-section through the furnace showing the arrangement of the new arches and the air space above, while Fig. 3 shows how the tile blocks fit over the I-beams. In Fig. 4 is shown one of these arches shortly after being installed.

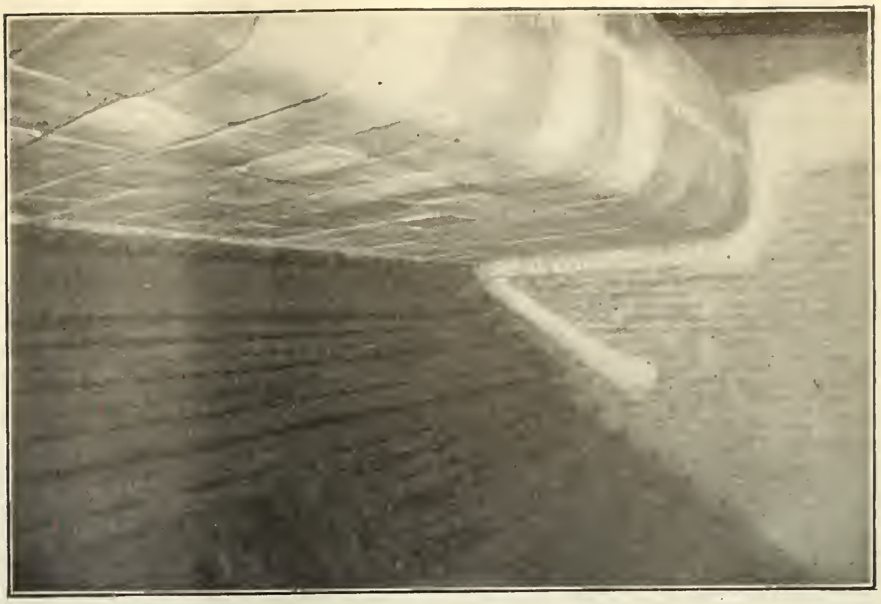


Fig. 4. One of the New Arches in Place

Bituminous coal is used, having a heat value of 14,500 B.t.u. and an average composition of,

	Per Cent.
Volatile matter	18.50
Ash	8
Sulphur	1.75
Moisture	1.50
Carbon	70.25

Natural draft is employed with steel stacks 275 feet high, and when running at maximum capacity with a draft of ⅝ inch of water over the fire and 1 inch at the base of the stack, as high as 45 pounds of coal have been burned per square foot of grate surface per hour.

An interesting fact in connection with the furnace changes is brought out by an examination of the results of the flue-gas analyses, these showing a marked increase of CO_2 with the new arches and stokers. The average for a number of samples taken before the changes were made showed 13.9 per cent. of CO_2 and the average for the same number of samples taken after the changes were effected showed 14.9 per cent. of CO_2, a gain of 1 per cent.

Graphite and Its Manufacture

By James J. Jenkins

To clear up misconceptions that have long existed about graphite the author tells of the great uses found for it in the fields of lubrication, electrochemistry and elsewhere.

Carbon is one of the most mysterious elements with which man has to deal. Diamond, lampblack and graphite are three allotropic forms of this element and it is very difficult to find an explanation of the difference between these three forms. Still, it is known that there is a very great difference.

Graphite has been known to humanity since the latter part of the fourteenth century. Excavations which opened graves of the ancients brought to light knowledge that they knew of some of the properties of graphite, but they left no record of their understanding. It seems that their recognition of its true nature was very slow, possibly because they had little need for a mineral possessing its properties. In reviewing the history of graphite, it is somewhat astonishing to learn that even to this day it is associated in some minds with lead as being "Black Lead" or "Plumbago," which names might very well have been discarded away back in 1779 when Karl Wilhelm Scheele, of Kopen, Sweden, learned the true chemical nature of graphite, or before that date when Heinrick Pott, a German chemist, demonstrated that graphite contains no lead. Considering the present-day use of graphite in making lead pencils, it is very clear that this form of carbon is well named "Graphite," a word taken from the Greek and which means "I write."

It is probable that the diamond has always had more in it of general interest than graphite, but, considering the extended variety of present-day uses of graphite and the service it renders the world in many fields, there can be no question which of these two forms of carbon is of the highest intrinsic value. The diamond, in its sparkling clearness, is fascinating in the extreme, especially when it graces the hair, neck or hand of an engineer's wife, daughter or other loved one, but the faithful engineer who day after day and year after year battles with his chief foe—Friction—is well aware that the black, lusterless form of carbon, known as graphite, is his true friend at those times when he needs an efficient lubricant.

While graphite has been known to man for centuries, it is worthy of note that in 1909 we passed the fiftieth anniversary of the drilling of the first petroleum oil well by Colonel Drake on Oil creek, Penn. Up to that time, the world depended upon sperm oil, lard oil and a few other mediums for its lubricants. It is certain, considering the great modern installation of machinery, that engineers would find it impossible to return to these methods of lubrication. Today the world is in full appreciation of petroleum oils and greases as lubricants.

It has been said that the life of the supply of petroleum oils is less than the years allotted to man, which is three score and ten. Assuming this conclusion to be correct, it is evident that unless a lubricant be found competent to take the place of this oil, the industrial world must stop; the great factories must cease to operate. Transcontinental and other trains must cease to run and the great steamships will no longer speed across the ocean. Lubrication is so essential that it is appalling to consider the results which would surely develop if the world were not provided with a lubricant of equal or greater merit, for, even if the supply were sufficient, it would be utterly impossible for man to operate his great modern installations of machinery with the crude lubricants of years ago. Man accepted petroleum oil and oil products as an advance over sperm-oil lubrication, a fact which should make it easier for him to realize that in graphite, as it is today available, he has at his command a lubricant which, it would seem, is destined to be a lubricant of future time and ages, inasmuch as the supply can be made such as to meet all present and future demands.

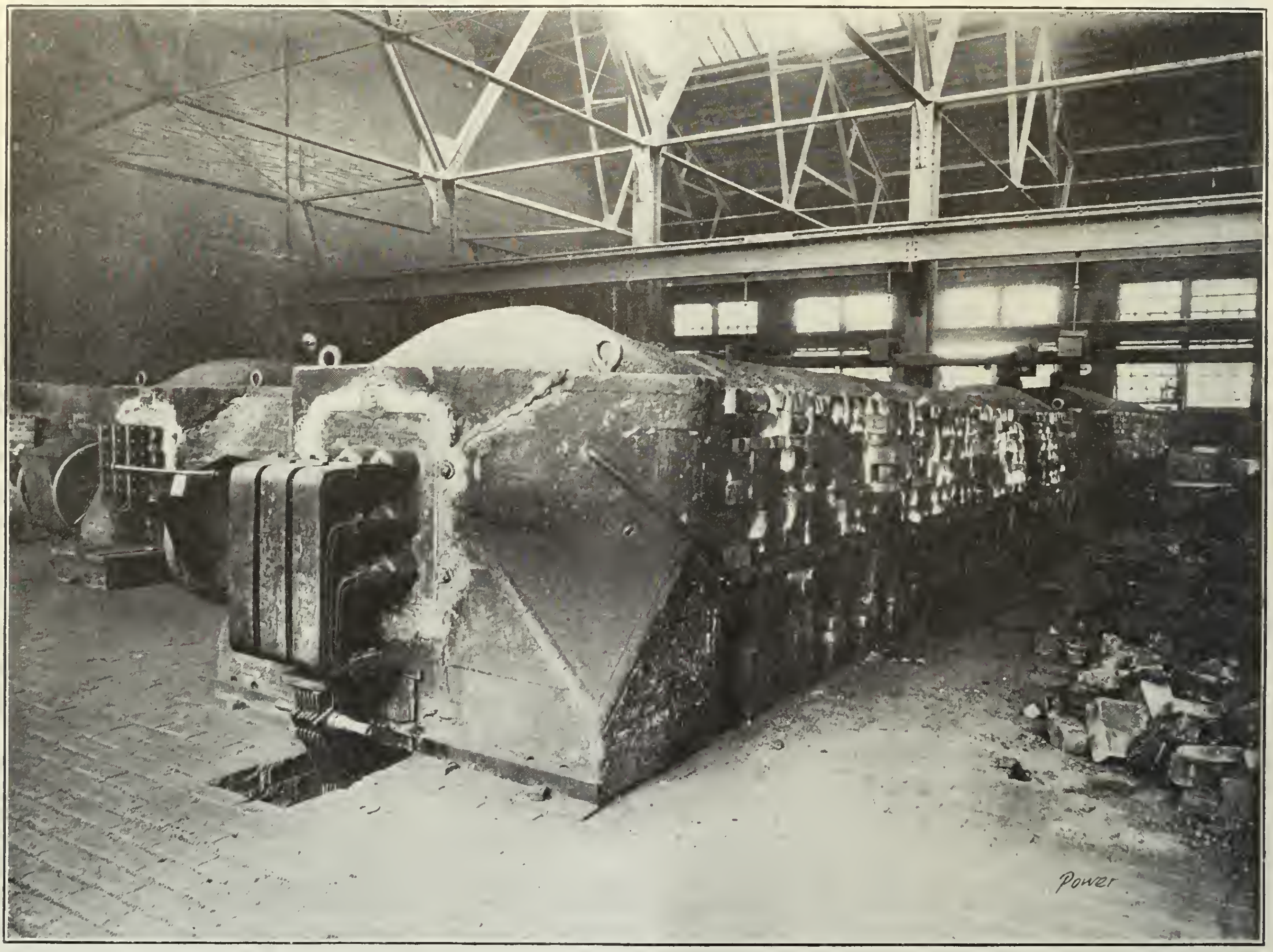


Fig. 1. Electric Furnace in which Pure Graphite Is Made Synthetically

We are coming to know more about graphite and its value every day. In the *Scientific American* there was an account of a new form of graphite that is very fluffy and of a greatly expanded nature, which graphite is said to have been developed from the so called crystalline graphite, in its new form lacking the luster of the flake and being extremely fine. There are graphite mines in Austria, Bavaria, Ceylon, Canada, England, Italy, Mexico, Siberia, the United States and other countries, and scientists have found that meteorites which have dropped to earth out of the sky, both in this country and in Europe, brought graphite with them. Many tons of graphite are imported annually by the United States, indications being that the consumption is on the increase. The presence of all of these deposits indicates that nature was very generous in her distribution of this mineral. She made it in two varieties, as we understand them, one of which is amorphous; the other is the so called crystalline variety. Generally speaking, graphite has a world of uses, and there are many applications where it is still used as "Black Lead" or "Plumbago."

And now, man has learned how to make graphite, a fact which supports the conclusion that destiny is working itself out and that, having gained this knowledge, man has simply placed himself in a position to meet the requirements of the consumption of this mineral as its value becomes better known. This conclusion is based on the supposition that as the world grows, as it progresses, there is revealed a way whereby men may assist one another with their knowledge for the general good of all mankind. It is not known how or when nature made graphite. Man makes it by the application of tremendous temperatures to raw materials he places in an electrically operated furnace. Nature has no choice of raw materials whereas man is more fortunate, having the world from which to select his.

In the electric-furnace process it has been found possible to impart certain definite chemical and physical properties to the graphite made. This product may be in the form of rods, tubes or plates, for use as electrodes or as lump graphite, this latter to be ground later to the form of impalpable powder for use in the various fields which it enters.

Not only has man learned how to make graphite that is practically chemically pure, but he has gone further into the art and invented a process whereby graphite is "deflocculated," as the inventors call it, signifying the supposed breaking asunder of the compound particles into the smallest particles in which the substance can exist while retaining its identity. Whether or not the graphite is reduced to the molecular or collodial state, it will remain suspended in oil or water, and passes with it through fine filtering paper. This process marks a very notable advance in the application of graphite to the field of lubrication and is that modern process above referred to as developing a modern lubricant for modern machinery.

In the first 50 years of the use of petroleum oils as lubricants, the warning was sounded that it is necessary to conserve the world's resources in this field. Man was never without the facilities of accomplishing those things the most progressive minds called for, and it is more than possible that in the new discoveries relative to graphite, the world is being prepared for methods which will afford greater endurance, higher efficiency and greater economy in the output of the magnificent installations of machinery man has built for doing the world's work.

In this connection it is instructive to review the synthetic production of graphite. There can be no doubt but that the rapid increase in the use of graphite and the consequent decrease in the supply of the natural product make man's ability to produce graphite of great value to the world. When the hundreds and thousands of chemical changes through which men are constantly putting the elementary bodies and the very prominent part taken by carbon in these reactions are considered, it would be astonishing if at some point in all this work man had not learned how to produce graphite superior to nature's. In the 1890 edition of Watt's "Dictionary of Chemistry," Mr. Muir mentions six recognized methods:

(1) By heating charcoal with molten iron, and dissolving out the iron with hydrochloric and nitric acids.

(2) By the slow decomposition of hydrocyanic acid, and boiling the product with nitric acid.

(3) By evaporating the mother liquors obtained in making soda. These contain cyanogen compounds which are decomposed at a certain concentration of the liquid with formation of ammonia and graphite.

(4) By leading carbon monoxide over ferric oxide at 300 to 400 degrees Fahrenheit.

(5) By the decomposition of carbon disulphide at high temperature, in contact with metallic iron.

(6) By leading carbon tetrachloride over molten pig iron.

It is doubtful if any of these methods could be developed to a commercial success. It was stated years ago that Cowles Brothers, of Cleveland, O., made report that sometimes they found that graphite occurred in or about the charge of an electric furnace they used. In 1872, F. Rose discovered a method of making graphite by exposing a cut diamond "bedded in charcoal" to a temperature equal to molten cast iron, whereby the surface is said to have become coated with graphite. Considering the value of diamonds, and the esteem in which they are usually held, it is a safe conclusion that this method will never become popular. In the 1890 edition of the "Encyclopædia Britannica" it is stated that "by heating to the high temperature afforded by a powerful galvanic battery both the diamond and amorphous carbon are converted into graphite." However, that diamonds will change to graphite when protected from chemical influences, under the influences of high temperature, is true, and, possibly, is not more than what might be expected from other known facts, but that pure, amorphous carbon will change to graphite as a result of heating is not proved.

Practically no progress of value had been made in the commercial production of graphite until the world was enriched to a wonderful extent by extended years of scientific research on the part of Dr. Edward G. Acheson. It was he who created carborundum, and, early in the history of the manufacture of that now most popular abrasive, he discovered that when carborundum was heated to a very high temperature decomposition occurred, the contained silicon passing off as vapor and a beautiful graphite left as a pseudomorph, or, as one might say, a skeleton of the original carborundum crystal. It was this discovery that started him on a new line of thought, the result of which was that after a long series of experiments the methods now employed in the making of graphite artificially were perfected.

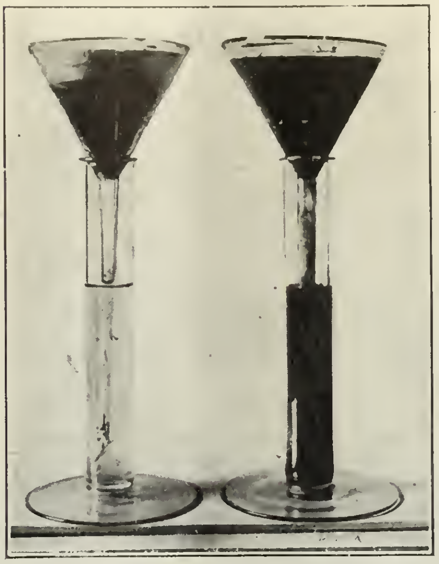


Fig. 2. Flocculated and Deflocculated Graphite

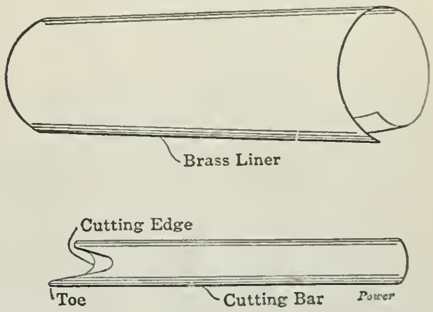
It is worthy of note in this connection that the first commercial graphitic products of the electric-furnace process, as used at Niagara, were in the form of electrodes for electrochemical work. Only recently a gentleman who occupies a very eminent position in the field of electrochemistry stated, "It may safely be asserted that electrochemistry owes its astonishing modern growth, first to the development of the dynamo and second to the production of Acheson-graphite electrodes." Under these circumstances we are led to conclude that while the production of graphite in electrode form has been so vastly beneficial to the field of electrochemistry, the production, through the operation of the same process, of practically chemically pure graphite is destined to be of equal value to the world as a lubricant.

By one process graphite artificially made is disintegrated by mechanical means, and in this form the graphite is valuable as a lubricant, or can be mixed with grease to serve as a carrier. By another process this graphite made artificially is deflocculated to such a wonderful state of fineness that it is practically molecular and will remain suspended in oil or water, the liquids serving as a carrier for the graphite. These are the things which offer recognition of the fact that as men have gone along planning and making machinery that is truly wonderful, another man, wholly unmindful of what these other men were doing, has dug out scientific secrets which go very far in making the operation of these machine installations safe through the perfect lubrication now possible through the use of synthetically produced graphite in the various forms in which it is now available.

Bar for Cutting Liners

By George J. Little

Some time ago I had occasion to remove the brass liners in a boiler-feed pump, and they were too tight to be removed without splitting. It was evident that if a cold chisel were used to split them it would be liable to damage the pump unless handled with utmost care. Therefore, I had a cutting bar made similar to that shown in the sketch. The toe at the end of the bar lifts the brass and directs it against the cutting edge. When this bar is made in the right way it causes the liner to curl so that it may be removed by hand.



Bar and Liner Showing Edge Turned Up by Toe of Bar

They hed a nigger paintin' th' 60-foot smokestack et th' power house a spell ago an' th' cuss fell off'n th' top uv it, landin' on his hed on th' cement pavement. Th' jedge fined 'im $10 fer destroyin' city property, an' th' company docked 'im 15 cents fer th' time he lost gettin' back up th' stack. Th' coon says th' only thing he feels sore 'bout is thet he didn't land on his feet 'stid uv his hed so's he cud hev brot suit fer damages agin th' company fer personal injuries he shore would haf got.

Compressed Air Transmission Tables*

By Frank Richards

A set of tables, based upon the generally accepted formula for the flow of compressed air through pipes, and showing the loss in head for different initial pressures and rates of flow with various lengths and diameters of pipe.

The accompanying tables of pressure losses during the flow of compressed air through pipes were computed by what may be termed the Rix-Johnson formulas. J. E. Johnson, Jr., gave Church's familiar formula in a simplified form in the *American Machinist,* July 27, 1899, which was, in substance, as follows:

$$R = \frac{K\,V^2\,L}{D^5}$$

where

V = Volume of free air in cubic feet per minute;

L = Length of pipe in feet;

D = Diameter of pipe in inches;

K = A numerical constant, which Johnson fixed as 0.0006;

R = Difference between the squares of the initial and terminal absolute pressures in pounds per square inch; that is, $p^2_1 - p^2_2$.

I have reason to believe that this formula is now used in this country more generally than any other in practical compressed-air work.

E. A. Rix, of San Francisco, uses Johnson's formula with 0.0005 for the value of K. In my own practice I have found it more convenient to use the reciprocal of this value of K, that is, 2000, and transfer it to the divisor, where in actual numerical operations it almost invariably disappears at once by cancelation. I also find it more convenient to use small letters instead of capitals; hence my working formula takes the form,

$$p_1^2 - p_2^2 = \frac{v^2\,l}{2000\,d^5}$$

This, of course, may be transformed to suit the special requirements, as:

$$l = \frac{2000\,d^5\,(p_1^2 - p_2^2)}{v^2}$$

$$v = \sqrt{\frac{2000\,d^5\,(p_1^2 - p_2^2)}{l}}$$

$$d^5 = \frac{v^2\,l}{2000\,(p_1^2 - p_2^2)}$$

Computations in this line do not invite, nor hardly permit, micrometrical precision, and refinements are out of place; hence it is quite permissible and very convenient to use 15 pounds for the normal atmospheric pressure, and this has been done in computing the tables herein given. A single example of the process will suffice.

Let there be 5000 feet of 8-inch pipe, through which it is desired to transmit 4000 cubic feet of free air per minute at an initial pressure of 105 pounds gage, or 120 pounds absolute. What will be the terminal pressure and the loss of head? The pressure here assumed is not unusual in the best practice of the present day.

Note that d^5 (8^5) is 32,768, and p_1^2 (120^2) is 14,400. By substituting these values the formula may be expressed as follows:

$$\frac{4000^2 \times 5000}{2000 \times 32{,}768} = p_1^2 - p_2^2 = 14{,}400 - p_2^2$$

Then,

$$p_2^2 = 14{,}400 - 1220 = 13{,}180$$

and

$$p_2 = 114.80 \textit{ pounds}$$

absolute terminal pressure; hence the loss of pressure is,

$$120 - 114.80 = 5.20 \textit{ pounds.}$$

The foregoing example shows what may be considered a suitable rate of pipe transmission, or a flow which should not be much exceeded in practice. The free air in this case being 4000 cubic feet per minute, and the initial absolute pressure being 8 atmospheres, the actual volume, assuming that aftercoolers are used and that the air is at normal temperature, will be only 500 cubic feet per minute. The volume content of an 8-inch pipe is 0.349 cubic feet per foot of length; therefore the rate of flow will be

$$500 \div 0.349 = 1432 \textit{ feet}$$

per minute. A handy limit figure to keep in mind is the round number 1500 feet per minute.

The loss of pressure will be a little more than proportional to the squares of the volumes of free air. That is, if in this case the volume of free air had been

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*Copyright, 1910, by Frank Richards.

LOSS OF HEAD IN TRANSMISSION THROUGH PIPES.

1-INCH PIPE.

Initial Pressure, 75 Pounds, Gage.						Initial Pressure. 90 Pounds, Gage.						Initial Pressure, 105 Pounds, Gage.					
Free Air per Minute, Cubic Feet.	Flow in Pipe per Minute, Feet.	Length of Pipe in Feet				Free Air per Minute, Cubic Feet.	Flow in Pipe per Minute, Feet.	Length of Pipe in Feet.				Free Air per Minute, Cubic Feet.	Flow in Pipe per Minute, Feet.	Length of Pipe in Feet.			
		100	200	300	400			100	200	300	400			100	200	300	400
12.5	385	0.05	0.09	0.13	0.17	12.5	330	0.04	0.08	0.11	0.15	12.5	289	0.03	0.06	0.10	0.13
25	771	0.18	0.35	0.53	0.70	25	661	0.15	0.30	0.45	0.59	25	578	0.12	0.25	0.40	0.52
50	1543	0.69	1.40	2.11	2.83	50	1322	0.59	1.19	1.81	2.41	50	1157	0.52	1.05	1.56	2.10
75	2314	1.58	3.18	4.82	6.49	75	1984	1.35	2.72	4.10	5.51	75	1736	1.10	2.25	3.40	4.68
100	3086	2.82	5.74	8.76	11.89	100	2645	2.41	4.88	7.41	10.00	100	2313	2.11	4.25	6.42	8.65

1½-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		250	500	750	1000			250	500	750	1000			250	500	750	1000
25	339	0.06	0.12	0.18	0.23	25	291	0.05	0.10	0.15	0.20	25	254	0.04	0.08	0.12	0.17
50	679	0.23	0.46	0.69	0.92	50	582	0.19	0.39	0.59	0.79	50	510	0.17	0.33	0.50	0.68
100	1358	0.92	1.85	2.79	3.74	100	1164	0.79	1.58	2.38	3.18	100	1019	0.69	1.38	2.07	2.76
150	2037	2.08	4.22	6.39	8.64	150	1746	1.78	3.59	5.43	7.30	150	1528	1.53	3.14	4.65	6.32
200	2716	3.74	7.64	11.73	16.06	200	2328	3.18	6.47	9.87	13.39	200	2037	2.76	5.55	8.51	11.54

2-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		200	500	1000	2000			200	500	1000	2000			200	500	1000	2000
100	764	0.17	0.43	0.87	1.75	100	655	0.15	0.37	0.75	1.50	100	573	0.13	0.32	0.65	1.30
150	1146	0.39	0.98	1.97	3.99	150	982	0.34	0.83	1.69	3.40	150	859	0.29	0.74	1.47	2.95
200	1528	0.69	1.75	3.54	7.24	200	1310	0.59	1.51	3.02	6.48	200	1146	0.53	1.32	2.64	5.28
250	1910	1.09	2.31	5.60	11.60	250	1637	0.93	1.97	4.76	9.75	250	1433	0.83	2.09	4.17	8.35
300	2292	1.57	4.92	8.18	17.28	300	1965	1.35	3.45	6.92	14.37	300	1719	1.22	3.06	6.13	12.25

LOSS OF HEAD IN TRANSMISSION THROUGH PIPES.

2½-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		1000	1500	2000	2500			1000	1500	2000	2500			1000	1500	2000	2500
100	489	0.29	0.44	0.59	0.73	100	419	0.25	0.38	0.51	0.62	100	367	0.22	0.32	0.43	0.55
200	978	1.16	1.74	2.32	2.89	200	838	0.98	1.47	1.96	2.47	200	734	0.86	1.29	1.73	2.15
300	1467	2.6	3.91	5.25	6.64	300	1257	2.22	3.31	4.45	5.64	300	1100	1.94	2.88	3.87	4.95
400	1956	4.64	6.96	9.30	12.20	400	1676	3.98	6.00	8.00	10.25	400	1467	3.45	5.18	7.00	8.86
500	2450	7.50	10.00	15.10	18.90	500	2100	6.25	9.38	12.56	15.72	500	1838	5.45	8.20	11.15	14.17

3-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		1000	1500	2000	2500			1000	1500	2000	2500			1000	1500	2000	2500
100	339	0.12	0.17	0.23	0.29	100	291	0.10	0.15	0.20	0.25	100	254	0.09	0.13	0.17	0.22
200	679	0.46	0.69	0.92	1.15	200	582	0.39	0.59	0.79	0.99	200	509	0.35	0.53	0.69	0.87
400	1358	1.86	2.79	3.75	4.69	400	1164	1.58	2.39	3.19	3.99	400	1019	1.41	2.12	2.76	3.48
600	2037	4.21	6.39	8.65	10.81	600	1746	3.59	5.43	7.31	9.14	600	1528	3.13	4.83	6.48	8.16
800	2716	7.64	11.74	16.11	20.15	800	2328	6.47	9.87	13.43	16.79	800	2037	5.71	8.72	11.78	14.65

4-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		1000	1500	2000	2500			1000	1500	2000	2500			1000	1500	2000	2500
250	477	0.17	0.26	0.34	0.43	250	409	0.15	0.24	0.29	0.37	250	358	0.12	0.19	0.26	0.32
500	955	0.68	1.02	1.37	1.71	500	818	0.58	0.82	1.17	1.47	500	716	0.51	0.76	1.02	1.28
750	1432	1.54	2.32	3.10	3.89	750	1227	1.31	1.98	2.65	3.32	750	1074	1 15	1.73	2.31	2.89
1000	1910	2.76	4.17	5.60	7.06	1000	1637	2.35	3.55	4.76	5.98	1000	1433	2.05	3.09	4.14	5.20
1250	2387	4.34	6.52	8.91	11.29	1250	2046	3.49	5.59	7.53	9.51	1250	1790	3.22	4.86	6.53	8.22

6-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		1000	2500	5000	10,000			1000	2500	5000	10,000			1000	2500	5000	10,000
1000	849	0.36	0.89	1.81	3.65	1000	727	0.31	0.77	1.54	3.10	1000	637	0.27	0.67	1.34	2.70
1500	1273	0.81	2.04	4.12	8.46	1500	1091	0.63	1.74	3.51	7.15	1500	955	0.61	1.51	3.04	6.16
2000	1698	1.44	3.64	7.45	15.64	2000	1454	1.23	3.10	6.31	13.05	2000	1274	1.08	2.70	5.46	11.19
2500	2122	2.26	5.75	11.93	26.14	2500	1817	1.93	4.89	10.07	21.25	2500	1591	1.69	4.27	8.69	18.12
3000	2547	3.28	8.45	17.86		3000	2181	2.79	7.14	14.84		3000	1910	2.43	6.16	14.55	27.05

8-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		1000	2500	5000	10,000			1000	2500	5000	10,000			1000	2500	5000	10.000
2000	954	0.34	0.85	1.71	3.46	2000	818	0.29	0.73	1.47	2.95	2000	716	0.26	0.64	1.28	2.57
3000	1432	0.77	1.92	3.89	7.97	3000	1227	0.66	1.64	3.32	6.75	3000	1074	0.57	1.44	2.90	5.87
4000	1909	1.37	3.46	7.06	14.77	4000	1636	1.17	2.95	5.98	12.35	4000	1432	1.02	2.57	5.20	10.65
5000	2386	2.15	5.46	11.29	24.51	5000	2045	1.83	4.64	9.51	20.06	5000	1790	1.60	4.04	8.23	17.61
6000	2864	3.10	7.98	16.83		6000	2454	2.65	6.75	14.00		6000	2148	2.31	5.86	12.05	26.16

10-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		2500	5000	7500	10,000			2500	5000	7500	10.000			2500	5000	7500	10,000
1000	313	0.06	0.14	0.21	0.28	1000	268	0.05	0.12	0.18	0.24	1000	235	0.05	0.11	0.16	0.21
2000	627	0.27	0.56	0.84	1.12	2000	537	0.24	0.48	0.72	0.95	2000	470	0.21	0.42	0.63	0.84
4000	1254	1.11	2.25	3.37	4.56	4000	1075	0.95	1.92	2.88	3.88	4000	941	0.84	1.68	2.53	3.38
6000	1881	2.52	5.15	7.73	10.62	6000	1612	2.16	4.38	6.57	8.95	6000	1411	1.89	3.81	5.76	7.75
8000	2508	4.54	9.37	14.06	20.06	8000	2150	3.86	7.82	11.73	16.54	8000	1881	3.38	6.87	10.46	14.17

12-INCH PIPE.

		Length of Pipe in Feet.						Length of Pipe in Feet.						Length of Pipe in Feet.			
		2500	5000	7500	10,000			2500	5000	7500	10,000			2500	5000	7500	10,000
2,500	530	0.17	0.35	0.52	0.68	2,500	455	0.15	0.30	0.45	0.59	2,500	398	0.13	0.27	0.39	0.53
5,000	1061	0.70	1.41	2.11	2.84	5,000	910	0.62	1.21	1.81	2.42	5,000	796	0.53	1.05	1 58	2.11
7,500	1591	1.59	3.15	4.73	6.45	7,500	1365	1.35	2.69	4.03	5.53	7,500	1193	1.18	2.37	3.59	4.81
10,000	2122	2.83	5.77	8.66	11.95	10,000	1820	2.40	4.89	7.34	10.05	10,000	1591	2.11	4.27	6.46	8 69
12,500	2652	4.45	9.19	13.79	19.59	12,500	2275	3.79	7.76	11.64	16.19	12,500	1989	3.32	6.73	10.25	13.88

doubled, making 8000 cubic feet instead of 4000 cubic feet, the loss of head would have been 22.44 pounds instead of 5.20 pounds.

It is worth while to note how the pressure loss is diminished as the pressure is increased, due to the reduction of volume. Thus, in 1000 feet of 1-inch pipe, transmitting 50 cubic feet of free air per minute, the pressure losses for diminishing initial pressures would be as follows:

Initial Pressures.	Loss of Press.
45 pounds gage	11.52 pounds.
60 pounds gage	8.86 pounds.
75 pounds gage	7.24 pounds.
90 pounds gage	6.14 pounds.
105 pounds gage	5.33 pounds.
120 pounds gage	4.71 pounds.
135 pounds gage	4.23 pounds.
150 pounds gage	3.83 pounds.

Of course, no tables can be compiled which will cover the various requirements of compressed-air practice, but the figures herein given may at least furnish a working idea of the probabilities, and may be of service in a general way in preliminary estimates, or may serve to detect errors or inconsistencies which are apt to occur in the most careful figuring. No precise agreement with actual practice can be expected, as conditions which affect the result are so numerous

Factors Affecting Plant Costs

By H. J. Macintire

A few suggestions as to the factors which determine the location of a steam plant and the selection of its equipment. A graphical method is shown whereby the most economical cutoff for a given engine may be determined.

In new construction the fact is often overlooked that local conditions govern the choice of the equipment. Therefore, in choosing the equipment for a plant, it is well to consider the following items:

First Cost of the Plant—This includes the cost of engines, generators and auxiliaries, the building itself, chimneys, the construction for the storing and handling of coal and ash removal, and the cost of land, right of way, etc.

Depreciation—This includes not only the allowance for wear and tear but also takes into account the fact of the machinery becoming obsolete. The latter has an important bearing in certain types, and the question of obsoleteness may make it necessary to replace immediately the old machinery with that of new design. Although it is proper to charge only 4 per cent. for the yearly depreciation on the entire plant (this varies from 3 per cent. for the building to 7½ per cent. for the generators and 10 per cent. for the boilers), yet the allowance for the uncertain factor of obsoleteness will make the total rate about 8 per cent.

Cost of Maintenance and Supplies—This includes packing, waste, oil and repairs, the latter differing with different types of engines.

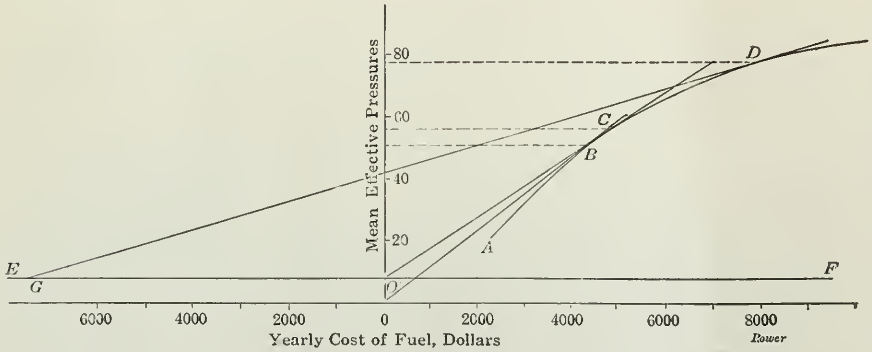
Fuel—This is often the factor of greatest weight. In the coal regions it is of less importance, but in those districts which require a long railroad haulage of fuel it becomes of considerable importance. In fact, the cost of the fuel determines the kind of engine that is practical, for a more expensive engine which can be run with economy is the better one to use where the cost of fuel is high, than a cheaper one using more steam. The three following factors govern the choice of design:

The cost of fuel per ton.

The number of hours of operation per year.

The load factor.

If power is used only a few hours a day the fuel cost can be practically disregarded and attention directed toward keeping the first cost as low as possible. A simple engine may give good economy at full load, but very poor at light loads. Therefore the load factor plays an important part in the selection of the engine.



Graphical Representation of Costs

Degree of Duplication of the Plant—In public utilities it is absolutely necessary that no shutdown occur, whereas in a mill or factory a shutdown, while undesirable, is not so important. Where everything is in duplicate there is a tendency toward employing cheaper engines, because of the smaller first cost. Also, it is often possible to use somewhat obsolete machines for emergencies, and have the uptodate and economical ones for the everyday work; this, however, is not so applicable to entirely new plants.

Utilization of Exhaust Steam—In steam heating, or in the absorption system of refrigeration where all the exhaust is used, nearly all the heat of the steam can be utilized; but in the condensing type it is possible, at best, to obtain an efficiency of only 15 to 20 per cent. In the first case a high-speed noncondensing engine, using 40 pounds of steam per horsepower-hour, can be employed; here the first cost is small and yet the economy is great. But in the case of the condensing engine the economy must be in the engine itself, for most of the heat carried to the condenser is lost.

Cost of Condensing Water—If the plant is on a river, lake or other body of water this is unimportant, but when not so located and especially when the plant is in a city or on the plains of the West, the cost of the condensing water becomes a governing factor. If a cooling tower is used, the interest on the investment and the cost of power for the fan are considerable.

Minor Factors—These include the location where absolute safety of the plant is important, as in office buildings, etc.

It is now a common custom to build double-decked power stations, putting the turbine directly over the boilers, thereby saving in piping and in floor space. This is possible with turbines because of the uniform turning moment and perfect balance of the moving parts. With such an arrangement a barometric tube condenser can be easily and cheaply employed.

To summarize, it may be said that where fuel is cheap, an inexpensive and uneconomical form of boiler and engine may be used, and where land is expensive it may be necessary to use water-tube boilers and turbines arranged on the double-deck plan, or to use high-speed engines. In pumping plants, or where power is used 24 hours a day and 365 days a year, the fuel is the most expensive item and an economical and necessarily expensive form of engine and boiler should be chosen.

Sometimes it is a question as to the most desirable method of operation. Professor Jacobus found in a certain engine test (American Society of Mechanical Engineers, Volume 24) that one-half load was the most economical, and that there was a saving of 5 per cent. of the coal (12.1 pounds of steam as against 12.76 pounds per indicated horsepower per hour) in operating at one-half load. The plant used 2000 tons of coal per year, which at $4 per ton amounted to $8000. A 5 per cent. saving of this sum amounted to $400 a year.

The cost of doubling the size of the engine to be able to run at one-half load would have been $22,000 and 5 per cent. interest on this additional first cost would have been $1100. Hence,

$$\$1100 - \$400 = \$700$$

loss due to the installation of the larger engine so as to run at the more economical load. Of course there is the item of the increase of power should it ever be desired, but the question of depreciation and obsoleteness would then have to be considered. It would be unwise in most cases to install much more than the required size of engine unless there is a reasonable possibility of more power being required later on.

There is a graphical method of representing the costs under various load factors which is very interesting. It has been shown that the steam consumption is not the only important item, but, on the other hand, the only thing that varies directly with the load is the fuel (except minor items, such as waste, lubricating materials, etc.). Suppose then that the mean effective pressure is plotted against the cost of fuel per year, the result will be a curve $A\,B\,C\,D$ as shown. The tangent $O'B$ will give a point in the curve where the most economical mean effective pressure for this particular condition lies, as the ratio of the cost of fuel to the mean effective pressure is greatest there. Having the mean effective pressure it is easy to figure back and obtain the corresponding cutoff. If a line $E\,F$ is drawn a distance above the lower line such that $O\,O'$ equals the mean effective pressure corresponding to the friction horsepower, then $O'\,C$ will give a point on the curve for the most economical brake horsepower.

So far only the cost of the fuel has been considered. Therefore, represent by $O'\,G$ the cost of all the other items, such as interest on the first cost of the plant, depreciation, taxes, maintenance, water, oil, waste, wages, etc. Then the tangent drawn to the curve from G will give a point D as the most economical cutoff under the actual conditions.

Operating Notes on Refrigeration

By F. E. Matthews

> *The discussion is limited to the defrosting of brine and direct-expansion coils, removal of oil and permanent gases from the system and the effect of incrustation on the condenser coils.*

DEFROSTING REFRIGERATING COILS

The problem of defrosting cold storage coils is one which is too often ignored at the time of making the installation. Where such has been the case the operator often finds himself working at great disadvantage, not only because of his inability to produce the required temperature, but also because of the decreased efficiency of the mechanical equipment entailed by the reduction in back pressures necessary to coax the heat from the cold-storage rooms through the additional resistance offered to its passage by the accumulated ice. This disadvantage affects principally the coal bill and fortunately for the operator, though unfortunately for the owner, is in the majority of cases not recognized, or, if apprehended, is not charged to the proper account.

The effect of a coating of ice on direct-expansion pipes may be shown as follows: Assuming a heat transfer of 10 B.t.u., in round numbers, per hour, per square foot per degree of difference in temperature inside and out, for a flat metallic refrigerating surface,[*] and an equal amount for a sheet of ice 1 inch thick, it follows that the heat transmission through a square foot of direct-expansion cooling surface insulated with a layer of ice 1 inch thick will be only one-half that of the uncoated surface. As a matter of fact, it would seem from the context that the value of 10 B.t.u. given as the heat conductivity of ice, applied to plate-ice conditions under which the wetted surface of the submerged ice will transmit materially more heat than a dry surface in contact with air. This would indicate that the decrease in heat-transmitting capacity of direct-expansion surfaces in air due to a coating of ice is even more than 50 per cent. This condition will be partially offset by the fact that on account of the increasing diameter, the layer of ice in the case of cylindrical surfaces such as pipes, which, together with the fact that such coatings usually present an irregular surface, further increasing the heat-absorbing area, may increase the heat transmission sufficiently to make up for the lesser heat transfer between the air and dry ice and make 50 per cent. at least a reasonable estimate of the loss in heat-absorbing capacity due to 1 inch of ice.[*]

Brine pipes may be readily defrosted by the circulation of hot brine. This may be accomplished through the main feed and return headers where the operation does not have to be performed very frequently, or as in abattoirs where the excessive amounts of moisture from the hot meats to be chilled make the accumulation of frost very rapid, or by a separate set of defrosting headers.

In the case of direct-expansion coils, the defrosting method probably most satisfactory where the cold-storage temperatures are above 32 degrees Fahrenheit is to install sufficient coil surface to allow a part of the coils to be shut off at any time, so that the frost will melt without artificial heat and at the same time produce a certain amount of useful refrigeration. If it is necessary to force the defrosting process by the use of outside heat, a hot gas line from the condenser may be connected to the liquid-line connections to the separate coils just inside the expansion valves. The hot gas, after melting the ice as it passes through the coils, returns to the compressor together with the return gas from the remaining coils.

Where the temperatures carried in the cold-storage compartments are below 32 degrees Fahrenheit, and in which the defrosting cannot be effected without the use of artificial heat, often very objectionable, two methods are available, viz.: that of forcibly removing the ice with scrapers, and that of suspending over the pipes trays of calcium chloride. This substance is exceedingly deliquescent salt, which in absorbing moisture from the air forms a saturated calcium brine which freezes at a very low temperature. In trickling down over the coils, the brine melts the ice, forming a more dilute brine which is then conducted away to the sewer, or, if the quantities involved warrant the expenditure of labor, may be evaporated and the calcium chloride recovered.

OIL IN THE REFRIGERATING SYSTEM

Next to unintelligent design which sometimes provides for the operating engineer a plant that cannot be made to develop nearly as high efficiency as operating conditions would warrant, and unintelligent operation which fails to get nearly as high efficiency as operating conditions and mechanical design would warrant, the worst foe to economy is foreign matter in the refrigerating system.

In order that the oils used in the system shall not stiffen prohibitively at the low temperatures encountered and not be saponified by the ammonia, only very light mineral oils can be employed. Such oils range from 22 to 30 degrees Baumé, corresponding to a specific gravity of from 0.924 to 0.88. These oils should have a cold test of about zero Fahrenheit, to obtain which they will have a flash point of between 310 and 400 degrees Fahrenheit This low flash point implies that a considerable amount of vapor will be given off at a much lower temperature. Since discharge temperatures of compression machines often approach these temperatures, it is obvious that a considerable amount of oil will pass to the condenser, not as a liquid but as a vapor. Under such conditions, since there is no material cooling effect in the oil separator, only liquid oil would be precipitated at that point.

Baffle plates on which the discharged gas impinges and to which particles of oil will tend to stick by adhesion as well as on account of the abrupt reversal of direction of flow of the gas which tends to separate out the slightly denser substances by centrifugal force, are precautions in the right direction; but except when discharge temperatures are low it is useless to expect to intercept all of the oil until it has been actually condensed. This operation being carried out most effectively in the condenser, the oil will pass to the receiver with the liquid ammonia, where, since there is a difference in specific gravity of 0.23 to 0.27, the separation can be more easily effected.

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[*] See Siebel's "Compend of Mechanical Refrigeration and Engineering," pages 190 and 207.

[*] Under average commercial conditions of intermittent frosting a square foot of direct-expansion surface in air is usually credited with a heat transmission of only from 2 to 4 B.t.u. per hour per degree difference in temperature.

The question of removing oil from the expansion coils, whether they are used for chilling cold-storage rooms, for chilling brine to be circulated through cold-storage rooms or for chilling the brine of an ice tank, is one which has received much attention, but, although much experimenting has been done and a few patents have been issued, no cheap and effective method has yet been devised where coils will drain so that there will be no danger of entrapping condensed moisture. The most effective method of removing the oil is to pump out the coils, disconnect them and blow them out with the highest-pressure live steam available, letting the steam blow through each coil as long as any oil appears at the exhaust end. The brine may be left in the tank and heated up by steam, so as to prevent, as far as possible, the condensation of steam inside the pipes, or if the brine has been removed the same result may be obtained by leaving the covers on the tank and blowing in live steam.

After the steam has been blown through the coils until no more oil appears, each coil should again be blown out with air to remove any traces of moisture from the steam. The moisture-absorbing capacity of the air can also be increased by heating it before it is introduced into the coils. The coils themselves may be heated as previously described. Long coils cannot be perfectly cleaned cold by air alone. Where coils will not drain in the tank as installed, the only effective way of extracting the oil is to remove the brine and disconnect them at such points as will enable them to drain, or remove them from the tank entirely. In case this method cannot be employed, ammonia purifiers may be installed with advantageous results. These purifiers work continuously when the plant is in operation, and their action can be facilitated by frequently pumping out the system. While it can hardly be expected that all of the oil can ever be removed by this means, their slow but continuous action is of great benefit in plants in which it is impossible to shut down in order to employ more effective methods, and will largely prevent the accumulation of oil in new or recently cleaned systems.

Permanent Gases

The action of so called permanent gases in the condenser is less detrimental than that of oil in the expansion coils which forms an insulating lining of fairly high efficiency on the inside of the heat-absorbing surfaces. The principal effect of these gases, if their operation can be limited to the condensers, is to occupy space that should be available for the ammonia gas. This reduces the effective area of the condenser cooler surface, causing an increased head pressure with the usual amount of cooling water and demanding an increased amount of cooling water to maintain the same head pressure.

These so called permanent gases are of rather uncertain origin and, as the name implies, are gases not liquefiable under the same conditions of temperature and pressure as ammonia. In a new plant or one recently overhauled they may consist almost wholly of atmospheric air. In an old one they are more likely to consist of nitrogen and hydrogen from decomposed ammonia, vapors of oil and mixtures of other hydrocarbon gases formed by the action of heat on hydrogen in the presence of vapors of oil, and water and other impurities contained in ammonia.

There is a considerable difference in opinion as to whether or not these gases are heavier or lighter than ammonia, and accordingly whether they can be most readily purged from the highest or the lowest part of the system. It seems reasonable to suppose that on account of the comparatively rapid flow of the ammonia vapors through the system, that sooner or later these gases, wherever formed, will find their way to the condenser. If the liquid outlets from the condenser are properly sealed, there will be no other way for them to escape than through the purge valve at the top of the condenser. If these gases are lighter than ammonia they will accumulate at the top of the condenser, and if heavier they will be driven to the top by the re-expanding ammonia in the bottom of the condenser as soon as the pressure is removed by the opening of the purge valve.

Condensers should be purged as often as the accumulation of gases indicate that they need it. To do this most advantageously, the liquid outlet and hot-gas inlet valves of the coils to be purged should be closed and a liberal supply of cooling water allowed to flow over them for some time. The permanent gases can then be purged out through a small rubber tube one end of which is connected to the purge valve; the other end should be immersed in a pail of water. If permanent gases escape when the purge valve is slowly opened, bubbles will rise to the surface of the water. If only ammonia escapes, the bubbles of ammonia gas will be dissolved in the water before reaching the surface, giving rise to a sharp crackling sound such as that caused by the condensation of steam in the process of heating water by the direct admission of steam through an open pipe.

When the water in the pail has become saturated and accordingly cannot absorb more ammonia, the bubbles will rise to the surface and it is difficult to discriminate between ammonia and the other gases. For this reason it is advisable to replace the aqua ammonia formed by fresh water, often enough so that it will not become saturated.

Incrustation on Condenser Coils

While the comparatively high working temperature of condenser coils, together with the usually ample provisions for draining each separate coil, prevents the accumulation of such large quantities of oil as are often lodged in expansion coils, condenser coils are exposed to another source of loss of efficiency from without, where the available cooling water is abnormally hard or carries a large amount of suspended matter. Ammonia condensers, and especially steam condensers, soon become coated with a deposit of scale or mud, which, if not properly removed, becomes a more or less effective insulator according to the composition of the deposit. The heat conductivity of metallic surfaces is not the same per degree difference in temperature at medium and low as it is for high temperatures, and it does not therefore follow that the resistance offered by the scale accumulating on the outside of atmospheric and submerged ammonia and steam condensers is the same as that of scale on the inside of a boiler. However, some slight idea of the extent of the loss may be gained from the fact that in steam-boiler practice, the insulating effect of scale results in a thermal loss corresponding to 2 per cent. of the fuel for each 1/64 inch in thickness of scale. Condenser surfaces like those of steam boilers, expansion coils or any other heat-transmitting surfaces should be kept as free as possible from deposits of foreign matter.

It seems superfluous to state that the heat-absorbing surfaces of brine-cooling coils and the heat-radiating surfaces of condenser coils should always be kept covered with brine and condenser water respectively. Nevertheless, it is not uncommon to see refrigerating plants operating with the brine so low in the tanks that the top expansion coils are exposed and the distribution of water over the condensers so irregular that a large percentage of the surface is dry.

Bill Grimes hez bin havin' a lot uv trubble keepin' th' packin' in th' cylinder hed uv his pump. It kept blowin' out so dumd fast thet he didn't hev any more left so he went over t' th' 'rest-yeraunt an borrered a pancake uv th' cook an' packed it with thet. He sez th' gol darnd pump ain't leaked a drap since.

Soot Blowers and Soot Suckers

By Warren O. Rogers

> *The steam soot cleaner has entered largely into the work of cleaning boiler tubes, because it is easy to operate and gives satisfactory results. Thirty-nine different designs are given attention.*

The simplest form of steam tube blower consists of an ordinary piece of ¾- or 1-inch steam pipe attached to a hose. While such a device will partly clean a tube, the current of steam will draw an excessive amount of air into the tube being blown, and when moving the pipe from one tube to another, soot is liable to be blown about the boiler room and surroundings, as pictured in Fig. 1.

An early type of tube cleaner is shown in Fig. 2; it was a combination tube blower and scraper. Steam was used to blow the soot and scale back through the tubes into the combustion chamber. The scraping portion was confined to the head and consisted of from five to seven annular rings having sharp edges cast upon its outer surface. These rings were made of such diameter as to easily pass into and through the tube, the sharp edges acting as scrapers upon the inside surface. The head was attached to a piece of pipe which was connected to the steam supply by means of a rubber hose.

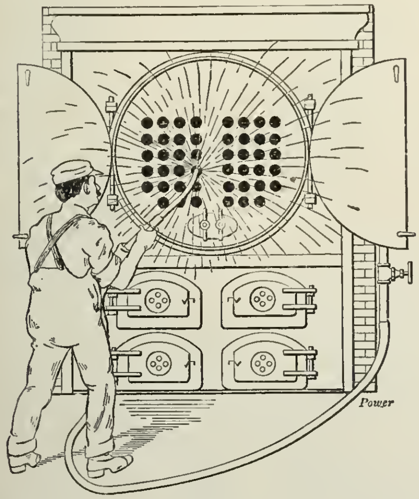


FIG. 1.

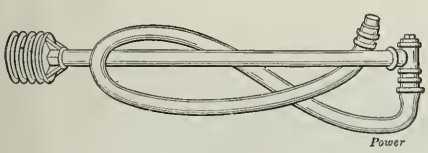


FIG. 2.

What is termed an automatic cleaner is shown in Fig. 3. Steam is admitted to the head through a valve which is controlled by a rod and thumb grip. The valve remains closed except when the head is placed at a tube end. When ready to blow, the operator's thumb is pressed on the grip and the steam blows the soot out of the tube. This arrangement saves steam and prevents soot from being blown out into the boiler room when passing the blower from one tube to another. It is made by the Jarecki Manufacturing Company.

Another design of blower, made by James A. Griffiths, is shown in Fig. 4. The head is beveled on the outside so as to fit snugly to the tube end. Steam enters the head through a tee, pipe and rubber hose. A spade handle makes it an easy matter to manipulate the cleaner. The steam pipe is fitted with a valve. By placing the head at the tube end and opening the valve, steam and air are blown into the tube. A sectional view of the head is also shown, the outer passage being for air.

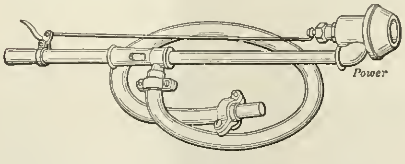


FIG. 3.

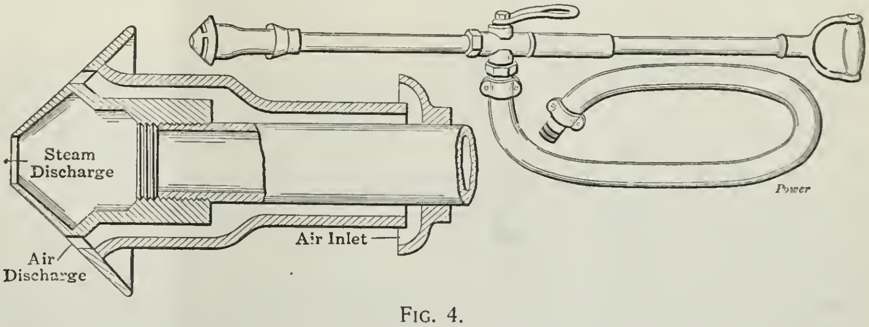


FIG. 4.

A device known as the McLaughlin cleaner and shown in Fig. 5 was made so that the outlet could be adjusted to regulate the supply of steam to any desired velocity by screwing the cap piece toward or away from the mouth of the orifice. As the opening was reduced the velocity of the steam increased, so that the blast would be effective at the far end of the tube. What was known as the "Perfection" steam cleaner is illustrated in Fig. 6. Its design permitted of shutting off the steam, while passing the head from tube to tube, by means of the steam cock placed between the handle and the hose. It was so arranged that it could be operated by the left hand while the cleaner was handled with the right.

The nozzle was made in the form of a truncated cone and fitted tightly to the tube end, thereby preventing the introduction of air and forcing dry steam the entire length of the tube.

Fig. 7 illustrates a steam tube cleaner manufactured by the Darling Pump and Manufacturing Company. The device consists of a hollow head with openings provided as shown at *A*. These outlets are so shaped as to give a swirling motion to the steam.

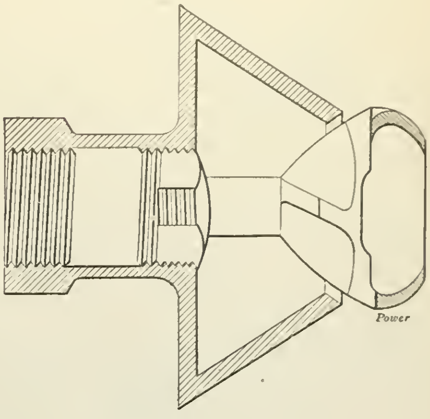


FIG. 5

What was called the "King Bee" cleaner is shown in Fig. 8. In operation, air and steam were admitted to the tubes being cleaned. The steam entered through the openings in the center of the head, the air entering around the spiral ribs on the outside of the head.

Fig. 9 shows another type of tube blower made by A. W. Chisholm & Co. The steam can be turned on full force without being wasted. The head, when pressed against the tube, is forced back, thus opening a valve which allows steam to pass through the hose only when the head is in position to blow a tube. The steam leaves the head through curved outlets which extend to its mouth, the aim being to blow the steam spirally through the entire length of the tube.

A "Cyclone" steam tube cleaner is shown in Fig. 10. The steam enters the cleaner head through the hose connection and is given a twisting motion as it passes through the head, the outlet being through the slot shown. It is made by the Crescent Manufacturing Company.

An injector blower, shown in Fig. 11, is made by John S. Bushnell & Co. A small amount of steam draws in a current of air which is heated and driven through the tubes. The steam is admitted to the center of the head through the small openings. The air mingling with the steam is forced through the tubes at high velocity.

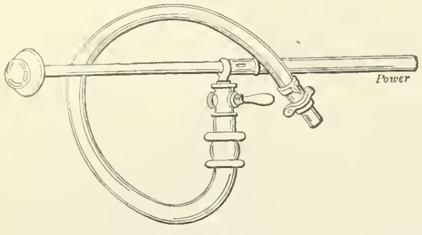


Fig. 6.

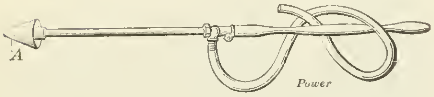


Fig. 7.

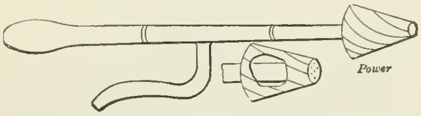


Fig. 8.

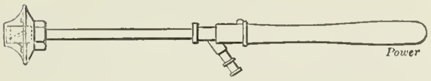


Fig. 9.

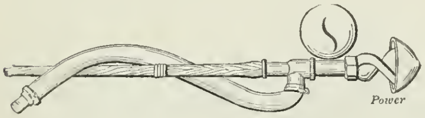


Fig. 10.

The cleaner shown in Fig. 12 was invented by William Coggeshall. The moisture in the steam is absorbed by the introduction of a current of air which is heated by the steam. This blast of hot air and steam expels the soot, ashes and rough scale. The steam is expelled through the center of the head and the air is admitted into the tube between the ribs on the head.

Another design of tube blower is illustrated in Fig. 13. It consists of a head having three steam outlets, each outlet being separated by a cone-shaped center piece. In operation the cleaner is used in the same way as most other devices of like type.

A whirlwind type of tube cleaner is made by the A. W. Chesterton Company. The internal construction of the head of this cleaner is shown in the sectional view of Fig. 14. The passage of the steam through the head keeps an inner revolving head in continuous rotation, which imparts a rotary motion to the steam and air in passing through the tubes.

The "Correct" steam flue cleaner, shown in Fig. 15, is automatic in its working, as it permits a passage of steam through the head only when in use. After opening the steam valve in the supply steam pipe, the full force of the steam pressure is on the head of the cleaner and within 2 inches of the end of the flue when in position to use.

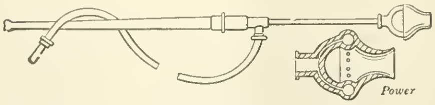


Fig. 11.

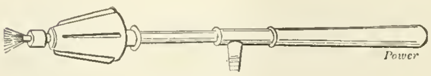


Fig. 12.

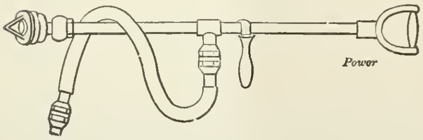


Fig. 13.

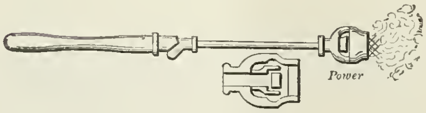


Fig. 14.

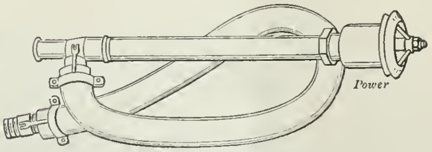


Fig. 15.

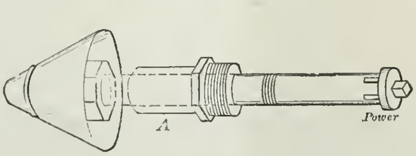


Fig. 16.

A slight pressure on the handle will force the deflector plate back and open a balanced valve the full size of the steam connection, thus instantly admitting the full force of the steam into the flue. With the steam confined close to the end of the flue, opening the valve suddenly produces an effect similar to that of a powerful blow. The steam cannot escape from the head except when the cleaner is pressed against the flue and there is, therefore, no escaping steam to dampen the soot and cause it to stick to the flue. The cleaner is made by the Michigan Lubricator Company.

Fig. 16 shows a steam tube cleaner made with an automatic sliding valve, so arranged that when the blower head is being moved from tube to tube the valve is closed and so prevents the escape of steam. The valve is placed in the casing *A* just back of the head. When the head is inserted in a tube a slight pressure on the handle causes the pipe to slide on the valve, exposing the ports, thus admitting steam to the tube through the slot in the head.

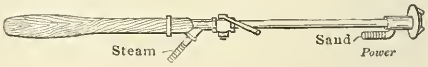


Fig. 17.

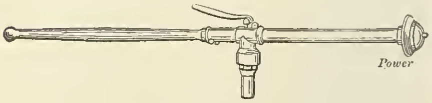


Fig. 18.

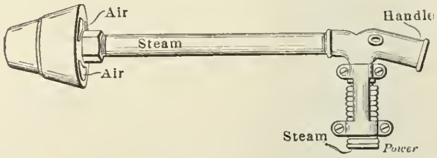


Fig. 19.

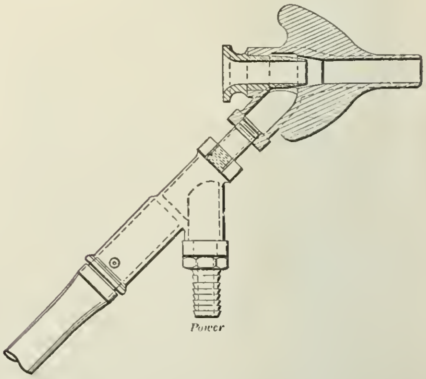


Fig. 20.

A type of sand-blast tube cleaner is shown in Fig. 17. A jet of steam is introduced into the head of the cleaner, creating a partial vacuum in the head which draws in fine sand. The combination blast of steam and sand is blown through the tubes with great velocity, thereby removing the soot and acting as a blower and scraper combined.

What is known as the "Red Jacket" steam flue blower is shown in Fig. 18. The steam is controlled by means of the valve and lever shown. The blower head is made with a rounding face which closely fits to the tube opening. When ready to clean a tube, the head is placed at the end and the valve opened by pressing down on the handle. The valve is closed while passing the head from one tube to another. This cleaner is made by H. W. Caldwell & Son.

The "Success" steam and air flue cleaner, made by Charles Nubring & Brother, is shown in Fig. 19. The soot is removed from the tubes by means of a small jet of steam and a large volume of air introduced simultaneously. The steam heats the blower head so that hot air is drawn into the tubes.

The "Parson" tube blower is shown in Fig. 20. It is manufactured by the Parsons Manufacturing Company. The head is made with two inlets and one outlet. One inlet is for steam, the other for air. The steam enters a chamber surrounding the air-inlet sleeve and, in rushing out around the air-inlet sleeve, forms a partial vacuum. Air is thus drawn into the head where it is mixed with the steam and then discharged through the tube. Owing to the shape of the head, several sizes of tubes can be blown with the same head.

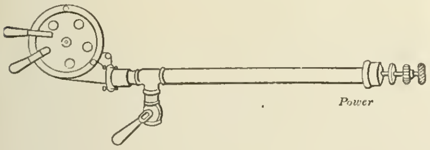


FIG. 21.

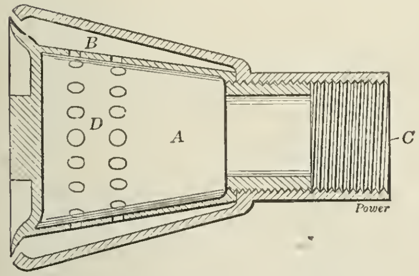


FIG. 22.

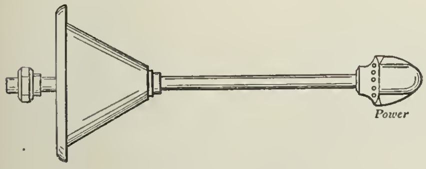


FIG. 23.

The old "Ball" cleaner differed considerably from the ordinary type of cleaner, and is shown in Fig. 21. It does not use steam as a cleaning agent, but utilizes it as a motive power to operate the two sets of rotating scrapers shown. These scrapers are attached to hubs which revolve in opposite directions, the centrifugal force throwing the scraper out against the surface of the tube. The end of the scraper is inserted in the tube, and the gasket near the end forms a steam-tight joint. Steam is admitted through the valve shown, and passing through and around a perforated follower disk shown back of the scrapers causes the latter to rotate rapidly in opposite directions and at the same time forces them through the tube as fast as the operator pays out a flexible steel ribbon from the reel. This ribbon passes through a stuffing box in the end of the cleaner and is attached to a short rod which carries the disk and scrapers, and serves to withdraw them when wound upon the reel.

Another design of steam cleaner is shown in Fig. 22. It consists of an inner cone-shaped section *A*, which is secured to an outer shell *B*, the outside diameter being such that a close fit to the tube is obtained. Steam enters the inner section *A*, through the inlet *C*, and escapes through the holes *D*, to the space between the inner and outer shells. In the front end of the head between the adjacent edges of these two sections is formed an annular series of steam outlets which incline outwardly and forwardly, so as to direct jets of steam against the boiler tube just in advance of the scraper head.

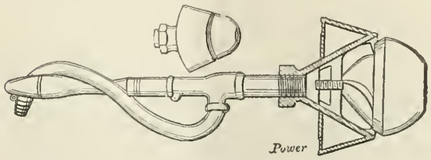


FIG. 24.

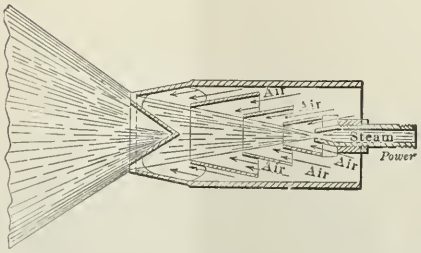


FIG. 25.

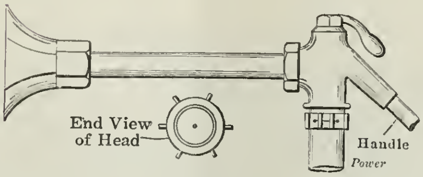


FIG. 26.

A design of soot cleaner, known as the "Vulcan," made by the Corbett Supply Company, is shown in Fig. 23. The cleaner is connected with a live-steam pipe, and is relieved of all condensation prior to entering the tubes of the boiler. In the head of the cleaner are small counterbored apertures, which aid in relieving condensation and also in causing an agitation of soot in the tubes as the cleaner is placed in them.

The tube cleaner is admitted into the tube for about 6 inches, until it strikes a shield which acts as a valve and drives the steam to the rear end of the tube, causing considerable agitation for its full length. By gently pulling out, the vacuum that is formed cleans the tube of soot and accumulation.

It is claimed that the steam is superheated and that it is utterly impossible to cause contraction, due to cold air entering the tubes or parts of the combustion chamber of the boiler. It is said that the cleaner will work in a very satisfactory manner with a steam pressure of from 10 to 15 pounds.

The A and D steam cleaner is shown in Fig. 24. It is made by the Cleveland Flue Cleaner Manufacturing Company. Owing to its design, a spiral motion is given to the jet of steam as it leaves the head. The outer surface of the head is turned true so that it fits nicely to the tube end. The sectional view of the head shows the general design.

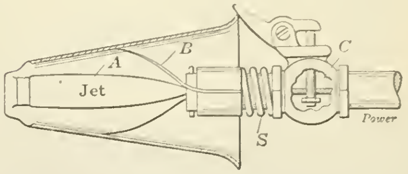


FIG. 27.

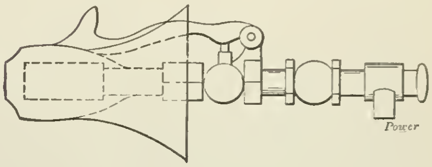


FIG. 28.

The "Tornado" cleaners, made by the Paul B. Huyette Company, is illustrated in Fig. 25. Owing to its design a large amount of air is admitted to the head when blowing a tube, thus reducing the amount of steam used. A steam jet is placed as shown, and with the truncated-cone arrangement shown in the illustration produces a vacuum when steam is turned on, thus creating a current of air containing but little moisture. The air and steam current is directed by the inverted cone at the front end of the cleaner against the inner wall of the tube for its entire length.

An improved steam blower, made by the Sherwood Manufacturing Company, is shown in Fig. 26. It is so designed that the flow of steam is not retarded in passing through the head. The device is fitted with a valve for turning on and off the steam. The same type of blower is also made for vertical boilers.

An early design of cleaner designated as the "Hurricane" is illustrated in Fig. 27. The main feature was that of admitting a jet of steam through the center of the head. A current of air was introduced through the bell-shaped nozzle *B*, and was thereby given a rotary motion by the spiral curved plate *A*. The rotating column of air and steam entered the tube, the idea being to produce the whirling action of a miniature hurricane to remove the soot. The large amount of air in proportion to the volume of steam prevented any objectionable amount of condensation to form a paste and bake upon the surface of the tubes.

The bell head was held in position by a catch attached to the same plate that imparted the spiral motion to the steam current. It could be easily detached, exposing the interior of the jet for cleaning. *A* and *B* were held in place by a collar which was retained in position by a short spiral spring *S*. When the bell was placed in a tube and forced against the tension of the spring its edge operated a lever and opened the valve *C*. Relieving the pressure with which the cleaner head was held against the tube automatically shut off the steam, when changing from one tube to another.

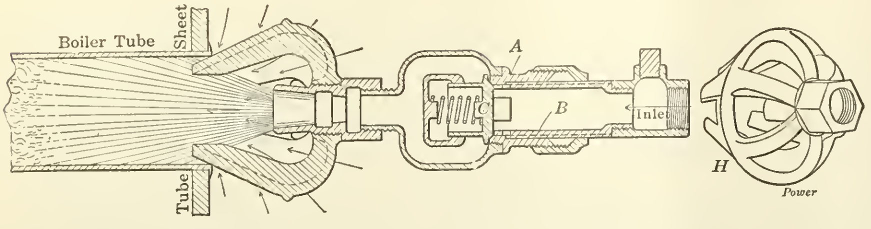


FIG. 29.

Fig. 28 shows another design of the same cleaner, the difference being in the operation of the valve. The lever operating the valve stem is extended on the outside of the head, and as the cleaner is forced against the tube, the lever is depressed, thus opening the valve. It is made by James L. Roberston & Sons.

The "Claflin" cleaner is shown in Fig. 29. The skeleton head is provided with six extending prongs which are ribbed on the inside. A divergent tapering steam nozzle passes through the center of the head. When the cleaner is in operation a partial vacuum is formed behind the nozzle by the rapid longitudinal expansion of the steam passing through the tube. The air is thus induced through the prongs in large volume, which relieves the vacuum by equalizing the atmospheric pressure at both ends of the tube.

The cleaner is used in conjunction with a telescopic automatic balanced valve, shown attached to the head. This valve will work on any steam pressure up to 120 pounds. When the cleaner head is forced against the tube the cylinder *A* slides on the plunger *B*, thus opening the balanced valve *C*, which is automatically closed by the steam when changing from one tube to another. This gave the advantage of having steam at full boiler pressure confined within a few inches of the end of the tube.

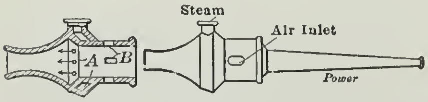


FIG. 30.

Fig. 30 illustrates an early type of the Grime injector cleaner. Steam is admitted into an annular chamber *A*, from which it escapes through small holes toward the mouth. The current of steam coming from these holes induces an air blast through the central pipe, shown in the sectional view of the head. The air enters through the holes *B*. The velocity of the escaping steam is communicated to the air, the idea being to reduce the amount of steam used in proportion to the work performed. The cleaner consists of one piece, has no movable parts, and the nozzle is so shaped that it can be used on tubes of from 2 to 4 inches in diameter.

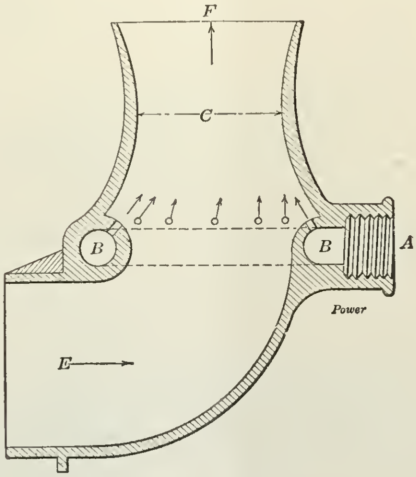


FIG. 31.

The cleaner shown in Fig. 31 differs from the ordinary tube cleaner, in that the soot instead of being blown out of the tubes into the back connection is drawn by a partial vacuum from the rear end to the front without admitting steam into the tubes.

The device operates on the injector principle; steam, being admitted at *A* and passing to the circular chamber *B*, is admitted to the small holes shown. As the tube is smaller at *C*, considerable velocity is given to the steam and a strong suction is produced at the end *E* which is held against the tube end. This draws the soot out of the tube and, passing through the cleaner, it is discharged through the end *F* to the smoke flue.

An improvement to the head illustrated in Fig. 31 is shown in Fig. 32. It consists of a plug in the steam ring so that in case pieces of rubber hose are blown into the chamber the plug can be unscrewed and the obstruction removed. This improved soot cleaner is made by the John S. Bushnell Company.

Fig. 33 illustrates a "Thompson" soot ejector made by Richard Thompson & Co. Steam is discharged from the central tip, creating a vacuum which draws the soot out of the tube and ejects it up the stack. Its inner construction is shown in the sectional view of the illustration.

A handy device for blowing the tubes of an upright boiler is shown in Fig. 34. It consists of a head having a hollow center through which the steam emerges. On the outside of the head three ribs are cast lengthwise for the purpose of centering the head. At a point on the horizontal rod at a sufficient distance from the elbow to allow the head to reach the innermost tube, is a hinged joint. This joint allows of swinging the blower head up and down when changing from one tube to another. The hinged section is fitted to a hollow stand and is held in place after being adjusted by a set screw. The adjustable feature permits of the device being used on boilers of different sizes. This blower was designed by F. R. Low.

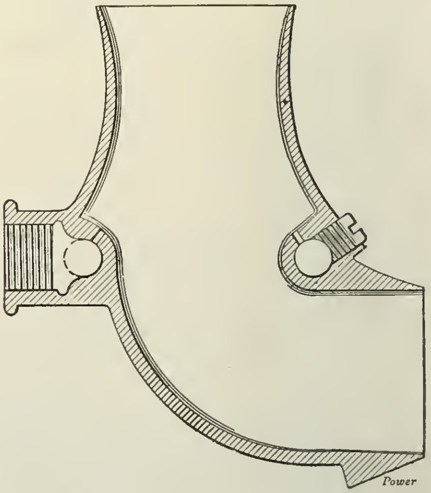


FIG. 32.

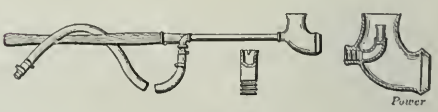


FIG. 33.

A variation of the type of cleaner illustrated in Fig. 11 is shown in Fig. 35. The only difference is that it is vertically mounted on a pipe, as shown, and is for use with upright boilers.

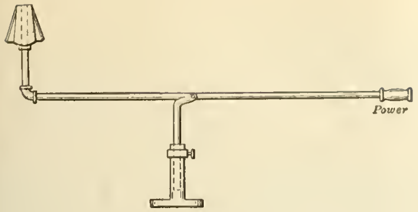


FIG. 34.

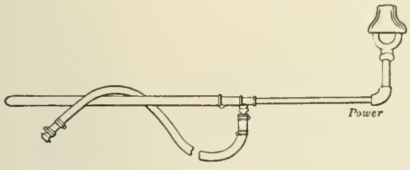


FIG. 35.

Fig. 36 shows the cleaner illustrated in Fig. 24, arranged for use with vertical boilers.

The tube blower shown in Fig. 37 is of the same design as that shown in Fig. 4, but is mounted on a pipe for blowing the tubes of vertical boilers.

Fig. 38 shows the same design of blower as appears in Fig. 10, arranged for blowing tubes of a vertical boiler.

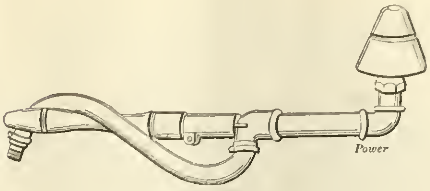


FIG. 36.

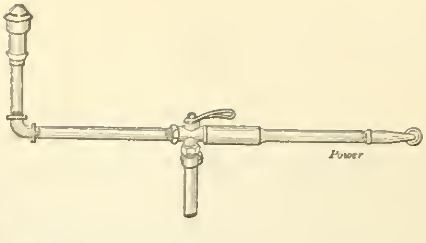


FIG. 37.

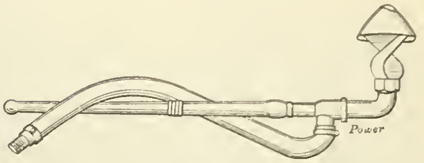


FIG. 38.

Engineer's Problem in Selecting Coal

By George H. Bayne

In a previous contribution under the above caption the author discusses the purchasing of coal upon the basis of actual boiler performance. In the present article a standardization test with three kinds of coal is given and sample computations are worked out, upon which the final selection is based.

A large manufacturing plant whose annual consumption runs into thousands of tons of bituminous coal, decides that the purchasing of its fuel shall be placed upon a more definite basis and that the coal shall be bought upon its value to them, irrespective of analyses and the statements of the several coal companies involved. To this end they institute a policy whereby their coal shall henceforth be bought through coöperation between the purchasing agent and the engineer in charge of the plant, the purchasing agent to determine from a business standpoint the relative values of the coals submitted to him and the engineer to test these to determine their actual values under his particular conditions, after which the two together can arrive at a conclusion as to which of the several kinds of coal submitted is the most economical.

The purchasing agent lets it be known to the different coal companies who are selling coal in his district that his concern is in the market for 25,000 tons of "run of mine" bituminous coal, clearly stating his requirements as to delivery, payments, etc., specifying that the coal which he wishes the companies to bid upon shall be free from slate and bone and shall not be slack; and that the coal shall be guaranteed to maintain the following analysis: volatile, 18 to 25 per cent.; ash, 6 to 10 per cent.; sulphur, 1 to 2 per cent.; B.t.u., dry, not less than 14,250. To this inquiry some fifteen coal companies put in their bids, together with a guaranteed analysis. After investigating these the purchasing agent cuts down the number of probable successful applicants to three, which, for convenience, may be designated as Nos. 1, 2 and 3. No. 1 bases his offer on coal mined from the *C*-prime vein of the patent field in the Clearfield district of Pennsylvania. No. 2 offers a coal of the *B* vein from the South Fork district of Cambria county. No. 3 proposes to furnish a coal from the *D* vein from Indiana county, Pennsylvania. The freight rate into the plant from all three of these districts is $1.85, so Nos. 1, 2 and 3 offer these coals, delivered to the plant, at $2.95, $3 and $2.85 per ton, respectively.

Having reached this point, the purchasing agent orders a sample car from each of these three companies for test purposes and turns them over to his engineer to decide which is the best for the plant under its existing circumstances. This is as far as the purchasing agent can go for the present; the proposition is now up to the engineer to determine the commercial efficiency of the three grades under consideration for this plant.

The boiler plant under his charge consists of two 520- and eight 264-horsepower Aultman-Taylor water-tube boilers equipped with shaking grates, hand fired and using natural draft. In preparation for the test the engineer designates one boiler in the line for test purposes, and for a matter of convenience takes one of the 264-horsepower units. This boiler is carefully cleaned, the tubes are blown and the furnace thoroughly repaired; the brickwork is pointed up and put in good condition; and the feed-water line is bypassed and connected to the auxiliary feed-water pump. As it is impracticable in this plant to weigh in barrels the water entering the boiler, a Worthington duplex hot-water piston meter is installed in the line between the pump and the boiler; also the feed-water line is tapped for a thermometer, which is placed as near as possible to the entrance into the boiler. For taking the flue-gas temperatures a pyrometer is placed in the breeching between the damper and the uptake. The stack is tapped and a draft connection made for taking draft readings over the fire in the first pass. Scales of the Fairbanks type are set up in the boiler room, and upon these all coal under test is weighed before being delivered to the fireman; also, the blowoff cocks are blanked off in order to eliminate any possibility of leakage from this source. One of the most experienced firemen in the plant is detailed to fire all three of the coals under test.

As this plant operates on an eight-hour day basis, it is decided that each coal shall be given a trial covering this period, which is considered sufficient. Preliminary to starting the test, the three grades of coal are dumped from the cars in three separate piles, which are marked No. 1, No. 2 and No. 3, respectively, and are delivered to the firemen accordingly. The engineer then provides himself with a notebook, in which the log of each test is to be recorded; this can be conveniently arranged in the following manner:

GENERAL DATA.

Number of boiler used ....
Name of fireman ....
Kind of coal (anthracite or bituminous).........
Supplied by.......................................
Car number...

Number of tests
Duration of test hours
Date of test
State of weather (clear, cloudy, rain or snow).....

COAL.

Size and condition (slack, run-of-mine or screened, well prepared or dirty)
Total weight of coal fired during test
Percentage of moisture in coal by weight
Total ash and refuse
From fire box
From ashpit
Nature of clinker (hard or soft, large or small)
Thickness of fire required and bar work necessary
Method of firing (spreading, alternate or coking)..

WATER.

The total weight of water fed the boiler during the test
(If a meter is used, this will be the difference between the initial and the final readings of the meter, corrected for the temperature of the feed water, and should be calculated in pounds.)

PRESSURES.

Barometer (if a barometer is not available, 14.7 pounds per square inch is correct for all practi-puposes)
Steam pressure by gage per square inch
Draft at stack
Draft in furnace, over fire
Draft in ashpit (this is only necessary when forced draft is used)

TEMPERATURES.

Temperature of feed water entering the boiler.....
Temperature of escaping gases (pyrometer).....

The percentage of moisture in the coal is found by placing 10 to 20 pounds of the coal in a galvanized-iron bucket, or preferably a dish pan, which is placed on top of the boiler at the hottest point and is allowed thoroughly to dry out. This will take between ten and twenty-four hours, according to the moisture that is in the coal. The difference between the initial and the final weights of the coal reckoned as a percentage of the moist coal is taken as the moisture in the coal, as delivered to the boiler room.

This done, the next point to take into consideration is the calibration of the meter, which is very important. A barrel is placed on the scales, which are set to compensate for the weight of the barrel. The meter is allowed to discharge into the barrel until the latter is full and the temperature of the water is taken during the process of filling. The barrel of water is then weighed and the difference between the initial and the final readings of the meter is noted and the amount of water passed through the meter, corrected for its temperature, is calculated. This result should coincide with the weight of the water in the barrels. The process is repeated at least five times, from which the average result is determined. The difference between the readings of the average weight in the barrels and that shown by the meter, when figured in per cent., gives the correction that has to be allowed, either plus or minus, to the total amount of water as shown by the meter to have been fed to the boilers during the test. For example, suppose that ten barrels of water have been drawn through the meter and found to weigh on the scales 3420 pounds. The first reading of the meter indicated 540 cubic feet and the final reading 600 cubic feet; in other words, 60 cubic feet have passed through the meter, according to the readings. Now the average temperature of the feed water was 203 degrees Fahrenheit. The weight of a cubic foot of water at 203 degrees is 60 pounds per cubic foot; therefore, the total number of pounds of water passed through the meter was

$$60 \times 60 = 3600 \textit{ pounds.}$$

The actual weight in the barrels showed 3420 pounds: therefore the meter was evidently reading fast by 180 pounds, and the percentage of overreading is

$$180 \div 3420 = 0.052 = 5.2 \textit{ per cent.}$$

Hence the total amount of water, as reggistered by the meter, should be multiplied by 0.948 in order to obtain the correct reading. If, on the other hand, the meter had been found to read low by 5.2 per cent., it would be necessary, in order to obtain the correct reading, to divide by 0.948.

Smoke

The engineer from time to time should observe the smoke issuing from the stack and note the quality on his log. This is essential when the plant is situated in a residential district or where a smoke ordinance is enforced.

All the readings of temperatures and pressures are taken every half hour during the test. With these points in hand, the engineer starts to test the coals, testing each grade separately and taking every precaution to have the load factor as nearly constant for each grade as possible. Using the alternate method of starting and stopping the test, as explained in a previous article under the same caption by the writer, the following comparative results are obtained:

Economic Comparison between Coals Nos. 1, 2 and 3.

Kind of Boiler—Aultman-Taylor.

Kind of Fuel—No. 1, No. 2, No. 3.

Kind of Furnace—Hand fired; shaking grates.

State of Weather—For No. 1, rainy; No. 2, clear; No. 3, rainy.

Method of Starting and Stopping Test—Alternate.

Number of Boilers Tested—One.

Type of Boiler—Water-tube.

Date of Trials—

Duration of Trial—Eight hours for each grade of fuel.

DIMENSIONS AND PROPORTIONS.

	Total.
Grate surface, square feet	63.75
Width of grate	8 feet, 6 inches.
Length of grate	7 feet, 6 inches.
Kind of draft	Natural
Water-heating surface, square feet	2640
Superheating surface	None.
Ratio of water-heating surface to grate surface	41.4 to 1

AVERAGE PRESSURES.

Barometer, inches of mercury. ....Not taken.

	No. 1	No. 2	No 3.
Steam pressure by gage, pounds per square inch	143	149	133

DRAFT.

Draft between damper and boiler, inches of water..	0.35	0.38	0.34
Draft in furnace, inches of water	0.23	0.24	0.27
Draft in stack, inches of water	0.8	0.8	0.8

AVERAGE TEMPERATURES.

Gases escaping from boiler, degrees Fahrenheit	525	540	580
Feed water entering boiler, degrees Fahrenheit	96.6	117.1	88.00

FUEL.

Size and condition	Run-of-mine, well prepared.		
Total weight of coal fired, pounds	9323	9170	7180
Moisture in coal (actual), per cent	5	3.1	2.34
Total weight of dry coal consumed, pounds	8857	8886	7012
Total weight of ash and refuse, pounds	854	942	812
Ash and refuse in dry coal, per cent	9.64	10.61	11.58
Total combustible consumed, pounds	8003	7944	6200

FUEL PER HOUR.

Dry coal consumed per hour, pounds	1107.1	1110.7	876.5
Combustible consumed per hour, pounds	1000.4	992.9	775

PROXIMATE ANALYSES OF COALS.

Moisture, per cent	2.6	1.7	1.2
Volatile matter, per cent...	24.2	17.3	23.0
Fixed carbon, per cent...	70.1	73.1	68.5
Ash, per cent	5.7	9.6	8.5
	100.00	100.00	100.00
Sulphur, separately determined, per cent	1.29	1.64	1.55
Color of Ash	Yellowish gray	Yellowish gray	Bluish gray

CALORIFIC VALUE OF FUEL.

Calorific value of coal per pound as received	14,433	13,944	14,380
Calorific value of dry coal per pound, B.t.u	14,795	14,236	14,782

WATER.

Total weight of water fed to boiler, corrected for meter calibration, pounds	79,223	76,997	56,199
Equivalent water evaporated into steam from and at 212 degrees, pounds..	92,231	88,054	65,854
Factor of evaporation.	1.1642	1.1436	1.1718
Equivalent water evaporated into steam from and at 212 degrees per hour, lb ..	11,529	11,006.8	8,231.7

ECONOMIC EVAPORATION.

Equivalent water evaporated per pound of dry coal from and at 212 degrees, pounds	10.41	9.91	9.39
Equivalent water evaporated per pound of combustible from and at 212 degrees, pounds	11.52	11.08	10.62

RATE OF EVAPORATION.

Water evaporated from and at 212 degrees per square foot of heating surface per hour, pounds	4.37	4.17	3.12

RATE OF COMBUSTION.

Dry coal consumed per square foot of grate surface per hour, pounds..	17.36	17.42	13.74

HORSEPOWER.

Builders' rating at 10 square feet of water-heating surface per horsepower, pounds	264	264	264
Horsepower developed (34½ pounds of water evaporated per hour from and at 212 degrees = one horsepower	334.1	319.04	238.6
Percentage of overrating developed	26.55	20.84	9.62 (underrating)

EFFICIENCY.

Efficiency of the boiler, including the grate, per cent	67.62	67.55	61.64

METHOD OF FIRING.

Kind of firing	Coking	Coking	Coking
Average thickness of fire	8′ 10″	6′ 10″	7′ 8″

ECONOMIC RESULTS.

Cost of coal per gross ton delivered	$2.95	$3.00	$2.85
Estimated gross tons of coal required	23,938.9	25,146.7	26,539.3
Estimated cost of coal per annum, based on price per ton	$70,618.75	$75,440.10	$75,637.00
Estimated cost of ash removal	$0.20 per gross ton		
Estimated total cost of ash removal, based on percentage of ash in coal	$461.54	$533.61	$614.65
Total estimated cost of coal per annum, including ash-removal charges	$71,080.29	$75,973.71	$76,251.65
Net difference per annum, in favor of No. 1 over No. 2 and No. 3	....	$4,893.42	$5,171.36

Before going into the discussion as to the relative merits of the three grades of coal under consideration, it would be well to consider how the engineer has arrived at the results, as tabulated. During the test he has observed the average temperatures, pressures, the amount of fuel delivered, the amount of ash obtained and the amount of water evaporated by the boiler, and has recorded these upon his log sheet. The method of determining the moisture in the coal has been explained, and this moisture was found to be 5 per cent. by actual weight in sample No. 1. Therefore, the total weight of dry coal consumed is the total weight of coal fired, less the moisture; that is,

$$9323 \times 0.95 = 8857 \textit{ pounds.}$$

The total ash and refuse, as shown by the log, was 854 pounds; hence the percentage of ash and refuse in the dry coal is

$$854 \div 8857 \times 100 = 9.64 \textit{ per cent.}$$

The total combustible consumed is the total dry coal less the ash and refuse:

$$8857 - 854 = 8003 \textit{ pounds.}$$

The weight of dry coal consumed per hour is

$$8857 \div 8 = 1107.1 \textit{ pounds.}$$

The next thing necessary is to find the factor of evaporation, which is in proportion to the steam pressure and the temperature of the feed water and is calculated in the following manner:

$$\textit{Factor of evaporation} = \frac{H_1 - H_2}{970.4}$$

where

H_1 = Total heat of the steam at the gage pressure (reckoned above 32 degrees Fahrenheit);

H_2 = Total heat in the feed water at the observed temperature, reckoned above 32 degrees Fahrenheit;

970.4 = Latent heat of vaporization at 212 degrees.

From the latest steam tables H_1 is found to equal 1194.3, for case No. 1, and H_2 is 64.6; therefore,

$$\frac{1194.3 - 64.6}{970.4} = 1.1642 = \textit{Factor of evaporation}$$

The equivalent water evaporated into steam from and at 212 degrees equals the total weight of water fed to the boiler multiplied by the factor of evaporation. This with coal No. 1 would be

$$79{,}223 \times 1.1642 = 92{,}231 \textit{ pounds.}$$

The equivalent evaporation into steam from and at 212 degrees per hour is equal to

$$92{,}231 \div 8 = 11{,}529 \textit{ pounds.}$$

The equivalent water evaporated per pound of dry coal from and at 212 degrees is equal to the total amount of water evaporated from and at 212 degrees divided by the total amount of dry coal consumed:

$$92{,}231 \div 8857 = 10.41 \textit{ pounds.}$$

The equivalent water evaporated per pound of combustible from and at 212 degrees is found by dividing the equivalent water evaporated into steam from and at 212 degrees by the total combustible consumed:

$$92.231 \div 8003 = 11.52 \textit{ pounds.}$$

The rate of evaporation, or the amount of water evaporated from and at 212 degrees per square foot of heating surface per hour, is found by dividing the equivalent water evaporated into steam from and at 212 degrees per hour by the number of square feet of heating surface in the boiler:

$$11{,}529 \div 2640 = 4.37 \textit{ pounds.}$$

The rate of combustion, or the dry coal consumed per square foot of grate surface per hour, is the total amount of dry coal consumed per hour, divided by the number of square feet of grate surface in the furnace:

$$8857 \div 63.75 = 17.36 \textit{ pounds.}$$

Thirty-four and a half pounds of water evaporated per hour from and at 212 degrees Fahrenheit has been set as the standard for one boiler horsepower. Therefore, the boiler horsepower developed per hour is equal to the equivalent water evaporated from and at 212 degrees per hour, divided by 34½ which is

$$11{,}529 \div 34\tfrac{1}{2} = 334.1 \textit{ horsepower.}$$

The percentage of overrating developed is the horsepower developed, multiplied by 100, divided by the builders' rating, minus 100:

$$\frac{334.1 \times 100}{264} - 100 = 26.55 \textit{ per cent.}$$

The efficiency of the boiler, figured in per cent., including the grate, is the evaporation per pound of dry coal from and at 212 degrees, multiplied by 970.4, and divided by the calorific value per pound of dry fuel:

$$\frac{10.41 \times 970.4}{14{,}795} = 67.62 \textit{ per cent.}$$

Having figured out the results of the test the engineer is ready to determine which of the three grades of coal is the most economical for him to use. He has a 3152-horsepower plant and estimates that in order to run this plant for one year of the required number of hours, including the holidays, shutdowns, etc., that it will require the evaporation of 558,218,209 pounds of water *per annum*. Therefore, he uses this as an arbitrary basis of comparison for the three coals. Coal No. 1 evaporates 10.41 pounds of water per pound of coal, from which he finds that 23,938.9 gross tons of this grade would be required:

$$558{,}218{,}209 \div (10.41 \times 2240) = 23{,}938.9 \textit{ gross tons.}$$

In a like manner he calculates that with coal No. 2 25,146.7 gross tons would be required, and with No. 3 26,539.3 gross tons. Figuring the costs of these three grades at their respective prices, the estimated gross cost of operating this plant, irrespective of firing charges and fixed expenses due to repairs, etc., would be $70,618.75 for No. 1; $75,440.10 for No. 2, and $75,637 for No. 3.

By previous experience, the engineer finds that the cost for ash removal is $0.20 per gross ton. Therefore, inasmuch as Nos. 1, 2 and 3 have, respectively, 9.64, 10.61 and 11.58 per cent. of ash and refuse, the approximate cost of ash removal that should be expected for the three grades of coal would be $461.54 with No. 1; $533.61 with No. 2, and $614.65 with No. 3. This makes a total net cost of the three grades of coal respectively, $71,080.29, $75,973.71 and $76,251.65. In other words, the engineer finds that by the use of coal No. 1 his estimated saving will be $4893.42 over No. 2 and $5171.36 over No. 3. When referring to his log he also finds that, as compared with No. 1, No. 2 and No. 3 have required more bar work and more attention from the fireman, which leads him to believe that the two latter coals would cause more damage to the brickwork and grate bars than the former. Also, upon an examination of the guaranteed analyses, as submitted by the competing coal companies, he finds that the sulphur and ash are higher in No. 2 and No. 3 than they are in No. 1. Again, by referring to the prices he notes that it is not the highest, nor the lowest, but the medium-priced coal that is best suited to his purposes; therefore he recommends to the purchasing agent that the contract for the fuel supply be placed for No. 1 coal.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Care and Maintenance of Induction Motors

By R. H. Fenkhausen

Notwithstanding the fact that the best types of modern induction motors are advertised as being almost indestructible, and are in fact marvelously rugged and trouble-resisting, it will be found that continued neglect, careless oiling and overloading will eventually cause such deterioration that extensive and costly repairs will be imperative and, unless spare motors are available, will cause much loss of time and money due to idle machinery and reduced output. As a general rule it will be found that ten cents expended in intelligent and systematic inspection of electrical equipment will mean a saving of from five to ten dollars in repairs later on.

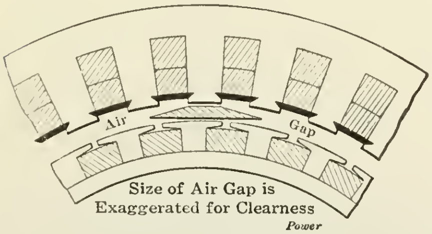


Fig. 1. Airgap Gage

This statement applies particularly to large motor installations in which one man can be detailed for motor inspection and his salary divided among so many motors that the cost per motor is small. The repair bill alone on one motor burn-out would often pay the man's wages for a month, to say nothing of the loss due to the shutdown.

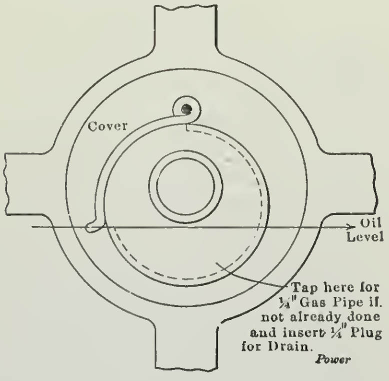


Fig. 2. Usual Form of Oil Well

In a plant that once came under my supervision no system of motor maintenance was applied. The motors were oiled by the various department oilers and ordinary red machine oil was used. No attention was given to motors until they gave signs of distress, whereupon they were shut down and taken to the shop, where several repair men were constantly at work. Shutdowns were numerous and maintenance costs high due to costly repairs.

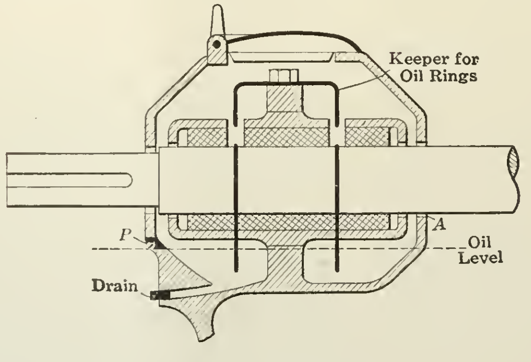


Fig. 3. Old-style Box

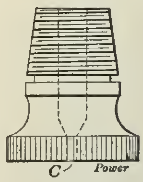


Fig. 4.

A careful investigation disclosed the fact that two causes were responsible for about nine-tenths of the motor troubles, namely, careless oiling and worn bearings. The first cause was also largely responsible for the second, owing to the poor quality of oil used. The first change made was to take the motor lubrication out of the hands of the various department oilers and make one man responsible for all motors. This did not necessitate an extra man because when released from motor oiling, each oiler was enabled to cover a larger territory and one was laid off. The man selected for motor maintenance was an electrical helper of the better class, and received $2.75 per day. He was furnished with a good quality of light engine oil, often called dynamo oil, and instructed to use nothing else.

All the motors in the plant were listed and divided into five groups, one of which was to receive attention each day. The sixth day was kept as a "catch-up" day, in case a serious breakdown necessitated the man's assistance for a day in the repair shop. The motors were in such dirty condition that a man and helper from the shop were detailed to overhaul all of them, one at a time. The work was done without removing the motors and with a very slight interruption of service, as only a day was required, on the average, for each motor. The motors were completely dismantled and washed clean with gasolene, special attention being given to the bearings, oil wells and oil-soaked coils. After being thoroughly cleansed of all oil, the windings were given two coats of a waterproof insulating varnish; this varnish was liberally applied, and was insoluble in oil. The bearings were tried and if worn were replaced with spares from stock. The motor was then reassembled, the oil wells filled with new oil and finally two coats of green enamel were put on the entire frame of the motor. This was done because it improved the appearance of the machine, and experience has shown that a nice-appearing machine will receive attention whereas an equally clean but less attractive one will be neglected. The motor inspector on his regular rounds kept the oil wells filled, tested the airgaps, wiped off the motor all over and inspected all starter contacts and controller fingers.

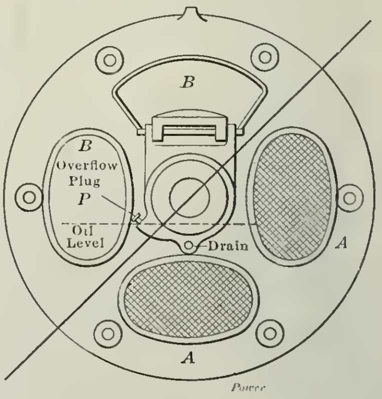


Fig. 5. "Open" and "Ventilated" Forms

From the very start it was possible to reduce the shop force and at the end of four months when all the motors were overhauled there was not one man engaged on motor repairs in the entire plant. One man now keeps up the entire motor equipment, and only six motors have been to the repair shop in four years; all of these were motors that had been oil-soaked before the new system was adopted. No appreciable wear has occurred in any of the motor bearings, except those in such locations as required short vertical belt drives. These are being replaced with chain drives as rapidly as conditions permit; short vertical belt drives are poor engineering and are indefensible in most cases.

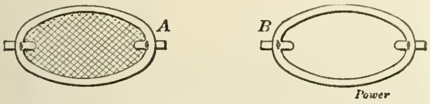


Fig. 6. Covers for Frame Openings

In testing airgaps, two gages are used irrespective of motor sizes. If the large one will not enter at four equidistant points, the motor is listed for new bearings at the first opportunity, but if the small gage will not enter, the machine is immediately shut down. The most serious trouble about bearing neglect is that if the rotor is allowed to come in contact with the stator the heat caused by friction usually burns up all the insulation on the rotor bars and on as many stator coils as are included in the arc of contact between the rotor and stator. The stator coils, of course, require renewal, but if the rotor insulation is evenly burned the rotor may be allowed to run, as one of the principal uses of the slot insulation on a squirrel-cage rotor is to hold the bars tightly and prevent vibration, and the absence of insulation would merely lessen the resistance of the rotor and therefore decrease its starting torque slightly. As a rule, however, the rotor insulation burns unevenly and the bars should therefore be reinsulated to prevent electrical and mechanical unbalancing.

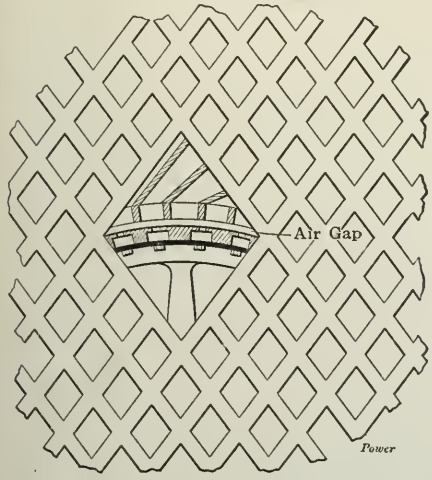


Fig. 7. Access to Airgap through Grating

If an airgap gage is not furnished by the motor manufacturer one should be made of spring steel and ground to the shape shown in Fig. 1.

When a motor shaft becomes reduced in diameter due to excessive wear, it should not be turned down unless it is in the only motor of the size in the plant. It is better to make a new shaft and keep up the standard size so that one pair of spare bearings will answer for all motors of the same original shaft size.

Careless filling of oil wells leads to oil being spilled on the windings and, in spite of varnish or other oil-resisting compounds, the oil will eventually work its way into the coils where it leads to insulation breakdowns and short-circuits. Most modern motors have oil wells similar to Fig. 2, which prevent overfilling, but the older types were equipped with oil wells like Fig. 3 and if oil was poured into the wells without removing the filling plug *P* the oil would overflow at *A* and drip down on the windings. To prevent this the plugs may be left out all the time but owing to the inevitable entrance of dust, it is better to drill and counterbore the plug as shown in Fig. 4; then excess oil will spurt out with considerable velocity at *C*.

The old "Type C" Westinghouse motors, of which many thousands are still in use, were equipped with heads having ventilating grids cast integral with the heads, as shown at *A* in Fig. 5. This construction renders inspection and cleaning very difficult and prevents the insertion of an airgap gage. These grids should be broken out and the openings filed smooth, as shown at *B*, Fig. 5. This was done to over a hundred motors in the plant under my charge, and besides facilitating cleaning and inspection it has increased the load the motors can carry without overheating, owing to the increase in ventilation secured.

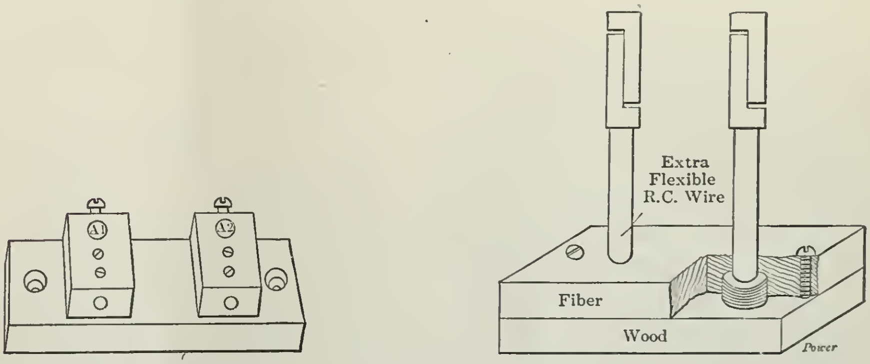


Fig. 8. Motor Terminals Fig. 9.

It will be found that the air drawn through the motor by the fans on the rotor carries considerable dust along with it, and this dust is intercepted by the arms of the rotor spider and thrown against the inside of the rotor laminations by centrifugal force. It cannot be successfully removed with the grids in place, even by means of compressed air, and several motors were found with the rotors packed solid with dust, which was baked hard. Of course, these motors ran cooler after being cleansed than they did before.

Motors installed on the floor in our plant, or where there is liability of chips being drawn into them, are fitted with removable ventilating covers held by thumb latches, as illustrated at *A*, Fig. 6, and motors exposed to the weather are provided with solid covers, as at *B*, Fig. 6, backed with rubber gaskets. Motors of over 50 horsepower would have been seriously weakened by removing the grids, so diamond-shaped handholes were cut in each grid opposite the airgap, as shown in Fig. 7. As most modern induction motors are constructed to permit access to the interior for cleaning without removing the heads, the foregoing remarks are not applicable to them.

Many induction motors are fitted with terminal blocks like Fig. 8, in which the wire is held in a drilled hole by the pressure of a set screw. The danger of this type of contact has been pointed out in a previous article; for motor service it is particularly bad, owing to the liability to short-circuit by accumulated dust or by accidental contact with chain tackles, etc. Fig. 9 shows a method by which motor terminals may be made to conform to the latest practice. Knife-blade connectors are used on the leads and the wrapping of twine between the two blocks prevents any strain from coming on the motor winding.

Old Pete Blowoff got drunk agin a spell ago an' got ter monkeyin' with th' valve gear uv his ingin. After he'd got dun he put th' indycater on it an' took off a keerd, sent it in t' th' Power fellers an' ast them what they thot uv it.

"Bill" Bangor writ back an' told him thet it looked jist like th' Stand Patter's perlitical map uv New Yark after Teddy Roosfelt hed got dun wipin' his feet on it up at Saratogy.

LETTERS

How Can One Identify Alternating or Direct Current

Can any of my fellow readers give me a simple and reliable means of finding out whether the current is direct or alternating from an ordinary lamp socket?

H. Priestley.

Patea, N. Z.

Drying Transformer Oil

I was interested in Mr. Hunter's articles on transformer oil, in the issues of October 18 and 25. While the chemical process may be applicable to drying out small transformers, I doubt very much that it would prove either expedient or practical with transformers of considerable size. Having served for a time as an engineer of electrical construction, I have had occasion to dry out oil for numerous transformers of from 300 to 2000 kilovolt-amperes capacity. In doing this I have found nothing to compare with steam heat for quickness, thoroughness and cheapness. Steam is used in nearly all stations for either heat or power, but where it is not ordinarily available any portable boiler may be used to supply it during the drying process.

With transformers designed for water cooling, the process is simple. The steam-supply pipe is connected to the water inlet of the cooling system and, by means of the valve in the outlet, about ten pounds steam pressure is kept on the coil. This will dry out the oil in a 300-kilovolt-ampere transformer in from 24 to 36 hours, according to the amount of moisture present. With larger transformers, while the oil is boiling I usually circulate it by means of a small centrifugal pump. In 500 kilovolt-ampere or smaller transformers, an occasional stirring with a long-handled paddle is sufficient.

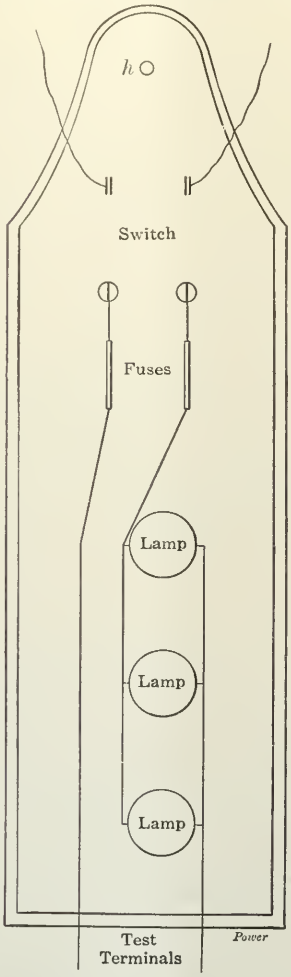
With transformers which do not have cooling coils, I dry out the case and coils by a blast of hot air from a small blower which forces air through a pipe embedded in a "salamander" into the base of the transformer case. To dry the oil I have a light tank large enough to hold about ten barrels of oil. This has a steam coil in the bottom, and the oil is boiled by passing steam through the coil. When the oil is dry it is transferred by the small centrifugal pump to the transformer case, which has been previously dried, and the transformer is ready for service. The tank is portable and taken from one job to another as needed. By using steam there is no danger of overheating the oil, the oil is kept perfectly clean and the only thing to be careful of is that the steam coil does not leak.

Earl R. Filkins.

Milwaukee, Wis.

Polarity Indicators

The method of testing for polarity, described by R. T. White in a recent issue of Power, is one which I have used quite often, but to avoid all possibility of a short-circuit I always use as a protecting resistance two or three 16-candlepower lamps, connected in a multiple group, the group being in series with the test terminals. I rigged up a small board, as shown in the accompanying sketch, the board being of a size convenient for carrying it around, and having a hole *h* bored at the top to hang it up by when not in use. I also find this very handy when it is necessary to reverse the polarity of a dynamo. For this purpose I use only one lamp, and connect to any convenient direct-current 110-volt circuit.



General Utility Lamp Bank

I do not agree with Wesley McArdell's intimation that a potato is a more convenient indicator than the water. The potato is all right, and is generally conceded to be a staple article, but in 10 or 12 years' experience as an engineer I have never happened to be employed in a plant where potatoes were included in the stock of supplies; on the other hand, there is always water available, as well as a container for it.

B. A. Searle.

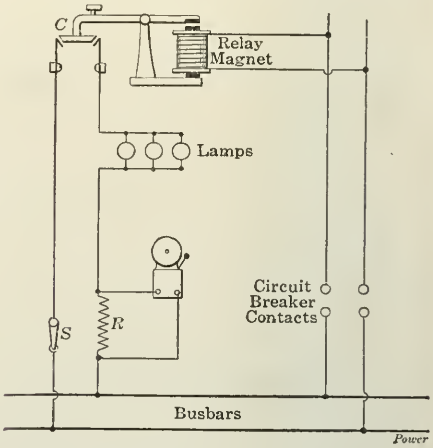
Independence, Colo.

Circuit Breaker Alarms

The circuit-breaker alarm described by Mr. Malcolm in the October 25 issue is open to several objections. To bring live connections onto the insulated handle is not very good policy and would not improve the appearance of the switchboard. The three lamps in series, unless of large size, would not pass enough current to operate the bell, and should one lamp burn out, the alarm would not operate.

Herewith is shown a diagram of an alarm which has no parts on the front of the switchboard and will operate when there is no voltage on the circuit to which it is connected. This arrangement has been in use in a large steel plant for two years and has never failed to operate.

The relay magnet coil is connected across the line terminals of the circuit-breaker and holds the copper bridge *C* out of contact with the alarm-circuit terminals until the main current fails, when the armature will drop the bridge across the terminals of the alarm circuit, permitting the generator to send current through the bell circuit and sound the alarm. By changing the number of lamps in parallel the current may be adjusted to operate any size of bell.



Mr. Blomberg's Alarm System

The relay magnet coil and resistance were obtained from an old motor starting box.

A single pole switch *S* cuts the alarm out when desired.

Carl A. Blomberg.

McKeesport, Penn.

"Fatty" Dinkelspeil went inter his biler a spell back an' when he tried t' crawl out he got stuck in th' hole. He's bin stickin' 'round there now fer th' last three days an' hez used up all uv th' cuss words in th' dickshunary besides a lot uv his own. He's jist finished th' last uv a dozen bottles uv anti-fat an' sez he thinks thet when he hez tuk a few more thet he kin break loose. "Fatty" is five feet high standin' up, an' six layin' down.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Gas Power Plant Costs

By R. S. Manning

What is probably the largest gas-driven station in this country supplying power for electric-railway work is located on the Wisconsin shore of Lake Michigan. The line supplied is about 112 miles long and the service is chiefly interurban.

The station equipment consists of two twin-tandem double-acting horizontal Allis-Chalmers gas engines supplied from Loomis-Pettibone bituminous producers and direct connected to Allis-Chalmers alternators rated at 1000 kilowatts each.

Both the engines and the alternators were designed to carry large overloads and have actually carried in service 1650 kilowatts for a considerable length of time.

The alternators generate three-phase currents at 25 cycles and 405 volts pressure, which is stepped up to 22,000 volts by a bank of seven 500-kilowatt transformers. For supplying nearby sections of the line, two 300-kilowatt rotary converters are used. Exciting current is furnished by two 50-kilowatt dynamos direct connected to three-cylinder vertical gas engines with cylinders 11 inches in diameter and a stroke of 11 inches. The usual switchboard equipment is provided.

Two double-generator sets of down-draft producers make the gas from bituminous coal. The generators are 11 feet in diameter and 18 feet high, the economizer boilers are 6 feet 6 inches in diameter, the wet scrubbers are 9 feet in diameter and the dry scrubbers are 12 feet in diameter and 12 feet high. Each double-generator set is rated at 2000 horsepower, but is capable of considerable overload.

Outside the building is a gas holder having a maximum capacity of 30,000 cubic feet, and this is provided with a 20-inch bypass pipe, with suitable valves.

The entire plant was erected by a firm of engineers whose fee was 10 per cent. of the aggregate cost of machinery, materials and construction, so that the final costs given in the accompanying table are 110 per cent. of the net costs of the installed plant.

The plant was arranged for the installation of a third engine unit of the same capacity and it was estimated that the cost of this unit, including foundation, piping and electric generator, delivered and erected, will not exceed $75,000. The whole station will then have cost $396,673.89 and have a rated capacity of 3000 kilowatts. The cost per kilowatt on rated capacity will therefore be about $132, but the cost per kilowatt on the maximum capacity of at least 1650 kilowatts per unit or 4950 kilowatts total will be about $80.

INSTALLATION COSTS.

Buildings, both producer and engine:		
Material	$ 24,415.64	
Labor	12,353.15	
Superintendence	370.00	
	37,138.79	
Engineering fee	3,713.88	
		$ 40,852.67
Machinery in engine house:		
Apparatus and material	195,421.01	
Labor	2,133.45	
	197,554.46	
Engineering fee	19,755.44	
		217,309.90
Machinery in producer house:		
Apparatus and material	65,993.07	
Labor	352.24	
	66,345.31	
Engineering fee	6,634.53	
		72,979.84
Machinery foundations:		
Material	4,680.43	
Labor	3,890.16	
	8,570.59	
Engineering fee	857.06	
		9,427.65
Piping complete, by contract, including labor	15,654.76	
Engineering fee	1,565.47	
		17,220.23
Contingent costs	5,666.91	
Engineering fee	566.69	
		6,233.60
Grand total		$364,023.89
Of the above, the rotary converters, substation switchboard, substation cables, substation step-down transformers main-line step-up transformers, material and labor cost	38,500 00	
Engineering fee	3,850.00	
		42,350.00
Total cost of generating plant		$321,673.89

Complete operating-cost figures are not available, largely for the reason that accurate power measurements have not been made because the load is subject to such frequent and heavy fluctuations. Perhaps a fair indication of the fuel rate may be gained from the record of an output of 14,025 kilowatt-hours in 19½ hours from one unit. On a basis of 1000 kilowatts rating this is a load factor of 72 per cent. average, while on a rating of 1650 kilowatts, which is much nearer the customary rating of gas engines, the load factor would be only about 43½ per cent. The total coal consumption for this output, including that necessary to put the producer in the same condition as at starting and after cleaning the fires, was 27,672 pounds, or about 1.97 pounds per kilowatt-hour.

The coal burned during this run was Eagle River (W. Va.) bituminous, of which the following is a representative proximate analysis:

	Per Cent.
Fixed carbon	65.45
Volatile matter	30.05
Moisture	1.20
Ash	2.75
Sulphur	0.55
	100.00

This coal costs $3.40 per ton in cars delivered at the plant.

Considering the low rating given the engines and generators, the results are very creditable and the first cost moderate.

Elementary Lectures on the Gas Producer

By Cecil P. Poole

Specific Heat of Gases

This lecture is not going to be near so fascinating as a detective story—in fact, it may be rather dry reading in several places, but if you have any desire to become a competent gas-power engineer, you must know about the specific heat of gases and some other things that may seem at first uninteresting. Just as a minor instance, if you did not understand specific heat and its effects, you would not know why producer gas should leave the generator as cool as possible, and merely telling you that that is so would not make much impression on your memory. Then again, you would not be able to understand some things about generator temperatures that you simply must know in order to handle a producer plant intelligently. So here goes for specific heat first.

The specific heat of any substance—wood, coal, brick, water, air, gas or anything else, is the quantity of heat required to raise the temperature of *one pound* of that substance *one degree*, Fahrenheit. Gases, however, have two specific heats, according to whether they are allowed to expand when heated or are confined to a constant volume. That is, if the gas is in a closed vessel and heat is applied to it, a certain quantity of heat will raise the temperature one degree, but if the gas is free to expand against the pressure of the surrounding atmosphere, more heat will be necessary to raise its temperature one degree, because the gas does work when it expands against the atmospheric pressure, whereas it does none when it is so enclosed that it cannot expand.

These two specific heats are called specific heat at constant volume and specific heat at constant pressure. When the term "specific heat" is used, without stating any quantity of gas, it is understood to refer to one pound. It is frequently convenient, however, to consider the heat required to raise the temperature of a given volume of gas or air, especially when it is confined so that it cannot expand, and when this is done, it is always stated as the specific heat *per cubic foot,* or whatever the unit of volume may be. For this reason Table 9, giving the standard specific heats by weight, has been extended to give the specific heats per cubic foot at constant pressure and constant volume. The weights per cubic foot are also given. All of the figures are based on a temperature of 32 degrees, which is the standard temperature for calculations of this sort, as explained last week.

TABLE 9. SPECIFIC HEATS AND WEIGHTS OF GASES.

	Specific Heat Per Pound.		Pounds Per Cubic Foot.*	Specific Heat Per Cubic Foot.*	
	Constant Pressure.	Constant Volume.		Constant Pressure.	Constant Volume.
Hydrogen	3.409	2.412	0.005589	0.01905	0.01348
Oxygen	0.2175	0.155	0.08921	0.01940	0.01383
Nitrogen	0.244	0.1727	0.07831	0.01911	0.01352
Air	0.2375	0.1685	0.08073	0.01917	0.0136
Steam	0.48†	0.36†	0.0502	0.0241	0.0181
Methane	0.593	0.4683	0.04464	0.02647	0.0209
Ethylene	0.404	0.333	0.07809	0.03155	0.026
Carbon monoxide	0.248	0.176	0.07831	0.01941	0.01378
Carbon dioxide	0.217	0.1535	0.1227	0.02663	0.01883

*Measured at 32 degrees and atmospheric pressure.
†At temperatures above 212 degrees and atmospheric pressure.

When a gas is heated, if it is free to expand against the pressure of the atmosphere, it will increase in volume; starting at the standard temperature of 32 degrees, the increase in volume is 0.00002035 per cent. for each degree by which its temperature is higher than 32 degrees; if it is confined so that it cannot expand, then its pressure will increase 0.00002035 per cent. with each degree of temperature increase above 32. This seems a small figure, but when the temperature runs up into hundreds and thousands of degrees, it counts up. For example, if the temperature be 1032 degrees, a cubic foot will increase to 3.035 cubic feet, if the gas is free to expand; if it is not free, the pressure will increase to 3.035 times the original pressure, which, of course, is by no means a negligible increase.

When the composite gas made by a producer issues from the generator at a high temperature, loss of heat energy is entailed as compared with the results obtained when the temperature of the delivered gas is lower, because, as just explained, heat units are used up in raising the temperature of gases, and the higher the temperature, the more heat is rendered useless for other purposes. For example, suppose a producer was delivering 10,000 cubic feet of gas an hour at a temperature of 1000 degrees, and the composition of the gas was as follows:

Carbon monoxide	22 per cent.
Hydrogen	13 per cent.
Methane	1 per cent.
Carbon dioxide	7 per cent.
Nitrogen	57 per cent.

The temperature of the gas would be 968 degrees higher than the standard temperature of 32 degrees, hence its volume would be (1 + 968 × 0.002035) = 2.97 times as great as it would be at 32; therefore, the volume at the standard temperature would be

$$10{,}000 \div 2.97 = 3367 \text{ cubic feet.}$$

Then the weights of the various gases, and their specific heats at constant pressure, would be as follows:

Carbon monoxide	741 cu.ft. × 0.07831 =	58.02 lbs.	Wgt. × 0.248	=	14.39 B.t.u.
Hydrogen	438 cu.ft. × 0.00559 =	2.45 lbs.	Wgt. × 3.409	=	8.35 B.t.u.
Methane	34 cu.ft. × 0.04464 =	1.52 lbs.	Wgt. × 0.593	=	0.90 B.t.u.
Carbon dioxide	236 cu.ft. × 0.1227 =	28.96 lbs.	Wgt. × 0.217	=	6.28 B.t.u.
Nitrogen	1918 cu.ft. × 0.07831 =	150.20 lbs.	Wgt. × 0.244	=	36.65 B.t.u.
Totals	3367 cu.ft. × 0.07162 =	241.15 lbs.	Wgt. × 0.2761	=	66.57 B.t.u.

The sum of the specific heats of the several gases is 66.57 and this means that for each degree of temperature to which the gases are heated, 66.57 B.t.u. are tied up. Consequently, if the same gases issued from the generator at 700 degrees instead of 1000, the gases would be heated to 300 degrees less temperature and there would be a saving of

$$66.57 \times 300 = 19{,}971 \text{ B.t.u.}$$

per hour. Now, 2545 B.t.u. per hour are the equivalent of one horsepower, so that the reduction of temperature would mean a saving of

$$\frac{19{,}971}{2{,}545} = 7.8 \text{ horsepower}$$

in the form of heat energy.

The foregoing example was worked through on the basis of the usual specific heats (per pound), but it could have been shortened considerably by using the volumetric specific heats at constant pressure in the fourth column of Table 9 instead of first calculating the weights of the constituent gases and then their specific heats, and the results would have been a trifle more accurate, theoretically. The accuracy, however, is not a consideration, because gas analyses and specific heats generally are open to suspicion beyond the third digit. Figuring in volumetric specific heats:

Carbon monoxide	741 cu.ft. × 0.01941 =	14.38
Hydrogen	438 cu.ft. × 0.01905 =	8.34
Methane	34 cu.ft. × 0.02647 =	0.90
Carbon dioxide	236 cu.ft. × 0.02663 =	6.28
Nitrogen	1918 cu.ft. × 0.01911 =	36.65
Specific heat of the producer gas		66.55 B.t.u.

The result, it will be noticed, is the specific heat of the 10,000 cubic feet of gas being considered, and, as just explained, it means that this number of heat units was used in raising the temperature of the total quantity each degree. Comparing the two methods of figuring again, the second method, dealing with volumetric specific heats, while requiring less calculation, is open to the objection that it does not give the specific heat of the gas per pound, while the first method gives the figures from which that can easily be determined. Thus, the weight was 241.15 pounds and the total specific heat 66.57 B.t.u.; the standard specific heat, therefore, is

$$\frac{66.57}{241.15} = 0.2761 \text{ B.t.u.}$$

As "specific heat" means heat per unit, it is evidently rather absurd to be talking of "total specific heat," meaning the heat required to raise the temperature of a given number of pounds or cubic feet one degree. The term "sensible heat" is in general use as meaning the quantity of heat that has been taken up by the gas or air in raising its temperature; this, however, does not mean the heat per degree of temperature increase, it means the total heat used in raising the temperature from the standard temperature of 32 degrees. What we are talking about, therefore, is the sensible heat per degree of temperature increase, which may be conveniently shortened to "sensible heat per degree." In the example that has just been given, the sensible heat per degree was 66.57 B.t.u. for the mass of gas considered; namely, 10,000 cubic feet.

To emphasize the distinction between the various kinds of heat that have been mentioned, perhaps a little list of definitions is advisable.

Specific heat: Heat units required to raise the temperature of one pound one degree.

Volumetric specific heat: Heat units required to raise the temperature of one cubic foot one degree.

Sensible heat: Heat units used in raising the temperature of a given weight from 32 degrees to the existing temperature.

Sensible heat per degree: Heat units required to raise the temperature of a given weight one degree.

Volumetric specific heat = Specific heat × weight per cubic foot.

Sensible heat per degree = Specific heat × weight of gas.

Sensible heat = Sensible heat per degree × temperature above 32 degrees.

Besides the disadvantage of wasting heat, as described, the delivery by a generator of gases at high temperatures entails the additional cost of excessive cooling in the scrubber. Of course, this is not so expensive a waste as that of carrying away a lot of heat that might be utilized in the generator, but it is a practical disadvantage. A further objection to high delivery temperatures, when bituminous coal is being gasified, is that more of the hydrocarbons are "broken up" into hydrogen and carbon dust, or lampblack, as explained in the last lecture, because high temperature in the issuing gas means that the top of the generator is hotter than necessary, and that is where lampblack is produced. The matter of generator temperatures, however, is sufficiently important to be given a whole lecture of its own, and that will be done the next time.

Joints and Gaskets on Gas Engine

By Frank E. Booth

The joint surfaces on the cylinder and cylinder head are usually machined with a rough thread-like finish to receive the gasket, which offers a much greater resistance to blowouts than was obtained with the old-time smooth surface.

The surfaces should not be too broad, as this distributes the pressure exerted by the studs over a large area, with a consequent reduction in pressure per square inch. On cylinders in which the head closes both the cylinder bore and the water-jacket space, trouble is sometimes caused by the packing giving way between the bore and the jacket, the water finding its way to the cylinder and interfering with operation. To obviate this difficulty some makers when facing the cylinder cover leave it a few thousandths higher on the part butting against the cylinder wall than on that next to the jacket wall.

When packing persists in giving out between two smooth surfaces, it is sometimes possible to get good results by chipping a groove around the end of the cylinder barrel, concentric with the bore. Ground joints for cylinder covers and valve cages give the least trouble.

On square or oblong surfaces, such as the packing surfaces of handhole plates on water jackets and inclosed crank cases, a thread-like finish is of no advantage unless it follows the contour of the plate; since this is not readily machined some makers cast the grooves in the plate and grind the surface straight. If the holes in the gasket have not enough clearance, the gasket will bunch up around the studs and prevent the cover from coming down solid.

If graphite be rubbed over the face of a gasket resting against the cylinder before the joint is closed, the head may be removed without breaking the packing. If the ordinary gasket cut from sheet asbestos be dipped in boiled oil before putting it in place it will give better results; the effect of the oil seems to be to toughen the asbestos. When a thick gasket refuses to hold between two smooth true surfaces an asbestos paper one will usually fill the bill.

When the packing gives out under the exhaust manifold on a multi-cylinder engine and the surfaces on the cylinders are found to be slightly out of line, use thick asbestos packing well soaked in water and pulled down solid while wet.

When pulling up the nuts on cylinder-head studs, go around the circle tightening each nut a little at a time, and after the engine has run for a short time, tighten the nuts again.

CORRESPONDENCE

The Armengaud Torpedo Turbine

With reference to F. R. Low's interesting notes in the issue of November 1 on the Armengaud gas turbines used for torpedoes, I presume that the power required to compress the air is included when reckoning the oil consumption per brake horsepower. If I am right, then one brake horsepower-hour is obtained at the expense of 0.9 pound of oil plus the energy in the compressed air. I would like it if Mr. Low would make this clear, because the success of this torpedo turbine is probably due to its utilization of previously compressed air. A gas turbine which had to compress its own air would be in a very different position.

R. M. Neilson.

Glasgow.

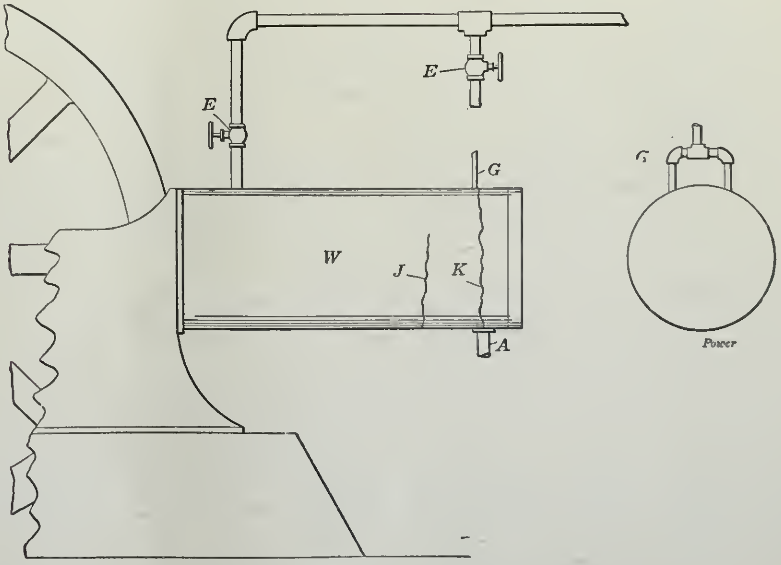
[The compressed air used by the turbine is previously compressed and stored in a tank in the torpedo hull. The fuel consumption stated probably does not cover the power expended in compressing the air; that is, the turbine uses 0.9 pound of oil plus the energy in the compressed air for each brake horsepower-hour developed.—Editor.]

Working with a Cracked Water Jacket

The accompanying sketch shows a gas-engine cylinder that was kept in commission after having two bad cracks *J* and *K* in the outer jacket wall; the crack *J* was on the outside of the water jacket and the larger crack was on the inside of the wall, starting at the top and running three-fourths around over the water inlet *A*. This crack *K* was about two inches from the end of the piston stroke and was patched with a plate, asbestos and cap screws, but the patch could not be made tight enough to stand the pressure of the water entering at *A*, and would leak enough to short-circuit and stop the engine. The water was then reversed and run in on top through the pipes *E*, *F* and *G*, and out at the bottom *A*. In this way the engine was kept cool and ran nicely for more than six months.

F. W. Sanders.

South Charleston, W. Va.



Working with a Cracked Water Jacket

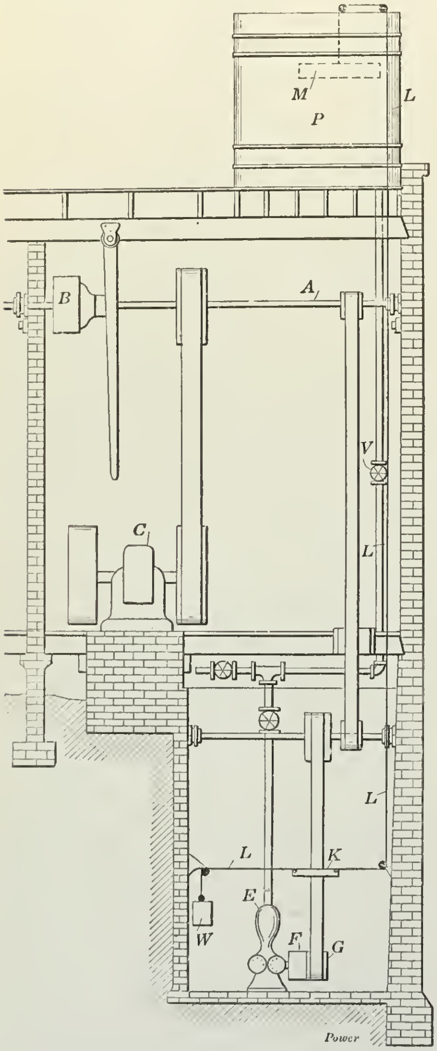
Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Automatic Water Supply System

The accompanying sketch shows an automatic water-supply system for small manufacturing plants, similar to one I installed several months ago which is now in operation. This system can be installed at a very small cost, and will be found to be very economical.

In the elevation shown, *A* represents the main line shaft for operating the machinery of the factory; *B* a clutch for throwing on and off the power; *C* a gas engine, or motor, belted to the shaft *A*; *E* a pump driven from the main shaft by means of belts and a jack shaft; *F* and *G* a loose and a fixed pulley; and *K* a belt shifter.



ARRANGEMENT FOR AUTOMATIC CONTROL

When the level of the water in tank *P* reaches the lower limit, the float *M*, through the cord *L*, pulls *K* over, thus shifting the belt onto the fixed pulley and putting the pump in operation. When the tank fills and the float reaches its upper limit, the counterweight *W* pulls *K* in the opposite direction and the belt is thrown back onto the loose pulley and the pump stops. Any desired water level may be maintained in the tank by adjusting the length of the cord.

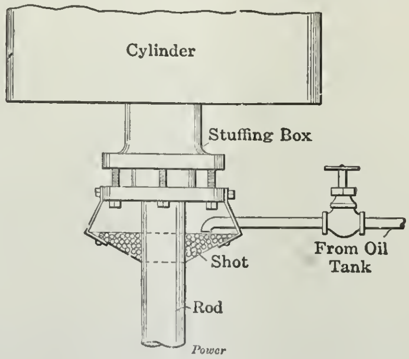
The advantages obtained by the use of this system are even water pressure without pulsations, the possibility of increasing the pressure in case of fire by closing valve *V*, the possibility of having water when the pump is not running, and the elimination of the loss of power due to the pump working against a high friction head, which is the case when only a small amount of water is being used.

F. B. HAYS.

Indianapolis, Ind.

Oiling Vertical Piston Rods

The accompanying sketch shows a simple and effective device for oiling the piston rods of vertical engines. A cone-shaped pan, having a hole slightly larger than the rod to be lubricated, is attached by suitable lugs or straps to the stuffing box, or bottom of the cylinder. The pan is filled with buckshot, onto which oil is fed from any available source. A sight feed placed in the pipe adds to the convenience of the arrangement.



OILING DEVICE IN PLACE

The upward and downward movement of the rod causes the shot to roll around against one another, distributing the oil and depositing it in a film on the surface of the rod.

C. J. LARSON.

Dubuque, Ia.

Why the Turbine Failed to Carry Its Load

A large cotton mill in the South was using what in that locality was called "second-class power." This is the term given to power that is sold at a cheap rate but its users are the first to suffer or be cut off in case of any disablement of the central station, which in this case was a hydroelectric plant.

The firm received notice that in twenty-four hours it would be cut off and that it would be obliged to wait at least two months before being supplied with any more power from the central station. Therefore it became necessary to put its own plant speedily into condition in an attempt to continue the operation of the factory.

The business had somewhat outgrown the plant, but the boilers were fired up and the plant started. The equipment consisted of two Babcock & Wilcox boilers and one steam turbine. As everything seemed all right, the turbine was brought up to sped and the load thrown on. After a few minutes, however, it began to slow down and every time the load was put on the turbine would stall. As the load was no heavier than had been carried previous to shutting down the plant, it seemed a puzzle why it would not again carry the same load. Therefore, an expert was sent for. After going carefully over the ground and finding nothing wrong with any part of the turbine, he went to the boiler room and immediately located the trouble. The boilers had been used for heating and to furnish steam for some other purposes but had not received much attention. Many of the tubes were almost completely closed with scale and it took a crew of men one week to clean those of one boiler, after which the load was easily carried.

EDWARD T. BINNS.

Philadelphia, Penn.

Trouble with a Vacuum Pump

Considerable difficulty was experienced with the vacuum pump on a condenser attached to a steam turbine. This pump was 12 and 36 by 18 inches and when starting up would run until a vacuum of about 20 inches was registered on the gage, after which it would gradually slow down and finally stop.

The piston was examined and found to be all right as was also the air cylinder. The decision arrived at was that the hot air from the condenser coming into the air cylinder expanded the piston, and the cylinder, being cooled by cold water, did not expand, hence the piston fitted too snugly.

However, this pump took steam from the end of a steam line from which four large, heavily loaded engines also received their supply and it was found that these engines used steam so fast that the pump just managed to get enough to enable it to pull a 20-inch vacuum and then would stop.

A line direct from the boilers was put in and there has not been any trouble since.

C. A. HELMAN.

Philadelphia, Penn.

Introducing Solvents into Boilers

While looking over various issues of POWER, I noticed that the subject of scale in boilers is often referred to and different qualities of boiler solvents are discussed, but very little is said regarding the method of introducing them. Some engineers introduce the solvent when washing out their boilers by way of the manhole. Of course, in such a case no extra expense is involved; but, as a rule, the periods of washing out are so far apart that the use of the solvent at these times only amounts to very little, for, in the natural course of running, it will be blown out and lost long before the next charge can be introduced. If the boiler is put out of service as often as the solvent is to be introduced, an unnecessary expense is involved.

In my opinion it is much better to introduce the solvent in small quantities at short intervals than to put in a large quantity at once; also if the water is very bad, a much larger quantity could be introduced more systematically during a given length of time and with better results. For example, with some waters, a charge of 30 pounds of soda ash once a month might cause serious trouble for a while after it was put in, but six or seven pounds a week would not injuriously affect the working of a boiler; and, moreover, would be beneficial.

In my boiler room, I have a convenient but simple and inexpensive rig that works to perfection. On the suction side of the feed pump a tee is placed, one branch leading to the feed-water supply and the other running in the opposite direction and turning down toward the floor, a valve being installed in the horizontal pipe of both sections. When it is necessary to introduce a charge of solvent the pipe leading down to the floor is removed, the solvent is dissolved in a pail of water which is placed under the pump connection, the piece of pipe that was removed is replaced and by closing the valve on one branch and opening the other the solvent is drawn into the pump and forced into the boiler with very few strokes of the pump; then the valve may be closed and the pail removed until wanted again. The same attachment can be made to the suction of a power pump or to an injector. Of course, owing to difference in circumstances which would be found in different places, the details of the connection might vary somewhat, but the principle would remain the same and any ordinarily intelligent engineer should be able to rig it up without difficulty and at no particular expense other than that of a few fittings which are usually carried in stock.

CHARLES H. TAYLOR.

Bridgeport, Conn.

Old Boilers in Massachusetts

In spite of the fact that the authorities in Massachusetts for the past twenty years have endeavored to protect the public from the dangers arising from the operation of boilers of faulty construction, there are still a great many in service whose age alone would seem to be sufficient reason for their being condemned or should at least occasion investigation as to why, violating as they do many of the rules that are rigidly enforced in new installations, they are allowed to remain a menace to life and property.

In most laws enacted to protect the public, such as the enforced adoption of air brakes, car fenders, automatic couplers, etc., a certain time was allowed in which to comply with them, after which a penalty was imposed for noncompliance, and the result is that the public is receiving the benefits intended. The boiler rules of Massachusetts contain no such provision and so long as a boiler installed previous to May 1, 1908, remains on its setting, it may be subject to many violations of the rules that would not be allowed in a new installation or even when relocating the same boiler. It is strange logic which prohibits the construction of boilers over a certain diameter having the lap joints and allowing boilers having that type of joint to continue to become old and menace the public with whatever danger the law sought to avert.

An example of this can be found in an old boiler, possibly the oldest in service in Massachusetts, which is located in Lynn, and which, although patched and rusted and possessing many of the prescribed evils, is still doing duty in the service of a large corporation. From information obtained from an old employee, this boiler furnished steam to drive the old "America," a locomotive purchased in the late 60s by the Eastern railroad and gradually descended from the dignity of hauling the fast express trains to the locals, then to a switch engine and then to furnishing steam for heating cars in the storage yard.

The inspector's certificate furnishes the following data: Age, unknown; length, 16 feet; tensile strength, 45,000 pounds per square inch; thickness of shell plates, ⅛ inch; type of joint, single-riveted lap; strength of joint, 49 per cent.; type of boiler, locomotive; diameter of shell, 48 inches; thickness of head, ½ inch; pressure allowed, 52 pounds.

The Massachusetts boiler rules provide that for this type of boiler a factor of safety of 5 shall be used in determining the safe working pressure. Using the formula

$$\frac{\left(\begin{matrix}\textit{Tensile strength} \times \textit{thickness}\\ \textit{of sheet} \times \textit{strength of joint,}\\ \textit{per cent.}\end{matrix}\right)}{\textit{Radius} \times \textit{factor of safety}} = \left\{\begin{matrix}\textit{safe working}\\ \textit{pressure,}\end{matrix}\right.$$

if the boiler were new a pressure of 68.8 pounds would be allowed, or 16.8 pounds more than is allowed after the boiler has been in service forty years.

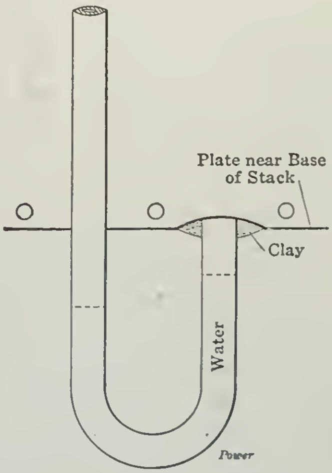
It would be interesting and possibly occasion some action in the matter if some of the readers of POWER would send in data and information concerning other old boilers, as it may be that I am wrong in assuming that this is the oldest boiler doing high-pressure service in this State.

CHARLES F. ADAMS.

Lynn, Mass.

A Homemade Draft Gage

The accompanying sketch shows how a practical engineer determined his stack draft without any expense. He took an ordinary water-gage glass and by heating it at the center bent it into the form illustrated. With a drift pin and hammer he then made a hole between the plates near the base of the stack. The glass was held in an inverted position and the U filled with water to the depth of a couple of inches. One leg was then inserted into the hole made by the drift pin and the space around the tube was filled in with clay to make it air tight. The readings obtained were plain and accurate enough for ordinary calculations.



GAGE INSERTED IN STACK

EDWARD T. BINNS.

Philadelphia, Penn.

Handling Men

Doubtless the most intricate factor in any manufacturing establishment is the human element. The relation of the different departments to one another, and of those in authority to their subordinates, has a far more vital bearing upon the economical operation of a plant than the performance of any mechanical apparatus in the department. Yet in most cases manufacturers leave the problems of labor and wages to work out their own salvation.

All costs of operation and production have been figured to a nicety, but when it comes to the management of men, and their scale of wages, the much needed system is often thrown aside, and the problem left unsolved from year to year; and always at an enormous cost to the company, as well as dissatisfaction among the employees. The frictional losses in the human machine are far in excess of those in a piece of mechanical apparatus.

Many times the chief engineer, by making small adjustments among the men, can produce large savings to the company when not nearly so much would be realized from weeks spent in the most careful adjustments of the various mechanical losses. I do not, in any way, wish to discredit the value of mechanical efficiency, or any of the many other things which fall to the lot of the painstaking engineer. But the chief should have perfect control of not only the mechanical element but the human element as well. He should be a man of character and strength, combined with kindness and good wishes for his men. He must be a man among men; a human force; a leader, not a driver; he should be to the human machine what the governor is to an engine; always keeping in touch with those under him and always ready to discuss any differences which may arise, giving credit to whom it belongs, and favor to none. Nothing will foster discontent, and throw the whole force into confusion, sooner than favoritism, harshness or injustice upon the part of those in authority.

The chief who tries to carry himself far above those whom he directs, forgetting that he, too, was once a subordinate striving for advancement, will soon find that his department is in need of better leadership.

The feelings the men have toward the firm are governed almost entirely by their feelings toward the man who directs them from day to day. The chief, as well as the head of the firm, must remember that the men have rights which must be recognized. The recognition of these rights, and the proper remuneration for their services, is the lubrication which keeps down the frictional losses in the human machine.

No work calls for more tactful attention than that of handling men. If the best services are to be obtained, the chief should have full authority to settle all questions which may arise in his department. He should be a man who appreciates effort on the part of his assistants, and he should discuss all topics with his subordinates which have anything to do with their welfare.

Efforts should be made to interest the men in economical methods of operation. One method which seems to give very satisfactory results to all concerned, is the inauguration of some well devised system by which the employer's gain is the employees' gain. This leads the men to feel that they are a part of the company, instead of just a cog in the wheel. In reality they are a part of the company, and should be recognized as such, for how long could any firm exist without the services of its employees? Give a man fair treatment together with what he earns, and a harmonious operation of all departments will be the satisfactory returns for the money invested.

J. M. Row.

Fort Flagler, Wash.

Pipe Vibrations

We were troubled for a long time by the vibration of our main steam header, a 14-inch extra-heavy pipe, about 45 feet in length. This shook so that the engineer had to run up the iron ladder and drive in wedges between the pipe and the wall.

It was finally decided that the vibrations of the pipe were in phase with some of the heavy reciprocating parts of the engines. To stop the vibrations, four heavy cast-iron weights were hung beside the pipe, these being suspended from the steel I-beam which formed the track for the traveling crane. These weights were then bolted to the pipe, the flanges being used as convenient points to which to bolt heavy iron straps which, in turn, were bolted to the weights. The weights stopped the vibration.

H. W. Randall.

Saginaw, Mich.

Uneconomical Operation of a Blower

Often when an engineer takes charge of a plant which has been in operation for some time, and in which changes and so called improvements have taken place, he finds auxiliary apparatus which is unsuitable for the purposes intended. Sometimes it is too large and other times too small; and particularly in the former instance, it is often difficult to convince the management that such is the case. Of course, it is better to err in having a piece of apparatus too large than in having it too small, but there should be some limit to size if economy is to be obtained.

A case of this kind came under my observation some time ago in which a horizontal balanced slide-valve engine, without any governor or regulator, was direct connected to a blower used for forced draft in the boiler plant. Owing to conditions existing at the plant, the steam pressure varied considerably and the fireman was accustomed to speed up the blower when the steam pressure dropped, regardless of the cause of the drop. By the time the pressure was considerably below normal the throttle would be nearly wide open.

When the load on the main engine was reduced or the normal steam pressure was restored through other means, the fireman was usually too busy with the fires and the water level in the boilers to look after the blower, with the result that occasionally the peripheral speed of the blower was sufficiently high to detach a blade, or damage the valve gear of the engine.

While, to a certain extent, the damage was the fault of the fireman, there were conditions over which he had no control, conditions which should have been made and kept right by someone higher up. In fact, the engine was three times as large as was necessary and this was primarily the cause of all the trouble. The fireman, knowing how hard it was to recover lost steam, usually speeded up the blower at the slightest drop in pressure, and instead of the pressure rising it often continued downward owing to holes being blown in the fires by the strong draft employed. Speeding the blower was the result of the fireman's education in that plant and he could not figure it out any other way but to do so when the steam pressure went down.

Some time after making the changes referred to, it was decided to rebore the cylinder of this engine and reduce its area by about 25 per cent. This was accomplished by inserting a liner and making a suitable piston, which was of the plug type, that is, one containing no packing rings. This change worked out satisfactorily in every respect.

About the same time, a throttling governor was also installed and set for a fixed maximum speed of about one-half that at which the engine had run previously. This eliminated most, if not all, of the troubles caused by excessive speed. It probably would have been impossible to reduce the speed by such an amount had other existing conditions continued, but changes were made in the operating force and the methods of operation and greater economy was attained.

In order to make the blower automatic, an ordinary stack-damper regulator was installed and connected to a cutoff valve in the steam pipe, which partially shut off steam when the boiler gages showed a certain predetermined pressure, never stopping the engine entirely, even though the pressure was near the blowing-off point, at which time it was running too slow to create much draft.

W. E. Rector.

Brooklyn, N. Y.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Rope Drives

I noticed in the October 25 issue an answer to my letter in the September 13 issue, referring to the design of rope drives.

The question of what speeds at which it is advisable to run cotton or manila ropes depends very largely upon the condition under which the drive is installed. There is no question but that centrifugal force has a great deal to do with the effective result upon a pulley or wheel using either belt or manila rope or any other material that is practically incompressible, and having a comparatively small friction coefficient. So soon, however, as the material is capable of being wedged into place, centrifugal force is, to a large extent, neutralized; and this is the condition which exists with cotton rope. Anyone who has watched a cotton-rope drive and a manila-rope drive will be struck by the difference in the way in which the ropes leave the pulley. There is a decided tendency of the cotton rope to continue in the groove, which does not exist with the manila.

A belt or manila rope depends entirely upon the friction due to the weight or pull of the rope or belt on the pulley or in the groove, and neither wedges into place.

The effect of centrifugal force even upon a belt or manila rope may be considerably neutralized on long centers by the weight of the belt or rope; but on short centers it has to be neutralized, if at all, by excessive tension on both sides of the drive. On the other hand, with cotton rope with more or less possibility of wedge, the necessary weight or pull may be considerably diminished; and the action of centrifugal force is somewhat neutralized by the wedging action, so that on long drives or short drives through the action of the weight or rope tension in addition to the wedging action of the rope in the groove, the cotton rope is much more effective on high speeds, and due to the wedging action the theoretical and practical values of the rope drive very nearly coincide, neglecting the action of centrifugal force; that is, manila rope acts almost exactly according to theory, whereas cotton rope does not.

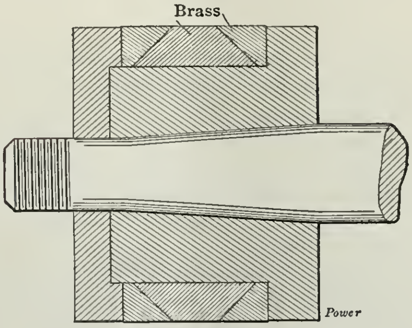
I quite agree with Mr. Davies that it is advisable to give full weight to the opinions of any eminent authority; but where a man has in his experience succeeded in getting certain results which are directly according to theoretical results, leaving out centrifugal force, it would seem advisable also to defer, not slightly but considerably, to his opinions. That is, it would seem to indicate that cotton rope can successfully be operated according to the theoretical value per rope at speeds up to 7000 feet per minute; whereas, on the other hand, it is found by practice that the limiting speed of manila rope is approximately 4800 feet per minute, which agrees very closely with the theory, taking into account centrifugal force.

HENRY D. JACKSON.

Boston, Mass.

Valves and Packing to Withstand Acid

In the issue of October 18, Frank Demarle asks about reliable packing and valves to use in an acid pump. I presume Mr. Demarle is using plunger pumps. In such a case the remedy is to use brass packing. The accompanying figure is self-explanatory. It should be noted that care must be taken about adjusting the nuts on the follower plate to prevent binding. When tightening the follower plate the outside rings are forced outward toward the cylinder. The two outside rings are split through with a thin hacksaw blade while the central ring is solid. All three rings must be bored to a neat fit on the plunger. This style of packing will be found to be decidedly better than flax.



BRASS PACKING FOR ACID PUMP

Regarding valves, use flat disks of brass having central pins to seat them true. Instead of only one week's service, Mr. Demarle will secure over a year's service, which will more than pay for the slight changes necessary to the plunger.

GEORGE H. HANDLEY.

Newburgh, N. Y.

License Laws

The letters on the subject of license laws and particularly those by J. J. Nigh are of much interest and they have led to some interesting and entertaining discussion in the columns of POWER. Without wishing to enter an argument on the merits of license laws, I would like to see some discussion on the question, "Should engineers defray the cost of bringing about license legislation?" A lecturer on license laws informed his hearers recently that rag pickers, swill gatherers, barbers, druggists, doctors and the members of a host of other vocations were licensed, and that it was for the good of the community that they were. Did all of these good people spend their time and money to bring about the enactment of laws which put a tax upon their pockets?

FRANK H. WILLIAMS.

New Haven, Conn.

From time to time articles have appeared in POWER for or against license laws, but none of them, in my opinion, hits the right spot. Our own law is a striking example of how license laws should not be used.

In 1892, I was appointed a committee of one by Queen City No. 1, National Association of Stationary Engineers, to try to get the city council to enact an ordinance to license engineers. Owing to the active and powerful influence of the plant owners, I was unsuccessful for three years in spite of the fact that I devoted all of my time and considerable effort and money to the purpose. In 1895 our city charter was revised and, having several warm personal friends on the revision committee, I succeeded in getting a license-law provision into the charter.

When it came to the appointment of an examining board, two obscure hardware dealers and a janitor were appointed to license the engineers. Again I was beaten and the law was dormant until 1900, when a fatal explosion occurred in which two men were killed. Then a demand for the enforcement of the law was made and three competent examiners were appointed. But, as soon as the scare had worn off a little, the personnel of the board was changed again and anyone who had the necessary $2 could get a license, whether he knew anything or not. I have actually seen Japanese that could not speak a word of English, possessing a license. I have seen men with licenses covering almost any plant in town come to work in a plant and ask the man there, upon being shown the feed pump, "Where does the steam go into this pump?" One man with the same grade of license came into my engine room and seeing me using graphite for some pipe work wanted to know what it was and where it could be bought.

The consequence of all this is that we have had more explosions since the license law was passed than we have ever had. I wish that I had never bothered my head about license laws. I would be hundreds of dollars and a good deal of time ahead, not to say anything of the worry and effort.

F. E. Albrecht.

Seattle, Wash.

Quarter Turn Drive

In the November 1 issue of Power, Mr. Bitterlich makes inquiry as to the design of quarter-turn drives for leather belts.

As a great many rules of various kinds have been proposed for determining the power of a belt, some varying from others by several hundred per cent., it is impossible to give any hard and fast rule that may be relied upon for ordinary straight drives. And, when the proper design of quarter-turn drives is desired, still less definite information is available and the best that is possible is a set of general rules.

The one basic principle in such drives, and which when once understood enables any drive to be designed properly, is that the center of one pulley must be in the same plane as the face of the other pulley. From this rule it will be seen that a belt after leaving one pulley must be delivered in line with the pulley it is running toward. The belt may be drawn sideways out of the plane of the pulley it is leaving, but it must always be delivered in the same plane as the face of the pulley it is running toward or else it will run off that pulley. It is thus evident that this drive will not run backward, and that if reversed the belt will leave the pulleys.

When a belt is pulled sideways, as is the case in this form of drive, the edge of the belt toward which the sideways displacement takes place tends to bulge out owing to the lateral stiffness of the belt and is no longer in close contact with the pulley. This action will decrease the contact and hence the driving power of the belt. To overcome this defect the angle at which the belt is drawn off should be kept as small as possible; also, in figuring the power of the belt a larger belt should be chosen than would be required for a straight drive transmitting the same amount of power. For greatest efficiency the drive should have the distance between shafts large compared with the diameters of the pulleys. Also, a wide belt should not be used.

Applying the above to Mr. Bitterlich's particular problem I should say that his drive would undoubtedly work, only the driving power would not be very large and could be materially improved by placing the pulleys further apart and using a 4-inch double belt, as suggested by the master mechanic, instead of the 6-inch single belt.

The speed of 1600 feet per minute is certainly on the safe side if the required amount of power can be supplied with that speed, but the speed should not much exceed 2000 to 2500 feet in order to avoid undesirable strains in the belt due to the change in lateral curvature where the belt is drawn off.

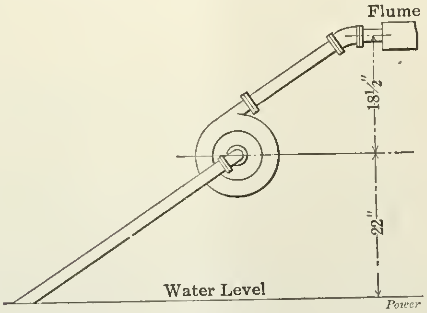
W. L. Durand.

Washington, D. C.

Irrigation Problem

In regard to the irrigation problem in the October 18, the proper way in which to put in the piping is shown in the accompanying figure. With this layout the friction in 55 feet of pipe, three 45-degree elbows and one 90-degree elbow is saved.

Approximately, each 45-degree elbow equals a foot of head and the 90-degree elbow 4 feet of head, thus making in all 7 feet. The one 30-degree elbow would add about a foot, so the net saving would be 6 feet of frictional head, and with 0.4 of a foot for the pipe, the net saving would be about 40 horsepower.



Improved Arrangement of Pump and Piping

William O. Webber.

Boston, Mass.

Central Station vs. Isolated Plant

It is a fact that some classes of engineers are being slowly but surely pushed out of existence by the large heat and power companies. You say that the agent of the company makes a flattering proposition, shows figures as to what it costs now to keep an engineer and other help in the isolated plant, how heat and power can be furnished without dirt, ashes to be removed, how the basement can be used for other than engine-room purposes and how the smoke nuisance can be eliminated. I could mention a hundred or more reasons why the central-station agent gets the willing ear of the owner or manager, and as sure as the sun will rise, the very existence of all of us will ere long be threatened. Now, what are we going to do about it? Shall we do as is usually recommended, get indicators, hot-water meters, calorimeters, etc., make thorough evaporation tests of the boilers, show the heat value of the fuel, the percentage of ash, the temperature of the flue gases, etc., which really is the only correct way in which to ascertain the cost of operating a plant? But who can afford to do this at salaries which are now paid in just such places where the danger of being pushed out is the greatest, where engineers get from $2.50 to $4 per day? The owner of the plant would not buy these instruments and the engineer cannot save enough from his earnings; so you see it is much easier to recommend than it is for the engineer to follow.

I want to suggest a way in which the cost can be approximately ascertained without the use of the proper instruments. The owner pays all the bills and that is the basis on which he forms his conclusions. The engineer should start by seeing that he gets a ton of coal weighing 2000 pounds. Start on that ton, have no other fuel, burn up that ton of coal and wheel out the ashes, weigh and record the amount, the date, the name of the coal dealer and the quality of the coal; then, keep a record of the load; if electrical, the ammeter tells; if heating, figure the total cubical contents of the rooms; if refrigerating, the procedure is the same; then, keep a record of the wages and labor cost, the lubricating cost, the supplies and repairs.

When the agent makes his "spiel" after he has carefully scrutinized all the bills, he uses his most expressive statement, "And last but not least, Mr. Owner, taxes, depreciation and insurance," he will wave his hand like a magician and say, "we do not even count them, just your own figures for actual expense." He quickly ascertains the average load and divides it by the cost, and here you are, so much per kilowatt-hour. Then it is up to the engineer to have his say, which should be something like the following:

"Now, Mr. Owner, allow me to show you why the power cost is so high and how it can be reduced lower than the agent's figures. First of all, the boilers are not clean enough. Then, I find that the coal you bought from so and so gave better results than the coal you bought from the other concern. If you will buy a good boiler compound to keep my boilers perfectly clean, I will reduce the coal bill all the way from 10 to 25 per cent. Then, I will save 20 per cent. more if I get that good coal, making a total saving of 45 per cent. Let us figure on last month's expense for the power plant at 45 per cent. saving and you will see that we can produce the power cheap enough. Then, if you will give me $5 per month more, and that is little enough, I will use it to buy a steam-engine indicator so that I can indicate my engine."

You may see a change in the owner's color, but do not be afraid; he will not faint, so continue:

"Then, I would like you to buy a thermometer to screw into the boiler-feed pipe so that I can keep a record of the temperature of the boiler feed; the water should be nearly 212 degrees Fahrenheit; every eleven degrees added to the water means 1 per cent. saving. I will also ask you to get a thermometer to put in the breeching, so that I can measure the flue-gas temperature, as I want the gases to escape at as low a temperature as the draft will stand, about 500 degrees. I will make a draft gage to see if there is sufficient draft to burn the coal economically, say not less than ¾ inch. Really, Mr. Owner, if you take a little more interest in me and the plant it will be money in your pocket, and the agent's second visit will not last five minutes. But should he come again and you still entertain any of his various propositions, let him make a test of all the plant, tell him you do not want him to guess about it; you want to know what each department will cost; you want him to show you how much water is evaporated; have your engines indicated and have all this done under the supervision of your trusted consulting engineer."

The owner will then begin to realize that he has a man in the engine room who represents him; who will always look out for his interests; who will, if given half a chance, make dollars for him.

William Nottberg.

Kansas City, Mo.

Trouble with Metallic Packing

In the issue of November 1, we notice a letter on page 1944 headed "Trouble with Metallic packing," in which it is said that a 100-horsepower engine was packed with metallic packing over one year ago. The writer states that the packing was composed of soft metal in small grains containing considerable graphite. The packing, being in cotton tubes, was placed in the stuffing box with a copper wire around the piston rod inserted before the packing, and a ring of fiber packing on top.

We manufacture a packing which in some ways corresponds with that described by Mr. Chapman.

We write this, not to criticize him, but in hope of enlightening him and other readers of Power on the use of plastic metallic packing, which we have been manufacturing for many years.

Mr. Chapman states that the packing gave no trouble all last year, and that the engine ran nicely. But, after being idle for about three months and started again, the packing began to leak, and then the gland was tightened only slightly, which partially stopped the leak, and, having no more of the packing on hand, the engine was allowed to run with the small leak.

If there had been more of the packing on hand to add to the old and properly fill the stuffing box, the leak could no doubt have been stopped at once. The fact that the packing did not leak during the first year or more would indicate that it was compressed into a steam-tight joint, although it was probably not thoroughly compressed but in more or less of a honeycombed condition, and the action of the hot piston rod gradually compressed the packing, until it became loose in the stuffing box, causing the leak and allowing the copper-wire bushing to become dislodged, leaving an opening around the piston rod through which the packing could be drawn into the cylinder.

This packing should be thoroughly compressed in the stuffing box, by keeping the gland snug against the packing, following it up frequently for the first hour or two. After the packing is properly compressed, the ordinary pressure of gland is sufficient, and the packing will last indefinitely without renewal or special attention. If the packing had been thus compressed, the leaking would have been avoided and the copper-wire bushing would have been held in place, preventing the packing from being drawn into the cylinder.

In using this class of packing it is very necessary to have a sufficient quantity on hand to properly fill the stuffing box, allowing for about one-third compression of the packing.

The packing should be composed of an alloy of antifriction metals, just soft enough so that it cannot injure the piston rod, and just hard enough for durability. The metal should be thoroughly mixed with the best of graphite and other lubricating materials, so that they will remain permanently in the packing. It will be seen that this packing could not possibly heat or injure any piston rod by the pressure of the gland.

When the piston rod has passed through the plastic metallic packing, the packing must necessarily fit the rod exactly, without binding, its form remaining permanent, making a packing that is self-lubricating and practically frictionless.

When the packing wears so as to leak, more of the same packing may be added, and the leak will stop.

Steel Mill Packing Company.

Detroit, Mich.

Metal for Cylinders

This important and interesting subject treated in the November 1 issue by Mr. Rector is one of the subjects much overlooked by the engine-room man, although the chemical make-up of the iron of the cylinders is a little out of the ordinary line for the engine room. Those who work in the engine room often have the chance to notice the difference in the wearing quality of an engine cylinder.

I recall sending out some men to bore a 20x48-inch Corliss engine cylinder. The cylinder was one composed of soft open-grain iron. A piston and new rings were fitted to the cylinder and the engine was started. The following day I was called by telephone, and notified that the cylinder was badly cut. Upon investigation I found that it was so badly scored that a light cut had to be run through the cylinder. A new head was made as well as new rings. The first head was made of iron which was too open and soft, it seemed to stick and cut. The second head was made of an iron having a closer grain.

I have found that a cylinder and rings should not be of like composition. The close-grain cylinder is more easily handled after getting started, while a soft, open-grain cylinder high in carbon will start off better. Mr. Rector asked about the steel-mixture cylinder. The use of steel scrap in the making of engine cylinders is very common. When this is done an iron much like charcoal iron is secured; however, much depends upon the skill used in the process. The so called semi-steel cylinder is a misnomer. Many people are still using such names when they know that it is not right. "Semi-steel," "ferro steel" and such like are all misleading names. When an engine cylinder is made by using steel in the melting furnace, the result is a reduction in the total carbon that the iron carries resulting in a close-grain iron. With a dense iron, of course, strength will follow density when the other metaloids are right. A low-carbon iron, such as produced by using steel punchings in the furnace, will make an excellent cylinder; in fact, cylinders for some of the most difficult duties are made in this way.

"Iron is iron" so long as it is suitable for the purpose for which it is intended and is of the proper composition. There is no difference between charcoal iron and coke iron so long as the analyses are the same. Charcoal irons generally are low in sulphur, a constituent that is hard to handle but somewhat under control in mixing steel with iron so long as a hot furnace is employed.

An iron low in manganese is not as suitable for a cylinder as one high in manganese, for manganese is a great strengthener when properly used.

For cylinders or castings needing a close texture, as in the case of steam cylinders, gas-engine cylinders, hydraulic cylinders, the following composition is suggested:

	Minimum. Per Cent.	Maximum. Per Cent.
Silicon	1.25	1.60
Sulphur	0.05	0.09
Phosphorus	0.40	0.55
Manganese	0.25	0.55
Combined carbon.....	0.75	1.50
Graphitic carbon.....	2.00	2.25

It is quite correct not to make bull rings and cylinders of the same composition. A good composition for the bull rings is:

Silicon from 2.25 to 2.60,
Sulphur from 0.10 to 0.13,
Phosphorus from 0.30 to 0.40,
Manganese 0.55.

This will make a good ring. The sulphur has a hardening effect but not enough to overcome the effect of the silicon which will make a soft but close grain; and this is what is wanted. Very true, many rings have worn out too soon because they were not made of the right iron.

The bull and snap rings should be soft as it is better for them to do the wearing and not the cylinder, for they can be renewed more easily than the cylinder. More care is needed in the selection of rings for high-pressure and high-speed work than for low pressure and low speed.

The composition of cylinders is quite an interesting subject for engineers, for it enables them to discover the causes of defects and to apply intelligent remedies.

C. R. McGahey.

Lynchburg, Va.

Common Sense

The principle of common sense, reasoned by Power in the first-page editorial of the October 18 issue, is one of too great import to pass unnoticed; it is the nucleus of worth, of a man's earning ability. Real common sense is symbolical of efficiency, and one is never present without the other. This principle appears in three degrees, near common sense, common sense, supernal common sense; they are seen in the majority, in the many, and in the few. It is the instinct of human nature to feel that one possesses common sense; but will logical reasoning admit of this? Do one's daily actions prove it? Narrow-mindedness, pedantry, sluggishness, prejudice, lack of rational argument, never indicate common sense; these are the direct means of decreasing a man's worth to himself or to his employer.

The ability to reason correctly, to do the right thing at the right time, to use considerate judgment, to prove results, to show efficiency—these are the attributes of business common sense; they signify merit, elimination of waste, working with intent of purpose, working for the company's interest. The efficient organization head displays common sense in choosing an efficient superintendent, the efficient superintendent by surrounding himself with efficient foremen, efficient foremen by having efficient subordinates, and the latter by efficient practice. Such a combination is born of common sense, else it would not be effective.

A display of common sense cannot help but mold progress. Application to duty, right principles, mind training, advancing abreast of the age, these precede a greater field for talents, business prosperity and greater earning powers.

L. R. W. Allison.

Los Angeles, Cal.

Installing Globe Valves

In the November 8 number there is a letter by James Johnson on "Installing Globe Valves."

According to my experience he did not use very good judgment in removing one of the valves in the blowoff line, although he says that he has had very little trouble since. We have tried single valves on the blowoff lines and always have had trouble from leakage.

For the last twelve years or more we have had two valves on every blowoff and have had no trouble from leakage, renewing the disks of the globe valves once each year. On two of our boilers we have asbestos-packed cocks next to the boiler, and angle globe valves beyond the cocks, which are opened first and closed last, using the cocks for blowing off.

On the remainder of the boilers we have angle globe valves next to the boilers and gate valves beyond. The outside valve will hold whatever water leaks past the first valve, and in case the thread should strip on the valve stem of the first valve, the second can be used until the boiler can be spared to put in a new valve, which is important in our case as it is almost impossible to cut out a boiler for about seven months in the winter. Some engineer who reads this may say, "How about cleaning the boilers?" We clean our boilers but once a year and the inspector says that they are in fine shape.

W. O. Perkins.

Bristol, Conn.

Diagrams for Criticism

In regard to the diagrams for criticism in the November 1 issue, Mr. Fryant has failed to give all the details of the conditions under which the diagrams were taken; hence I am compelled to offer several reasons for the jagged expansion lines. The type of engine, the steam pressure and the scale of the indicator spring should be given when a question of this nature is asked.

There may not be sufficient tension in the indicator cord on the reducing motion or wheel.

The operator might have failed to blow steam through his indicator two or three revolutions before taking the diagram.

There might have been water in the cylinder of the engine.

There might be lost motion in the engine or the reducing motion of the indicator.

I am under the impression that the jagged expansion is due to the spring in the indicator being too light for the steam pressure at the time the diagram was taken.

J. E. Strother.

Yuma, Ariz.

In the issue of November 1, I saw some diagrams which were submitted for criticism by H. J. Fryant. If he will relieve some of the cushion in the dashpots, he will find that the jagged expansion lines will disappear. I had a similar experience with a 14x36-inch Corliss engine running at 72 revolutions per minute. I attribute the trouble to the cushion causing the dash rods to bound high enough to cause a partial opening of the admission valves and I found that on relieving some of the cushion I had a smooth expansion line. However, I would like to hear from other engineers on this point.

G. E. Goodwin.

Burlington, Ia.

What Causes the Engine to Run?

In the November 1 number, E. R. Teer tells how his engine continues to run when the bleeder to the steam pipe is open. I should say that the two drain valves on the cylinder must be open also and that the pipe going into the exhaust is smaller than the bleeder, or that it is stopped up a little to cause the steam to go into the cylinder through the drain valves and start the engine. This is the way in which I have a 10x14-inch engine connected, only I have a valve on the exhaust leg of the bleeder which I close. This is an easy way of starting the engine, for when the steam enters the cylinder the valves work so that the pressure is always on the piston in the way the engine is set to run. I use ¼-inch pipe all round.

D. F. Crowther.

East Boston, Mass.

Experience of an Indicator Man

I would like to make some comments on the article in the November 15 issue entitled "Experiences of an Indicator Man." I judge the engine was a Corliss and that the cutoff rods to the steam-valve hooks had been shifted.

Now, when the engine laid down with the load the regulator would drop and if the cutoff was equal, both steam valves would stop cutting off at once, but if they were unequal one hook would hold on and the other would unhook, thus making the mean effective pressure very unequal in the two ends of the cylinder. But, at the same time, the dashpots on this type of engine are always very noticeable and it seems to me that even a novice would see where the trouble was without applying an indicator.

W. E. Hopkins.

Torrington, Conn.

Conducting a Steam Boiler Test

By H. S. Flynt

Points to be observed and the apparatus required are enumerated. A system of weighing the water is described and it is then shown how a test should be figured.

A boiler test is not a difficult undertaking, providing care is exercised in the arrangement of the simple preliminaries and the taking of accurate notes throughout the test. In conducting a test the following arrangements must be made, and apparatus obtained for making a simple evaporative test.

Weighing the Water

The blowoff cock of the boiler must be examined, and if the slighest suspicion of a leak exists it must be blanked off during the test. If a hot-water meter, newly tested and calibrated, is available it may be connected in the feed line between the feed water heater and the boiler and used to meter the water. As this apparatus is rarely kept on hand, except in the very large plants, the common and safe method of weighing the water in barrels is recommended. Three barrels are generally used, two being fitted with an overflow outlet close to the top, and a discharge pipe close to the bottom of each. They are carefully weighed on platform scales when filled with water up to the overflow pipe, and again when only filled with water up to the point level with the discharge pipe at the bottom of each barrel.

The barrels are then numbered and the calibration of each is noted in pounds. These two barrels are then placed upon a staging at a convenient hight for filling and discharging into the third barrel located below, from which the boiler-feed pump draws its supply. Each time a barrel is emptied into the lower barrel the measure of water should be marked with chalk upon the side of the barrel when its contents have been discharged, and always in this order; otherwise confusion in the count will result before the end of the test.

Apparatus Required

A reliable thermometer should be kept submerged in the reservoir barrel for taking the temperature of the feed water, while a reliable platform scale is required for weighing the fuel, the weight of the barrow or other holder being noted, and a uniform quantity of fuel weighed out each time and dumped on the floor as needed. A reliable thermometer, graduated to not less than 700 degrees Fahrenheit should be inserted through a hole in the uptake at some point close to where the gases leave the heating surface of the boiler. This thermometer may be removed from time to time, and a small iron pipe inserted, to which a water draft gage is attached by rubber tubing, for ascertaining the draft or "pull" of the chimney. This observation should be taken with the damper full open, half open and closed, and before and after fire cleanings.

Immediately before starting the test a piece of string should be tied round the gage glass at the hight of the water, and whatever the variation may be throughout the test, the water must be brought back to this mark as the test ends, which should not be of less than eight hours' duration, and should start and end directly after the fire has been cleaned. During the test 10 pounds of the fuel should be gathered from different parts of the heap and weighed carefully on a scale sensitive to one-half ounce, and then spread out on an iron plate on the top of the boiler setting for a few hours to dry. It is absolutely necessary to consider the performance of the boiler in comparative tests from the dry-coal basis, and by this means the approximate percentage of moisture in the fuel may be readily calculated. The ashes made during the test must be collected as carefully as possible without wetting, and from their total weight in relation to the total dry coal, the approximate percentage of incombustible matter in the fuel is determined.

What to Observe

Throughout the test the following observations should be made and noted on the record sheet every fifteen minutes:

Weights of coal and water, feed-water temperature, steam pressure and temperature of the escaping flue gases. At the close of the test, the figures under each heading are totaled and divided by the appropriate number of observations made for an average. From these averages the performance of the boiler, according to the standard conditions established by the A. S. M. E., may be calculated by aid of the steam and water tables. But the evaporation under actual conditions is merely a matter of simple division between the total water and total coal.

Figuring a Test

Assuming that the test has been with a 100-horsepower boiler of the horizontal return-tubular type, having a grate 6x5 feet, and operated by natural draft, and that at the completion of the test the following data have been collected:

Total water used was, 15,000 pounds; weight of coal fired in the damp state, 2200 pounds; average steam pressure, 75.3 pounds per square inch, and the feed-water temperature, 60 degrees Fahrenheit, both obtained from the quarter-hourly observations. It is assumed, for the sake of simplicity that the water was not passed through a feed-water heater before entering the boiler; but in such a case the temperature of the feed must be taken at the heater instead of at the reservoir barrel, and the subsequent calculation of the factor of evaporation based upon this temperature instead of the initial temperature at the barrel. The average temperature of the gases leaving the heating surface of the boiler was 400 degrees Fahrenheit. On reweighing the coal put aside to dry, the loss in weight due to expelled moisture was found to be eight ounces. The dry ashes resulting from the fire cleaning weighed 330 pounds.

Now, we find that 15,000 pounds of water were raised from 60 degrees Fahrenheit and transformed into steam at 75.3 pounds gage pressure by 2200 pounds of the fuel; therefore the equivalent evaporation under the actual conditions of the test was 6.82 pounds of steam for each pound of coal fired. But it is obvious that if the same quantity of heat had been applied to water at the boiling point (212 degrees Fahrenheit) instead of 60 degrees Fahrenheit and this had been transformed into steam at atmospheric pressure instead of 75.3 pounds gage pressure, considerably more water would have been evaporated.

There are 970 B.t.u. absorbed in transforming one pound of water from 212 degrees Fahrenheit to steam at 212 degrees Fahrenheit, or from a liquid into the gaseous state, at normal pressure. Referring to standard tables of the properties of steam we find the total heat in one pound of water at 60 degrees Fahrenheit to be 28, and the total heat in one pound of steam at 75.3 pounds, gage pressure, to be 1184.4 B.t.u. Subtracting the total heat of the feed water at the observed temperature from that of the steam at the observed pressure and dividing the remainder by 970 we have

$$\frac{1184.4 - 28}{970} = 1.192$$

the factor of evaporation; and on multiplying the total pounds of water used under actual conditions by this figure, the equivalent evaporation from and at 212 degrees Fahrenheit is obtained. Having found the approximate percentage of moisture in the fuel, in the manner described, it is an easy matter to find the pounds of water evaporated from and at

212 degrees per pound of dry coal.

The established standard for a boiler horsepower is 34.5 pounds of steam per hour from and at 212 degrees Fahrenheit, or, what amounts to the same thing in heat absorption, 30 pounds of water transformed from 100 degrees Fahrenheit into steam at 70 pounds gage pressure. Therefore, to express the performance of the boiler in terms of horsepower, divide the total water evaporated, from and at 212, by the number of hours, duration of the test, and the result by 34.5.

To find the pounds of coal used per horsepower divide the coal used per hour by the horsepower developed.

The fuel cost of the power per hour is in the same proportion to the cost per ton as the quantity used per hour to the ton. Assuming that we have been dealing with the short ton of 2000 pounds,

$$\frac{275 \textit{ lb.} \times 400 \textit{ cents}}{2000} = 55 \textit{ cents}.$$

Test Sheet

The finished results of the test calculations should finally be brought together in something like the following order, which is perhaps the most concise and comprehensible:

Date............19..

BOILER TEST AT............

Boilers, one 100-horsepower—H.R.T. type.

Grate 6x5 feet, area	30 square feet
Duration of test	8 hours
Fuel used	anth. pea coal
Price per ton	$4.00
Total coal burned	2.200 pounds
Coal burned per hour	275 pounds
Coal burned per hour per square foot of grate	9 pounds
Moisture in coal (8 ounces in 160 ounces)	5%
Ashes (proportion of dry coal), 300 pounds	14.35%
Total combustible in coal (approximate)	1,790 pounds
Steam pressure, average	75.3 pounds
Feed-water temperature, average	60° Fahrenheit
Factor of evaporation	1.192
Total water evaporated (actual)	15,000 pounds
Total water evaporated from and at 212 degrees Fahrenheit	17,880 pounds
Total water evaporated per hour from and at 212 degrees Fahrenheit	2,235 pounds
Horsepower developed per hour	64.8
Horsepower developed—proportion of boiler rating	64.8 per cent.
Pounds of coal used per horsepower	4.24
Coal cost per hour	$0.55
Water evaporated (actual) per pound of coal fired	6.84 pounds
Water evaporated from and at 212 degrees Fahrenheit per pound of dry coal	8.55 pounds
Water evaporated from and at 212 degrees Fahrenheit per pound of combustible	9.99 pounds
Temperature of the gases leaving the boiler, average	380° Fah.

The efficiency of the boiler is readily calculated by comparing the evaporative results, from and at 212, per pound of dry coal, obtained in practice, with the theoretical evaporative value of one pound of the dry coal, as shown by the laboratory test.

It requires 970 B.t.u. to transform one pound of water at 212 degrees Fahrenheit into steam at the same temperature, and the test shows a performance on this basis of 8.55 pounds of steam for each pound of dry coal; or that 8293 B.t.u. were absorbed in useful work out of the total heat generated by the combustion of each pound of dry coal. Now if the laboratory report states that one pound of the dry coal was equal to generating 13,000 B.t.u., then the efficiency of the boiler would be found to be

$$\frac{8293 \times 100}{13,000} = 63.8 \textit{ per cent.}$$

This, under natural-draft conditions, represents a very ordinary performance, but as high as 70 and 75 per cent. efficiency is regularly attained in the best managed plants, using forced draft to control the supply of air and using coal at about half the price stated in the foregoing.

The Generation of Power

A lecture was delivered before the Franklin Institute of Philadelphia, on the evening of November 10, by Dr. D. S. Jacobus, advisory engineer of the Babcock & Wilcox Company, and formerly of the Stevens Institute of Technology, on the above topic. In this lecture the great developments of the art within the last few years were discussed, the subject being treated from a scientific standpoint in order to compare the failings and advantages of different means of generation, and a description given with numerous lantern slides of modern power plants and of marine practice.

The type of boiler best suited for power-plant work was fully discussed, the requirements as set forth being that while the boiler should be capable of operating economically at an ordinary load it should also be capable of being driven to a capacity that is only limited by the amount of coal which can be burned in the furnace, and that it should be of a type which can be gotten under steam quickly and cut in line, either from a banked fire or from a cold state. He did not believe in adding to the water capacity to increase the thermal-storage factor to a degree which would sacrifice the ability to get up steam quickly, as an increase in the thermal-storage capacity could improve matters but little, for the conditions to be met in ordinary power-plant work compared with the ability to get a reserve boiler in on the line in the shortest possible time. The best boilers to use are those which will respond quickly, and by having a number of the boilers in the plant under steam and others in reserve the steam pressure may be maintained practically constant. To make thermal storage a material factor the steam pressure must fall, say, 25 pounds or so, and after the pressure falls the thermal storage will act against the recovery of the pressure and the rate of combustion of the fuel and the heat imparted to the boilers must be greater than with the boilers having a less thermal capacity.

The great problem in power-plant engineering is to carry economically enough reserve capacity to meet the daily peaks of the load. Then, again, there are exceptional peaks which only occur at rare intervals so that a considerable percentage of the available power may be developed only for a few hours every month, or, for that matter, for a few hours every year. It is good modern practice to drive water-tube boilers daily at double the ordinary rating during heavy-load periods. This does not mean that the boiler is driven to an overload in the sense that it is under a strained condition of running, as the rating of 10 square feet per boiler horsepower is simply a nominal one. He believed that eventually boilers would be driven harder than this in daily practice during peak-load periods, and that additional plant economy would be secured thereby. He further stated that he had demonstrated from actual tests that under proper operating conditions boilers could be driven safely and economically at much higher ratings than the present practice adopted during peak-load periods calls for.

The steam turbine is becoming more and more the standard for large power-plant work, both on account of its fuel economy and the low cost of attendance. The most economical fuel consumption under operating conditions that has so far been published was secured in a test with piston engines where a kilowatt-hour was turned out of the station for each 25,000 B.t.u. contained in the fuel. The plant referred to is the Redondo plant of the Pacific Light and Power Company, of California. The fuel was California crude oil. The load curve had two high peaks and there was a lay-over period of 4½ hours per day during which the entire plant was shut down. The 25,000 B.t.u. represent the heat of combustion of the oil used per kilowatt-hour net electrical output for the entire fifteen-day period during which the test was run. It is a fact worthy of note that this result approaches that to be expected for the plant economy of large gas engines, for the class of service considered.

While no results for plant economy as good as the above have been published for steam turbines, it is only fair to say that the figures for water consumption for turbines show that better than this economy can be secured under uniform load conditions.

Power
and The Engineer

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for November, 1910, *was* 166,000.

December 6............ 34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Editorial

Common Sense Coal Analysis

The main object of a coal analysis should be a determination of the "free-burning carbon," and not merely the fixed carbon; for it is common to find the behavior of the coal in a boiler bearing very little relation to the face value of the analysis. This is due to the fact that in some coals all the carbon is not free burning and, therefore, is not available for producing heat. This is a point which deserves careful study, for the real test of any substance is not in the laboratory but under the actual conditions in which it is to be used. A heat determination of any coal in a calorimeter may show the ultimate heat value if the coal were burned under ideal conditions, but such may bear only a remote relation to the actual conditions under which the coal is burned in the furnace.

Coal may contain carbon of several kinds, some of which is not free burning, but instead is in a semi-graphitic state or over metamorphosed, as the geologist would term it. Carbon in this form is found in Rhode Island coal. This may fittingly be termed "fixed carbon," for it requires a high degree of forcing to burn it.

Another point of discrepancy between the chemist's report and the actual performance as viewed by the fireman, is the condition of the ash. This may be of a fusible or semi-fusible kind, and as the coal tends to burn away to a clean ash, this partly melted ash covers part of the free-burning carbon, keeping the air from it and preventing combustion. This condition may be detected by the chemist if he is careful to note how the carbon burns away to ash—if it takes a long time to burn away to a clean ash the chances are that the action, just mentioned, is going on and will also take place in the furnace.

Such unreliability was recently illustrated by a coal which in the calorimeter test showed eighty-two per cent. of fixed carbon, but which under actual operating conditions did not produce as much evaporation as another coal containing only seventy-five per cent. carbon.

This discrepancy is now becoming to be recognized and unless the facts are squarely met the coal analysis is liable to become a jest among commercial men. There should be the most friendly and intimate relations between the chemist and the fireman, and the buyer for a large plant should not be misled by an analysis which, although intended to protect him, really misleads him into purchasing a coal which does not have all its fixed carbon in the form of free-burning carbon.

Leakage in Steam Engines

If the volume of a steam-engine cylinder up to the point of cutoff and including clearance be multiplied by the number of strokes per hour the product will be the volume which has to be filled per hour. The weight per cubic foot of the steam at the cutoff pressure can be found in a steam table, and the weight of the hourly volume determined. With a condensable vapor like steam, however, it will be found that the weight actually supplied to the cylinder, as determined by weighing the water supplied to the boiler or taken from the surface condenser, will be twenty or thirty per cent. more than that required to fill the cylinder up to cutoff the required number of times, if the acting fluid were all steam at that point.

The explanation is that some of the steam coming into contact with the piston and cylinder walls which have just been exposed to the relatively cool exhaust, is condensed and exists in the cylinder as water at the point of cutoff, so that there is a greater weight present than that of the dry saturated steam necessary to fill the volume.

Recent measurements of the variation of temperature of the containing surfaces seem to throw doubt upon their ability to take up enough heat at each stroke to produce the amount of condensation necessary to account for this "missing quantity," and it has been suggested that a considerable portion of it may be due to piston and valve leakage.

In a recent issue of POWER, George Mitchell, in the report of a test to determine the leakage of a piston valve, finds twenty-two per cent., enough to account for the ordinary missing quantity all at once. The test is not replete with numerical quantities, but deductions from those given show that the engine must have been using over sixty pounds of steam per horsepower-hour.

This is an abnormal condition, for piston-valve engines of even small sizes show much better economies than this. It is possible, of course, for an engine to wear into a condition where it would use this amount or more, but this test was made with a newly fitted valve.

Too general conclusions should not be drawn from a test of this kind. It is admitted that a solid plug running in a hole will leak after a time. The tendency of the wear is to enlarge the hole and to reduce the plug. And wear there will be, but it is not likely to be excessive. The valve is balanced and the pressure on the wearing surfaces is light. The fact that it is balanced and is so simple often leads to its adoption when the power required to move it and the simplicity of replacing it and reboring its seat are weighed against the heavy drag and wearing pressure of an unbalanced valve and the greater complexity of an adjustable balance. These considerations have led to its extensive adoption upon locomotives.

We do not know that the missing quantity, which includes the leakage, has been found to be notably greater upon piston-valve engines than upon other types. It would be interesting to establish the true situation in this respect. Why does not somebody with the necessary facilities make a series of tests upon an engine of this type first with the valve as tight as it can be run and then with increasing amounts of clearance, keeping the load, pressure, speed and all other conditions constant, weighing the condensed steam, indicating the engine, measuring the diagrams for steam accounted for and observing the effect of the leakage upon the missing quantity as well as upon the total steam consumption. Such a test would bring out data of real value.

Value of Power Plant Records

Agitation pertaining to the question of central versus isolated steam plants goes on apace, and, while a few engineers are awake to the full significance of what the attitude of the central-station managers means, there are scores of engineers who continue to rest secure in the belief that there is no danger of their plant being supplanted by outside interests.

There are two sides to every question and the one confronting the chief engineer of the isolated plant is, can he produce a kilowatt-hour at the switchboard cheaper than the central station can do it for him? The solicitor from the central station says he can sell current cheaper than the small plant can make it and, in the majority of cases, the isolated-plant owner cannot obtain figures to prove, even to his own satisfaction, that this is not so.

The central-station solicitor is a genial fellow, persuasive in his ways, convincing in his arguments and a producer of facts from his point of view. It is but little wonder that after a few visits from such an individual, the owner of the small plant is convinced that he has been throwing away money and that current from the central station might be had at a lower cost per annum. A contract is duly signed and the equipment of the isolated steam plant, which is in many cases taken over by the central station, is dismantled. It would be a paying investment on the money tied up in the old plant if the machinery were covered with a suitable rust preventative and held for future use, as the time may come when the discarded steam plant will be required for isolated service again. Having accomplished their ends, the central-station men are satisfied; the owner of the isolated plant sits back and rubs his hands for a few months, thinking he is saving money (the awakening comes later when he discovers that it is costing more for heat, light and power than before the change), and the engineer is looking for another job, wondering the while how it all happened.

It is hardly the thing to hit a man when he is down, but the engineer who has allowed outside interests to come in and absorb his steam plant has no one but himself to blame. If he has not been able to show in dollars and cents just what it is costing to operate his plant he cannot justly blame his employer for listening to the man who can and does produce figures.

Steam-plant records and intelligent management are the only solution of the problem for the isolated-plant engineer. His competitors are gaining additional footholds each day, and just because the engineer cannot positively state how much it costs per kilowatt-hour at the switchboard in his own plant.

It is not enough for the engineer to send a report to the office with a total of the amount of coal burned for the month, the amount of current generated and the total labor expense. Such a report means nothing and is a waste of time. A properly kept record should consist of daily, weekly and monthly reports of all that goes on in the engine and boiler rooms. Such reports not only give accurate plant data from which the actual operating cost can be calculated, but enable the engineer to check one day's or week's operation with another, and this puts him in a position to detect, at the start, any leaks in the operation of the plant.

Engineers who keep a proper system of records need not worry much about the central station absorbing their plants unless their reports show that the operating costs are excessive. An instance recently came to notice of a plant where the central-station people had repeatedly tried to get a contract signed to supply current to a large office building and displace a very neat and efficient plant. Owing to a very complete system of keeping the plant records the engineer was able to show that the plant was heating the building, furnishing current and paying for the cost of operating and upkeep at a cost of 1.36 cents per kilowatt-hour. Although the current generated by this plant was sold to the tenants at the very low price of four cents per kilowatt-hour, the plant made a profit of $704.67 for the month of October over the operating and maintenance expenses.

What can be done in one plant can be done in another. If one engineer can generate electricity cheaper than the central station can sell it, so can others. The two things necessary are to operate economically and to keep proper records so that figures can be produced when called for by those higher up.

Do Your Own Thinking

The average man peruses his daily paper with the sole object of reading as much news as possible during the allotted time; he sees a mass of statements in print, accepts them merely as such, and they, being mostly of but passing interest, are forgotten shortly after the paper is laid aside. The motto seems to be quantity rather than quality Indeed it is the exception rather than the rule to find a person carefully selecting his topics, digesting them and seriously reflecting upon them. This practice of promiscuous reading as a mere pastime has led to one of the worst habits a person can fall into —that of not thinking for himself, as a result of which he becomes only too willing to accept the statements of others rather than expend any of his own gray matter.

It would not seem strange if among the mass of current technical literature there appear some unjustifiable statements, but the man who challenges and analyzes them is really benefited, for in so doing he not only tests his own knowledge of the subject, but also opens up new lines of discussion which very often benefit others.

Again, there is another class of individuals, who, although capable of thinking for themselves, are willing to think only along certain lines; in other words, they are prejudiced. They have been trained to do a thing in a certain way and because it has served their purpose, they are willing to accept it as the best, shutting their eyes to all other methods without considering the relative merits.

To neither of these classes can the engineer afford to belong. To be successful he must be trained to think for himself; he must read for the sake of broadening his knowledge, and be always ready to consider the possibilities of anything that will increase the efficiency of his plant.

"Photography, the electric telegraph and the steam engine are the three great discoveries of the age. No five centuries in human progress can show such strides as these." Inscription on the statue to Daguerre, erected at Washington in 1890.

Inquiries of General Interest

Horsepower of Belting

What sized pulleys should be used for a 4-inch belt which is to connect two shafts running at 400 revolutions per minute and transmit 14 horsepower?

L. J. K.

A commonly used rule says: A belt 1 inch wide traveling 1000 feet per minute will transmit 1 horsepower. Expressed as a formula, it reads,

$$\text{Horsepower} = \frac{w\,v}{1000}$$

in which

w = Width of belt;
v = Velocity of travel.

Substituting

$$14 = \frac{4\,v}{1000}$$

$$v = 3666.66.$$

At 400 revolutions per minute a pulley one foot in diameter would have a rim speed of 1256 feet per minute. To have a rim speed of 3666.66 feet per minute at 400 revolutions, the diameter would be

$$\frac{3666.66}{1256} = 2.9 \textit{ feet}$$

or 2 feet $10\frac{13}{16}$ inches.

Cover for Tail Rod

I have a vertical condensing engine and wish to cover the tail rod. If I take the packing out, can I put on the gland and put the cover over it and do away with the gland along with the packing and will ⅛ inch be enough clearance between the rod and the cover?

T. R. C.

Tail-rod covers are usually made with an opening at the base permitting access to the stuffing box. If desired, one could be made steam tight, but it would add to the clearance in the cylinder. The gland should not be removed as it serves as a guide to keep the rod central; ⅛ inch clearance between the rod and cover is more than enough.

Pressure on Morrison Flue

What is the allowable pressure on a Morrison furnace flue 50 inches in diameter and ⅜ inch thick?

P. M. F.

The collapsing pressure of corrugated furnace flues may be found by multiplying the square of the thickness in thirty-seconds of an inch by 1200 and dividing the result by the product of the diameter in inches into the square root of the length in inches. As no length is given in the question it is assumed to be 72 inches. Then

$$\frac{(12)^2 \times 1200}{50 \times \sqrt{72}} = 407 \textit{ pounds}$$

With a factor of safety of 4½ the allowable pressure would be 90.44 pounds per square inch.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Weight of Cast-iron Sphere

What must be the diameter of the mold in which to cast an iron ball which will weigh 600 pounds?

N. A. P.

Cast iron weighs 0.2604 pound per cubic inch, and 600 pounds will measure

$$600 \div 0.2604 = 2304 \textit{ cubic inches.}$$

The ratio of the volume of a sphere to its diameter is expressed by the formula

$$\sqrt[3]{\frac{\textit{Volume}}{0.5236}} = \textit{Diameter}$$

which applied in this case gives 16.4 inches as the necessary diameter of the sphere. As cast iron shrinks in cooling and an allowance of ⅛ inch per foot, or 1/96 of each dimension, is made for shrinkage, the mold for a cast-iron sphere 16.4 inches in diameter will have to be

$$16.4 + \frac{16.4}{96} = 16.57 \textit{ inches}$$

in diameter.

Safe Working Pressure

Please give me the rule for figuring the safe working pressure of a steel boiler 44 inches in diameter using 6 for the factor of safety. The tensile strength of the ⅜-inch plate is 60,000 pounds per square inch.

S. W. P.

The safe working pressure of a boiler shell or drum is found by the use of the formula

$$WP = \frac{TS \times t \times \%}{R \times FS}$$

in which

WP = Working pressure;
TS = Strength of plate;
t = Thickness of plate;
$\%$ = Efficiency of longitudinal seam;
R = Radius of shell;
FS = Factor of safety.

In the question the efficiency of the joint was omitted but it is assumed to be 70 per cent. Then,

$$\frac{60{,}000 \times \frac{3}{8} \times 0.70}{22 \times 6} = 119.31 \textit{ pounds}$$

safe working pressure.

Expansions in a Compound Engine

How many expansions would be obtained in a compound engine having cylinders 10 and 20 inches in diameter and cutting off at quarter stroke in the high-pressure cylinder?

G. E. R.

Assuming no clearance (since you do not state it), the steam will expand to four times its original volume in the high-pressure cylinder. The low-pressure cylinder has twice the diameter, and therefore four times the area, of the high-; the steam will expand, therefore, to four times its final high-pressure volume when it fills the low-pressure cylinder, and the volume at the end of the low-pressure stroke will be

$$4 \times 4 = 16$$

times the volume at the instant of cutoff.

Mean Effective and Terminal Pressure

With an initial pressure in the cylinder of 100 pounds per square inch and a cutoff at ¼ stroke, what would be the mean effective and terminal pressures in the cylinder, allowing nothing for clearance or back pressure?

C. W. F.

The mean effective pressure of expanding steam is the mean pressure minus the back pressure. Neglecting clearance condensation and back pressure above that of the atmosphere the mean pressure of steam at 100 pounds gage pressure expanded four times will be

$$114.7 \times 0.582 = 66.7 \textit{ pounds absolute.}$$

Subtracting the atmospheric pressure the mean effective pressure on the piston will be

$$66.7 - 14.7 = 52 \textit{ pounds.}$$

Expansions in Single Cylinder

In good practice, how many expansions are allowable in a single-cylinder Corliss engine?

E. S. C.

The range of economical use of steam lies between 3 and 5 expansions for ordinary pressures.

Slide Valve Proportions

With steam ports one-half inch wide, what would be the dimensions and travel of a plain slide valve which would cut off at ½ stroke?

S. V. P.

To give full port opening the valve travel would be 3 inches and the lap 1 inch.

New Power House Equipment

Willard Horsepower Slide Rule

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

This rule was devised to obtain from two settings the horsepower developed by any type of double-acting steam engine. It is a time saver and eliminates errors in figuring indicator diagrams. Cylinder ratios on compound engines can be found with but one setting.

Scale *A* shows the indicated horsepower developed from 30 to 3000; scale *B* shows cylinder diameters in inches from 10 to 100; scale *C* shows the stroke of engines from 10 feet to 6 inches, part of the divisions being marked in inches and part in feet; scale *D* shows the number of revolutions per minute from 10 to 1000; the same scale also shows the piston speed in feet per minute up to 1000, and scale *E* shows the mean effective pressures from 2 to 150 pounds. The rule is used as follows:

To find the horsepower when the mean effective pressure, revolutions per minute, stroke and cylinder diameter are given—first, set the stroke on revolutions per minute; second, move the two slides together and set the arrow near the center of slide *D* on the mean effective pressure scale and then over the given cylinder diameter, find the indicated horsepower.

To find the cylinder dimensions when the mean effective pressure and indicated horsepower are given—first, set the arrow at the extreme right of scale *C* near the word Stroke on the maximum allowable piston speed for the type of engine, usually not over 800 feet per minute; second, move the two slides together until the arrow near the center of slide *D* coincides with the given mean effective pressure, and under the given indicated horsepower read cylinder diameter. The coinciding divisions on scales *C* and *D* give the stroke and revolutions per minute that can be used, not to exceed the piston speed selected.

It may be seen that there are many combinations and the best suited for the conditions can be selected at a glance. Take the following example: With a mean effective pressure of 25 pounds; revolutions per minute, 80; stroke, 5 feet; cylinder diameter, 40 inches, find the indicated horsepower.

First, set the point indicating 5 feet on scale *C* at the point indicating 80 revolutions per minute on scale *D*, which is equivalent to 800 feet piston speed per minute, as shown by the arrow at the extreme right of scale *C;* second, move the two slides together until the arrow on slide *D*, near the center, coincides with 25 pounds mean effective pressure, and over 40 inches cylinder diameter on scale *B*, read 763 indicated horsepower on scale *A*.

Assuming a mean effective pressure of 140 pounds; revolutions per minute, 100; stroke, 4 feet; cylinder diameter, 36 inches; find the indicated horsepower.

If the arrow near the center of slide *D* is used, as should be done at first in all cases, it will be found that the indicated horsepower scale does not extend far enough to enable the result to be read. In such a case use the arrow at the right end of the slide and read the result in indicated horsepower toward the left of scale *A*, adding one cipher. Thus, using the values given above and proceeding as in the first example, over the 36 inches cylinder diameter read 346 indicated horsepower. Adding a cipher to this gives 3460 indicated horsepower. This case is very unusual as it is very seldom that as high a mean effective pressure as 140 pounds is ever used. It shows, however, the large range of the rule.

Again, assume a given mean effective pressure of 55 pounds, and an indicated horsepower of 250, find the cylinder diameter, stroke and revolutions per minute.

First, assume a maximum piston speed of 800 feet per minute, setting the arrow on slide *C* at 800. This setting gives all the combinations of revolutions per minute and stroke that can be used, indicated by coinciding divisions; second, move the two slides together and set the arrow near the center of slide *D* on the mean effective pressure 55, and under 250 indicated horsepower on scale *A* the cylinder diameter will be indicated between 15 and 16 inches. Sixteen inches would be the nearest standard even diameter, so set 16 under 250, moving the two slides together. It will now be seen that the mean effective pressure has been reduced to about 51 pounds. If it is still desired to keep the mean effective pressure of 55 pounds as given, set the arrow at this value by moving slide *D*. The piston speed will now be found to be about

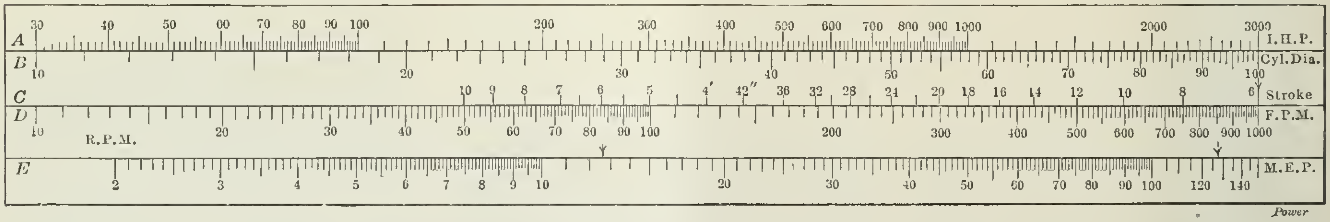


Fig. 1. The New Willard Horsepower Slide Rule

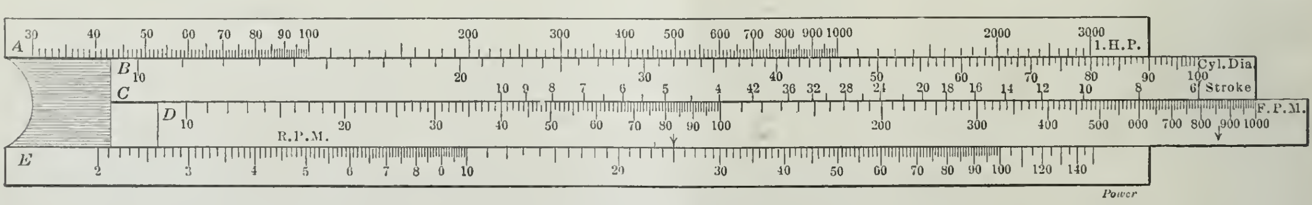


Fig. 2. Finding the Indicated Horsepower

745 feet per minute. This corresponds to several combinations of revolutions per minute and strokes as follows: 16x48-inch cylinder at 93 revolutions per minute; 16x42-inch cylinder at 108 revolutions per minute; 16x36-inch cylinder at 123 revolutions per minute; 16x30-inch cylinder at 148 revolutions per minute.

The usual practice in figuring compound engines is to base the calculations on the low-pressure cylinder, determining the high-pressure cylinder by a certain ratio of volumes. Thus, for compound engines to determine the cylinder sizes, etc., proceed as for a simple engine, using the proper mean effective pressure reduced to the low-pressure cylinder. After the low-pressure cylinder diameter is found and the ratio of the volumes of the two cylinders is known, to find the high-pressure cylinder diameter proceed as follows: On scale *A* beginning at 100, each division from there on can be considered as a scale of ratios; thus 200 would be a ratio of 100:200 or 1:2 and 400 a ratio of 100:400 or 1:4.

Suppose a case where the low-pressure cylinder has been found to be 40 inches in diameter and the cylinder ratio is 1:4. Set the 40-inch cylinder diameter under 400, and under 100 read 20-inch cylinder. If a 1 : 3 ratio is wanted, set 40 under 300 and under 100 read 23 plus. If 24- and 40-inch cylinders are used, the ratio would be 100.278 or 1:2.78. This is found by setting 24 under 100 and reading 278 over the 40-inch cylinder diameter.

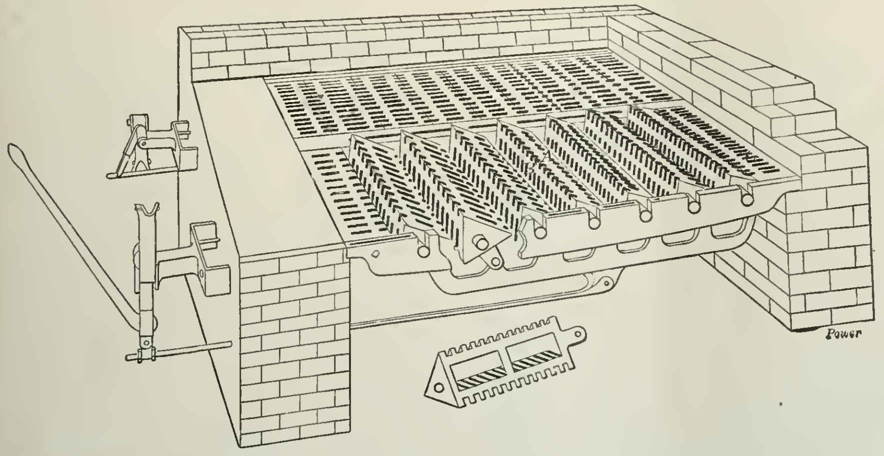
When figuring the horsepower from indicator diagrams, it should be remembered that the rule is proportioned to give the horsepower for a double-acting steam engine, and that if the horsepower from one end of the cylinder only is desired, the result should be divided by 2, the same as when using the rule for a single-acting engine.

Accompanying the slide rule is a table giving the mean effective pressures ordinarily used for different steam pressures and different vacuums for both single and compound engines, when operating condensing and noncondensing. With this information before the user, it is a comparatively simple matter to determine the proper size of cylinder required for any condition or to determine the results of a test from a large number of indicator diagrams in a very short time.

The rule is made with a patented adjustment to take care of wear and effects caused by changes in the climate, which makes it a very desirable and efficient outfit for anyone who has any occasion to do any amount of estimating or steam-engine testing. It is the invention of L. L. Willard, 71 Broadway, New York City.

The Long Duplex Shaking Grate

This shaking grate consists of one, two or more rows of grate bars, each having two usable sides *A* and *B* cast in one piece in the form of an inverted V or triangle.



Showing Design and Application of the Duplex Shaking Grate

Only one of these sides can be used at a time, for when the side *A* is in use, the side *B* is on the underside cooling off. It is claimed that this feature adds strength to the bars and prevents them from warping.

When the grate is being shaken, the side *A* of one grate bar moves opposite to the side *B* of the grate bar next to it, exerting a tremendous crushing effect on the clinker and ashes, and at the same time the clean side of the grate bar that is coming into use rakes the unconsumed fuel off of the top of the ashes, leaving the clinker to drop through into the ashpit. The illustration shows how the grate operates.

The shaking mechanism locks in both the forward and backward position, and the grates are thus held level. Suitable means are provided for any adjustment necessary to level the bars due to expansion, contraction, etc.

The grates can be placed under any boiler without altering the firebox. The only cutting necessary is a 1½-inch hole through the boiler front for the stoker rod. The grate is made by the Long Grate Bar Company, 611 White building, Buffalo, N. Y.

New Bayer Soot Blower for Water Tube Boilers

The accompanying illustrations show the new Bayer soot-blower system designed to be applied to water-tube boilers of the Babcock & Wilcox type.

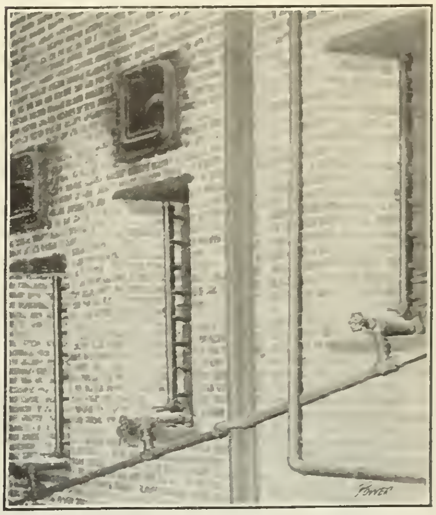


Fig. 1. Blower Installed on B. & W. Boiler

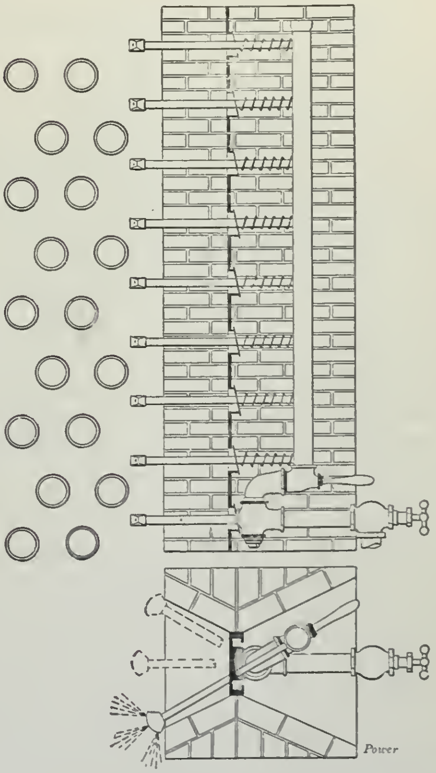


Fig. 2. Sectional and Plan Views of Blower Installation

The upright headers, shown in Fig. 1, are mounted on a special brass swing joint and carry jets with three-way nozzles. These nozzles enter through the cleaning ports and blow across and between the tubes, sweeping the entire surface. Adjustable cover plates, the construction of which is apparent in Fig. 2, are fitted to the flue-blowing apparatus so that no cold air is admitted to the interior of the boiler setting. The removal of soot is accomplished by opening the valve wide and moving backward and forward the handle attached to the header mounted on the swing joint.

One of the features of this system, which is manufactured by the Bayer Steam Soot Blower Company, St. Louis, Mo., is that it can be applied to water-tube boilers without altering the setting or without doing masonry work of any kind. It can also be removed in a few minutes if desired.

Monarch Water Tube Soot Cleaner

This is a new soot blower designed for cleaning the tubes of water-tube boilers. The cleaner consists of a series of jets placed in the side walls, which blow all soot deposit in the direction of the draft. It is not exposed to the heat and is practically indestructible. It can be applied to any type of water-tube boiler and does not require any alterations in the setting. Owing to its simplicity there is nothing about the apparatus to get out of order or to cause trouble.

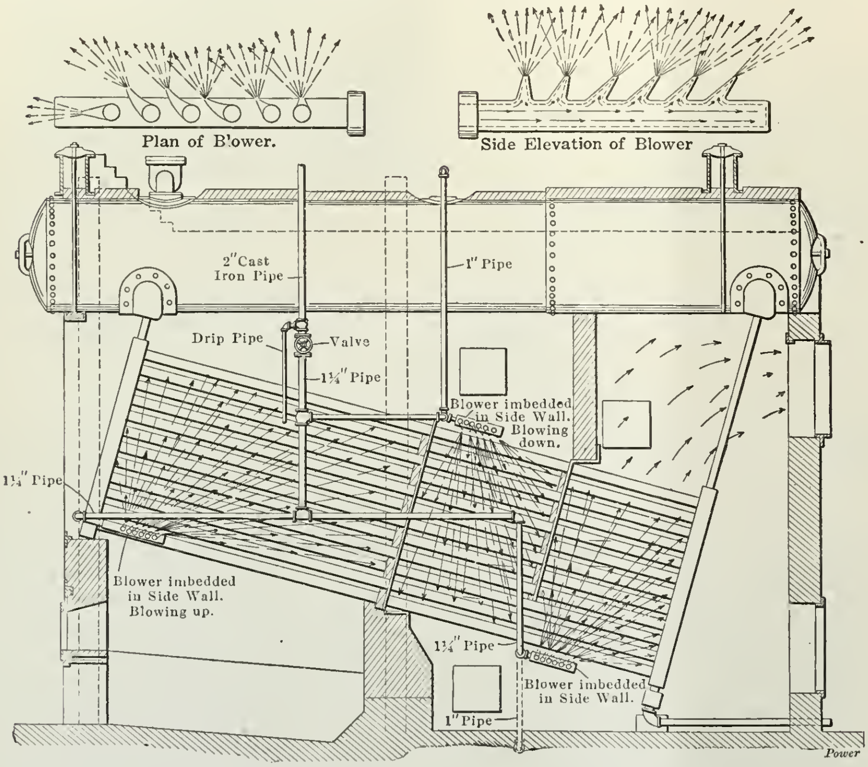

Plan of Blower.
Side Elevation of Blower
2" Cast Iron Pipe
1" Pipe
Drip Pipe
Valve
1¼" Pipe
Blower imbedded in Side Wall. Blowing down.
1¼" Pipe
Blower imbedded in Side Wall. Blowing up.
1¼" Pipe
Blower imbedded in Side Wall.
1" Pipe
Power

Fig. 1. Side Elevation of Tubes

Two views of this device are shown in Figs. 1 and 2; Fig. 1 shows the soot cleaner applied to a water-tube boiler and Fig. 2 shows a top view which indicates the direction in which the steam jet is blowing. With this device in place it is simply a matter of opening a steam valve in order to clean the tubes of the accumulation of soot. This permits operating the boiler without interruption when cleaning the tubes. The device is manufactured by the Monarch Steam Blower Company, 187 River street, Troy, N. Y.

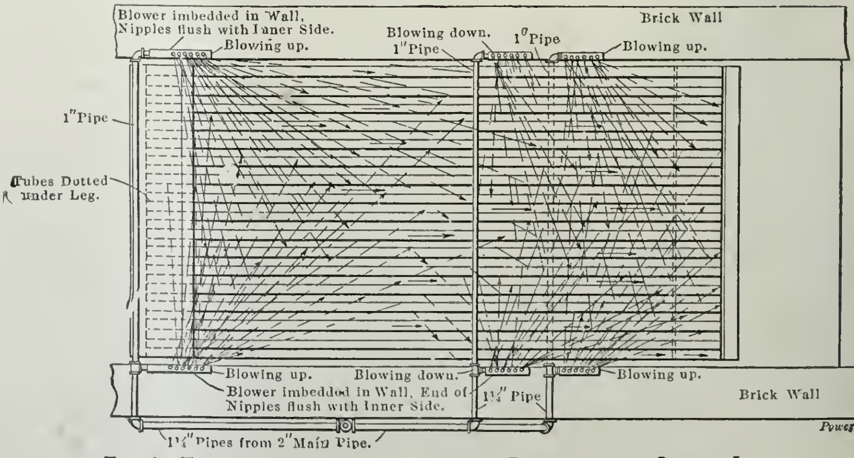


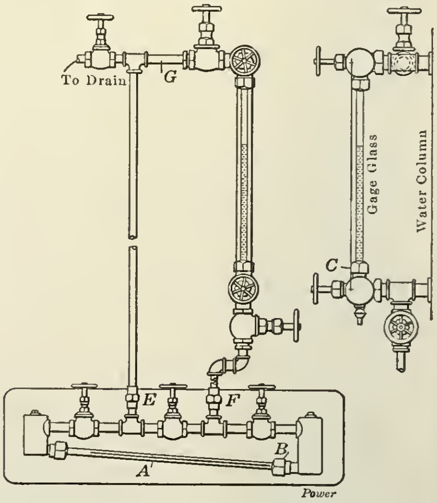
Fig. 2. Top View of Boiler, Showing Direction of Steam Jets

The T. & W. Water Gage

This gage is applicable to any type of boiler, its principal advantage being that it may be located at any point either in the boiler or engine room. Its special application is in connection with vertical boilers, as the gage can be placed low down on the boiler front immediately before the eye of the fireman.

The gage indicates the hight of water in the boiler by mercury in the glass A, as shown in the drawing, the distance from the gland B to the end of the mercury column always being equal to the distance from the gland C to the water level in the gage glass. This is accomplished by the mercury balancing the difference between the constant head of water in E, and the variable head F due to the hight of the water in the boiler.

The pipe E is always kept full by the condensation in the pipe G, any excess running back into the water column. There are no moving parts whatever inside the gage. It operates entirely by a difference of hydrostatic pressure. In operation the gage is always cold, there being no variation of temperature to break the glass.



T. & W. Water Gage

After the device has been properly connected up and charged with mercury it needs no further attention, and the water column can be blown out and the boiler put in and out of service without manipulating any of the valve connections.

This water gage is manufactured by McMeans & Tripp, 607 State Life building, Indianapolis, Ind.

Dennis Flanagan, th' Frenchman thet runs th' light plant, hez a wooden leg. He lost th' original when he wuz servin' in' th' Salvation Army. Sez thet it kums mighty handy when his ingin stops on th' center.

Ironmongers to Hold International Exposition at Budapest

Under royal protection and with the assistance of the Board of Trade of Hungary, the National Association of Hungarian Ironmongers will hold an international exposition at the Industrial palace at Budapest, in May and June of next year. Articles intended for exhibit will be transported on the Hungarian state railways at one-half the ordinary freight rate, and the committee on exhibition will take care of the insurance and the operation and representation of exhibits sent from abroad without attendance. Every exhibitor will be presented with a commemorative medal and an exhibitor's diploma, and the Hungarian government has placed diplomas, as well as gold and silver medals at the disposal of the committee. The national Hungarian trade societies and corporations will also offer awards and medals for exhibits of particular merit. Joseph Baneth, of Budapest, is in this country in the interests of the exposition and will be located at the Herald Square hotel in New York until the middle of December.

Notes on Causes of Boiler Failure

In the November 1 issue we published an article under the title, "Notes on the Causes of Boiler Failure" in which a slight typographical error was permitted to escape correction. At the top of the middle column on page 1936 the word "tension" was used in place of "compression." From the sense of the preceding text the error is perfectly obvious.

The sentence containing the error should have read, "Since in a plate that is bent, there exists a tension on one side *a*, Fig. 2, and a compression on the other side *d*, and in addition there is an initial stress of 12,000 pounds to the square inch, the resulting total stress in the joint is

$$18{,}000 + 12{,}000 = 30{,}000$$

pounds tension at *a* and

$$18{,}000 - 12{,}000 = 6000$$

compression at *d*."

The John Fritz Medal Award for 1910

On November 30 at the house of the American Society of Civil Engineers, Alfred Noble, a past president, was awarded the John Fritz medal, which was established in 1902 in honor of Mr. Fritz on his eightieth birthday. Dr. Samuel Sheldon presided and addresses were made by Isham Randolph and Dr. R. W. Raymond.

The awards of the John Fritz medal have been as follows: In 1905 to Lord Kelvin, "for his work in cable telegraphy and other scientific attainments"; in 1906 to George Westinghouse, "for the invention and development of the air brake"; in 1907 to Alexander Graham Bell, "for the invention and introduction of the telephone"; in 1908 to Thomas Alva Edison, "for the invention of the duplex and quadruplex telegraph, the phonograph, the development of a commercially practical incandescent lamp and the development of a complete system of electric lighting, including dynamos, regulating devices, underground-system protective devices and meters"; in 1909 to Charles T. Porter, "for his work in advancing the knowledge of steam engineering and in improvements in engine construction."

Gas Engine Trades Meet at Racine

At the meeting of the National Gas and Gasolene Engine Trades' Association which will be held at Racine, Wis., December 12 to 15, the following program has been announced:

"The Installation of a Mechanical Ignition System Complete," by H. F. Apple, Dayton, O.; "Does the Efficiency of a Gas Engine Depend on the Equipment?" Charles F. Kratsch, Chicago; "Salesmanship vs. Manhood," J. E. Hozier, Chicago; "Practical Electricity on the Farm," C. H. Roth, Chicago; "Gas," H. I. Lea, Chicago; "The Gasolene Engine in the Agricultural Field at Home and Abroad," George Cormack, Jr., Plano, Ill.; "Gas Producers," H. F. Smith, Lexington, O.; "Gas Engine Ignition," Otto Heins, New York City; "Gasolene Engines on the Farm," J. C. Finkbeiner, Jackson, Mich.; "Guaranteeing Gas Engine Power," C. O. Hamilton, Elyria, O.; "Comparisons of Various Methods of Testing Engines," James Tracey, New York City.

Other papers are promised by Messrs. J. C. Miller and Ferdinand Lammert, of Chicago, and others. Every effort is being made to make this a successful and instructive meeting and all who are in any way interested are cordially invited to attend. The ladies are especially invited.

PERSONAL

John A. Dent, recently with the oil pipe-line service of the New York Transit Company, has been appointed instructor in mechanical engineering at the University of Illinois.

H. B. Dirks, instructor in mechanical engineering at the University of Illinois, has resigned to accept a position as assistant to the general manager of the National Machinery Company, Chicago.

John I. Rogers has opened a New York office in the City Investing building at 165 Broadway and will now use it as his main office. He is making a specialty of machine shops and power plants and of general iron and steel-works engineering. Mr. Rogers resigned from the Midvale Steel Company, of Philadelphia, about one year ago to take up professional practice and since that time has been engaged in consultation work and design along the above lines.

Charles D. Thurber, engineer of the Pepperell Manufacturing Company, Biddeford, Me., and an occasional contributor to Power, has just celebrated his golden wedding. Although 72 years of age, he is still in active charge of this large plant, embracing steam-turbine and electric transmission as well as reciprocating engines and water wheels. His early engineering experience was gained at the Corliss works, and he is full of interesting reminscences of the younger days and the older people of American steam engineering.

SOCIETY NOTES

The Boston members of the American Society of Mechanical Engineers will hold their monthly meeting in the auditorium of the Edison Electric Illuminating Company of Boston, 39 Boylston street, on Friday evening, December 16, at 8 o'clock. Samuel B. Fowler, a non-member, will read a paper on the subject of the "Mechanical Handling of Freight."

NEW PUBLICATIONS

"North Dakota Lignite as a Fuel for Power Plant Boilers" is the title of Bulletin No. 2 just issued by the Bureau of Mines. This bulletin describes a series of tests at the pumping plant of the United States Reclamation Service at Williston, N. D. The Reclamation Service has a large project there and had installed steam boilers with furnaces designed to burn a brown lignite that was mined on adjacent Government land.

The furnace is of the semi-gas-producer type and has an external resemblance to the so called dutch oven. The results of the tests on the lignite show that this fuel, though generally considered unsatisfactory, may be used with fair economy under boilers that generate their full rated capacity.

The tests were conducted by the Technologic branch of the Geological Survey, which is now a part of the Bureau of Mines. The authors of the bulletin are D. T. Randall and Henry Kreisinger. The bulletin will be of interest to fuel engineers, especially to those located in lignite territory. It may be obtained by addressing the director of the Bureau of Mines, Washington, D. C.

Locomotive Explodes at Altoona

On Saturday afternoon, November 19, a locomotive assisting a westbound train through Altoona, Penn., as a pusher exploded near Thirtieth street in that city. As the boiler shot off obliquely across the tracks it was struck by an incoming eastbound train, the two locomotives of which were toppled over, killing an engineer and two firemen and seriously injuring another engineer, a fireman and a brakeman. The crown sheet came down and the explosion was very evidently due to the always disastrous attempt to make steam without water.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

ROTARY ENGINE. Andrew Renik, Kenosha, Wis. 975,637.

ROTARY ENGINE. George B. Seitz, Hammond, Ind. 975,643.

THERMODYNAMIC MOTOR. Philip K. Stern, New York, N. Y. 975,651.

HYDRAULIC POWER-TRANSMITTING APPARATUS. Carlton R. Radcliffe, New York, N. Y., assignor, by mesne assignments, to the Scientific Research Company, New York, N. Y., a Corporation of New York. 975,795.

INTERNAL COMBUSTION ENGINE. Carlton R. Radcliffe, New York, N. Y., assignor to the Scientific Research Company, New York, N. Y., a Corporation of New York. 975,796.

INTERNAL COMBUSTION ENGINE. Thomas Vritch, Dumont, N. J., assignor to the Scientific Research Company, New York, N. Y., a Corporation of New York. 975,809.

AIR MOTOR OR WINDMILL. Otte E. Sill, Los Angeles, Cal. 976,003.

GAS ENGINE. Jesse B. Brown, Belding, Mich. 976,034.

TURBINE. Calvin W. Wellman South Boardman, Mich., assignor to Hattie M. Wellman, Oshkosh, Wis. 976,109.

BOILERS, FURNACES AND GAS PRODUCERS

SMOKE CONSUMER. John Rose Blaik, Montreal, Quebec, Canada. 975,391.

BURNER. James Edward Fliteroft, Ocean Grove, N. J., assignor to Otto Bernz, Newark, N. J. 975,518.

SMOKE CONSUMER. John S. Green, Philadelphia, Penn. 975,688.

STEAM BOILER. Willard S. Tuttle, Philadelphia, Penn. 975,736.

FURNACE HEAD FOR GAS FURNACES WITH CHANGING DIRECTION OF FLAMES. Bruno Versen, Dortmund, Germany. 975,738.

CRUDE-OIL BURNER. George C. Yale, Boynton, Okla. 975,740.

FURNACE STRUCTURE. James Eggins, Ottawa, Ontario, Canada. 975,754.

BOILER FURNACE. Gustav de Grahl, Zehlendorf, near Berlin, Germany. 975,763.

BOILER. William F. Sellers, Wilmington, Del., and Charles J. Davidson, Milwaukee, Wis., assignors to Edge Moor Iron Company, Edgemoor, Del., a Corporation of Delaware. 975,897.

UNDERFEED STOKER. Paul L. Crowe, Jersey City, N. J. 976,049.

APPARATUS FOR GENERATING MOTIVE FLUID. George B. Hayes, Denver, Colo. 976,077.

HOLLOW WATER-HEATING GRATE. Ludwig Kaufman, New York, N. Y., assignor of one-half to James Murray, New York, N. Y. 976,080.

BOILER. William F. Sellers, Wilmington, Del., and Charles J. Davidson, Milwaukee, Wis., assignors to Edge Moor Iron Company, Edgemoor, Del., a Corporation of Delaware. 976,103.

OIL BURNER. William A. Stephens, Lawton, Okla., assignor of one-half to James M. Powers, Lawton, Okla. 976,105.

POWER-PLANT AUXILIARIES AND APPLIANCES

AIR PUMP. Gregory J. Spohrer, Franklin, Penn. 975,473.

OIL BURNER. Edwin W. Tucker, San Francisco, Cal. 975,482.

VALVE. William L. Hamilton, Bangor, Mich. 975,524.

PROPELLER TURBINE PUMP. Ernest K. Hood, Indianapolis, Ind. 975,526.

MOTOR SUCTION PUMP. William H. Keller, Philadelphia, Penn., assignor to Keller Manufacturing Company, Philadelphia, Penn., a Corporation of Delaware. 975,532.

CONDENSER. David G. Galbraith, Mineral Wells, Tex., assignor to Nannie L. Galbraith, Mineral Wells, Tex. 975,612.

CARBURETER. Walter H. Potthast, Manning, Iowa. 975,635.

CARBURETER. John H. Koontz, Culver, Ind. 975,696.

REDUCING VALVE. John H. Derby, New York, N. Y., assignor to General Fire Extinguisher Company, New York, N. Y., a Corporation of New York. 975,838.

VALVE REGULATOR. Thomas W. Wilkins, East Randolph, N. Y., assignor to Ziermore Regulator Company, Johnsonburg, Penn., a Corporation of Pennsylvania. 975,915.

CENTRIFUGAL PUMP, CONDENSER AND COMPRESSOR. Edmund Scott Gustave Rees, Wolverhampton, England. 975,997.

PISTON PACKING. Hugh S. Studdert, Seattle, Wash. 976,007.

VALVE. John C. Thompson, Belmont, Cal. 976,010.

GAGE GLASS. Fred R. Bispham, Bemidji, Minn. 976,030.

VALVE FOR STEAM TRAPS AND THE LIKE. Arthur E. Duram, Chicago, Ill. 976,055.

VALVE. Elias L. Grooms, Gulfport, Miss. 976,070.

LUBRICATOR. Frederic Charles von Haxthausen, St. Petersburg, Russia, assignor to Flotteur Gesellschaft mit beschraenkter Haftung, Berlin, Germany.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC FURNACE WITH MAGNETICALLY ROTATED CHARGE. Henry Noel Potter, New Rochelle, N. Y., assignor to Geo. Westinghouse, Pittsburg, Penn. 975,571.

ELECTRIC CABLE JOINT. Wilbur O. Wilson, Chicago, Ill. 975,592.

ARC LAMP. Tito Livio Carbone, Charlottenburg, Germany. 975,935.

ELECTRODE FOR REVERSIBLE GALVANIC BATTERIES. William Morrison, Des Moines, Iowa. 975,981.

ALTERNATING-CURRENT CONTROLLING APPARATUS FOR HYDRAULIC ELEVATORS. David L. Lindquist, Yonkers, N. Y., assignor to Otis Elevator Company, Jersey City, N. J., a Corporation of New Jersey. 976,130.

MAGNETO ELECTRIC GENERATOR. Gottlob Honold, Stuttgart, Germany. 974,967.

ELECTRIC COUPLING AND SWITCH. Reinhold H. Wappler, New York, N. Y., assignor to American Cystoscope Makers, Inc., New York, N. Y., a Corporation of New York. 975,090.

ELECTRIC BAKING OVEN. James I. Ayer, Cambridge, and Horace B. Gale, Natick, Mass., assignors to Simplex Electric Heating Company, Cambridge, Mass., a Corporation of Massachusetts. 975,107.

ELECTRICAL CONNECTOR. James J. Burns, Worcester, Mass., assignor to W. H. Leland & Co., Worcester, Mass., a Corporation. 975,118.

NERNST LAMP BODY. Otto Foell and Max Harris, Pittsburg, Penn., assignors, by mesne assignments, to Nernst Lamp Company, Pittsburg, Penn., a Corporation of Pennsylvania. 975,126.

CIRCUIT BREAKER. Ford W. Harris, Wilkinsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 975,421.

POWER PLANT TOOLS

WINCH. Frederick Metcalf and John E. Carson, Cleveland, Ohio, assignors to the Chase Machine Company, Cleveland, Ohio, a Corporation of Ohio. 975,559.

WRENCH. John H. Ferguson, Dayton, Ohio. 975,609.

WRENCH. Walter F. Bailey, Los Angeles, Cal., assignor of one-half to Mitchell S. Mickelson and Andrew P. Jensen, Los Angeles, Cal. 975,671.

Engineering Societies

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power
and The Engineer

Vol. 32 NEW YORK, DECEMBER 13, 1910 No. 50

IN GIVING his reasons for changing his vocation, an engineer with long years of experience says, "I have been connected with this business for half a life time and am retiring discouraged and disgusted.

"Conditions are getting worse all the time and the practical man who depends on his knowledge of the business and his merits to hold his job without a 'pull' is going down and out. He must make way for the paper and pencil engineer with big ideas, who shows on paper how to run boilers and engines without burning coal. It is too much for me and I am done with it forever."

In a heated political campaign some years ago a strong, earnest man, expressing his contempt of the half-hearted efforts and his disappointment in the tactics of a weaker brother intrusted with an important mission, said, "God Almighty hates a quitter."

There is honor in honorable defeat, but a failure to conquer the unconquerable should be only a spur to the man who takes it rightly. It should stir his spirit and urge him on to redoubled effort in the next battle.

It is a safe bet that if one has never made a failure he has never scored a success worthy the name.

What does a real knockdown mean? It means a chance to rest while taking the count, a chance to spit on the hands and come up with the determination to do better next time.

The chief factor in success is patient, dogged industry that neither flinches nor falters—the unconquerable courage that will never submit or yield. It is success itself.

There are many successful engineers in the world, with perhaps no two alike in physical endowment and mental attainment, but all possess this one moral quality—a grim tenacity of purpose to hang on and hold out through everything until the desired end is reached.

"If you have never met more than your match, it is likely you have seldom toed the scratch."

Modern Manufacturing Power Plant

By Warren O. Rogers

> *In this turbine plant the arrangement of the machinery is ideal. Nothing is cramped, and the plant is light and well ventilated. The method of handling the coal and ashes is convenient and up-todate. A turbine-driven centrifugal pump is used as a boiler feeder.*

If an engineer were to step into a power plant and find in one large, light, well ventilated room steam turbines and generators, air compressors, hydraulic pumps, apparatus for a hot-water heating system, exciter units and refrigerating machines, all delivering their output to neighboring manufacturing buildings, he would doubtless exclaim, "a combination central manufacturing power plant." The following deals with such a plant:

The location is ideal, being close to the main line of a certain railroad; it is also but a short distance from a navigable river. The building is erected on a plot of land practically free from other buildings; the engine room is well lighted and the ventilation is all that can be desired, due to the large adjustable windows in the side walls of the engine room. A concrete roof supports two monitors which have adjustable windows extending their entire length. All of these windows are adjusted from the floor by means of handwheels and extension rods. The room is so spacious that the heat from the various machines is hardly noticeable and within it is comfortable even on a hot summer's day. Fig. 1 presents an excellent view of the generator room, and shows the general arrangement of the machinery.

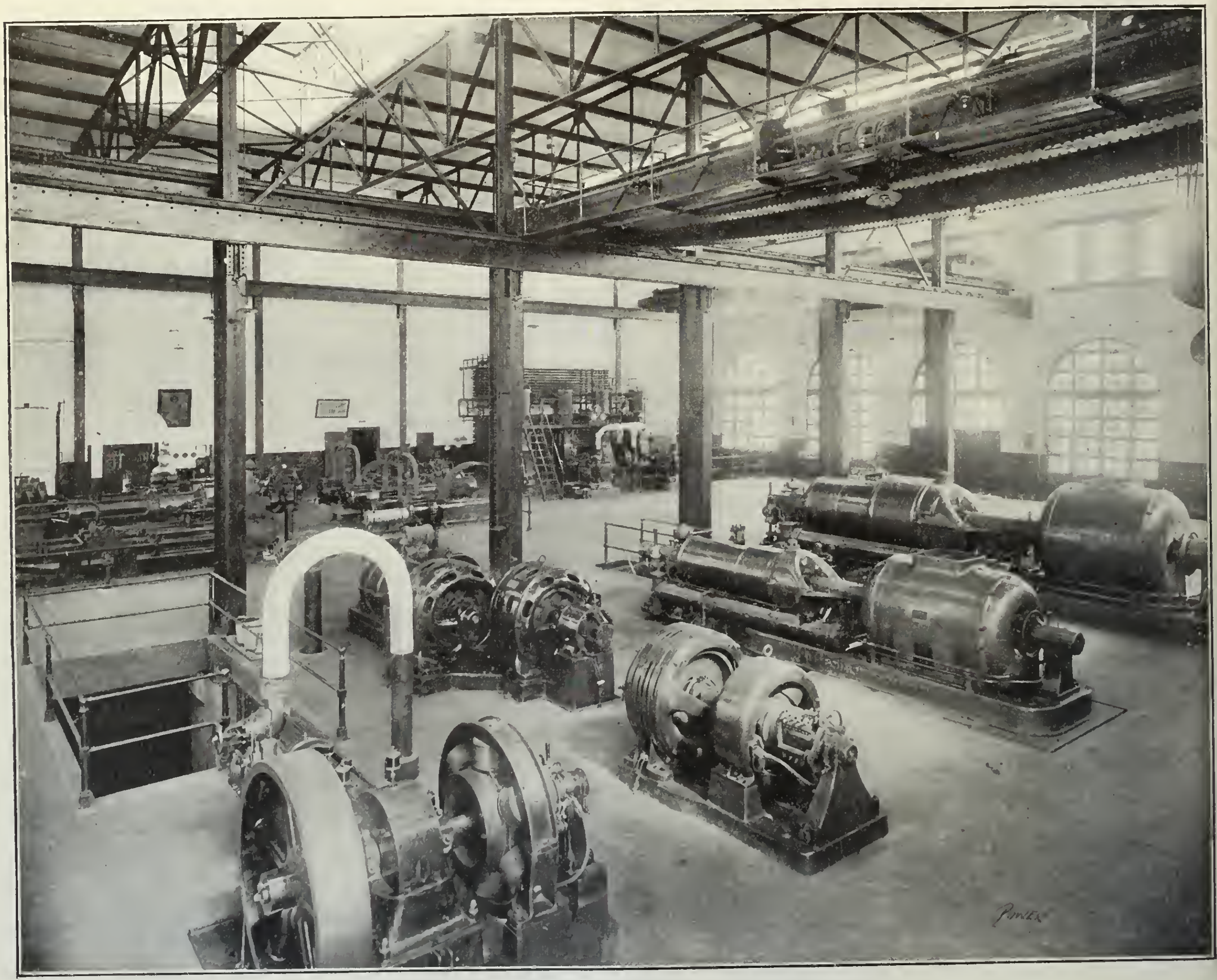


Fig. 1. Main Portion of Turbine Room, Showing General Layout of Plant

Coal Delivery

Fuel is delivered to the yard in coal cars and is dumped either into the conveyer tipple or in a pile under a trestle to serve as a reserve supply. Fig. 2 shows the coal and ash tipple. A car of coal is on the scale ready to be discharged into the coal tipple, which is constructed similarly to the ash tipple located above.

When coal is desired in the coal bunkers, which are located above the boilers, the link-belt conveyer, which is 596 feet long and contains 298 buckets, is started by throwing in a switch located in the boiler room. The crusher rolls, which are set below the coal chute shown in front of the concrete coal tipple, are also started at the same time. As the coal chute leading to the coal-crushing rolls is constructed with a movable bottom, having a forward and back or shaking motion, the coal is automatically fed to the crushing rolls.

After the coal is crushed it is elevated to the top of the bridge and carried to the concrete coal-storage bin, which extends the entire length of the boiler room. This storage bin is divided into six compartments, each capable of holding 250 tons of coal. Two delivery outlets lead from each of these compartments, the delivery openings being in the center of each boiler and directly over the dutch-oven furnaces. When a stoker requires more fuel a Standard coal scale, which runs on a suspended track, is pushed under the coal-supply outlet and the coal is weighed a second time, and then discharged to the stokers. Each row of boilers has a separate coal-supply chute and scale.

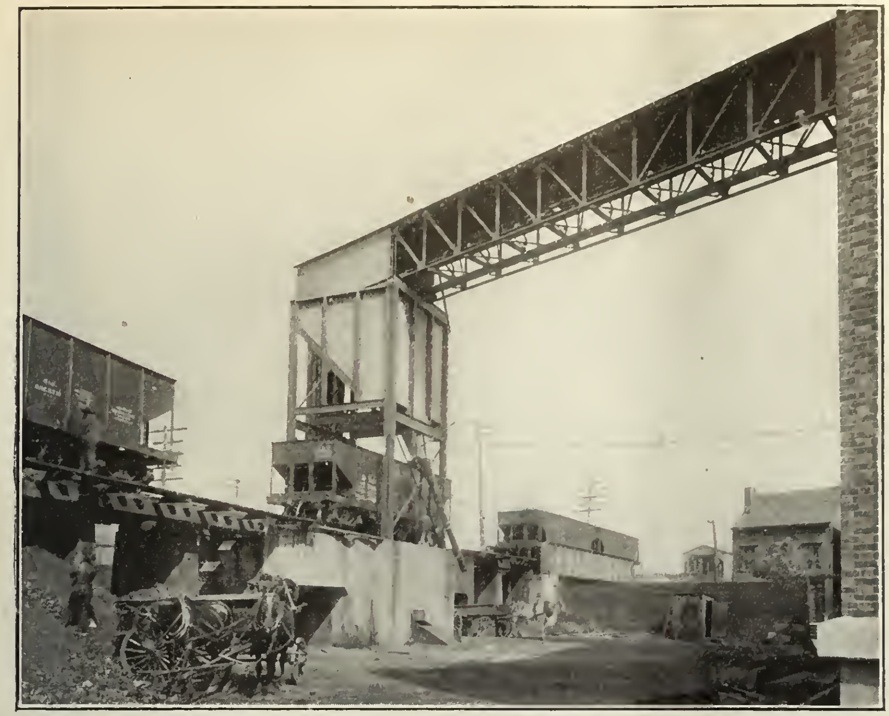


Fig. 2. Coal and Ash Tipple, Scales and Conveyer

Boiler Room

There are six 500-horsepower Babcock & Wilcox boilers, carrying a steam pressure of 160 pounds per square inch. They are set in three batteries of two boilers each, and each boiler is equipped with two Murphy stokers. The stokers of each boiler are driven by a small steam engine, as shown in Fig. 3, which also presents a view of one side of the firing alley. The boilers are of the three-drum type and the brick setting of each battery is covered by a metal casing. This prevents air leakage through the setting, and a higher efficiency is obtained from the furnaces than would be possible with the ordinary setting.

The massive supporting beams of the overhead coal bin are shown in Fig. 3, also the arrangement of the boilers and weighing machine. The flooring is made of concrete and as the ashes are handled below the boiler-room floor the place is easily kept clean.

At the rear of each battery of boilers is placed a Green economizer. Fig. 4 shows the arrangement of the two economizers that are set behind the boiler shown in Fig. 3. The tube scrapers on both economizers are operated by one motor by means of a belt drive. A bypass flue is built beneath the superheaters so that either one or both can be cut out of the circuit of flue gases passing to the stack. This is accomplished by means of dampers, as shown. The gases from one or both batteries of boilers can be passed through one or both of the economizers, and so far the two economizers have been utilized.

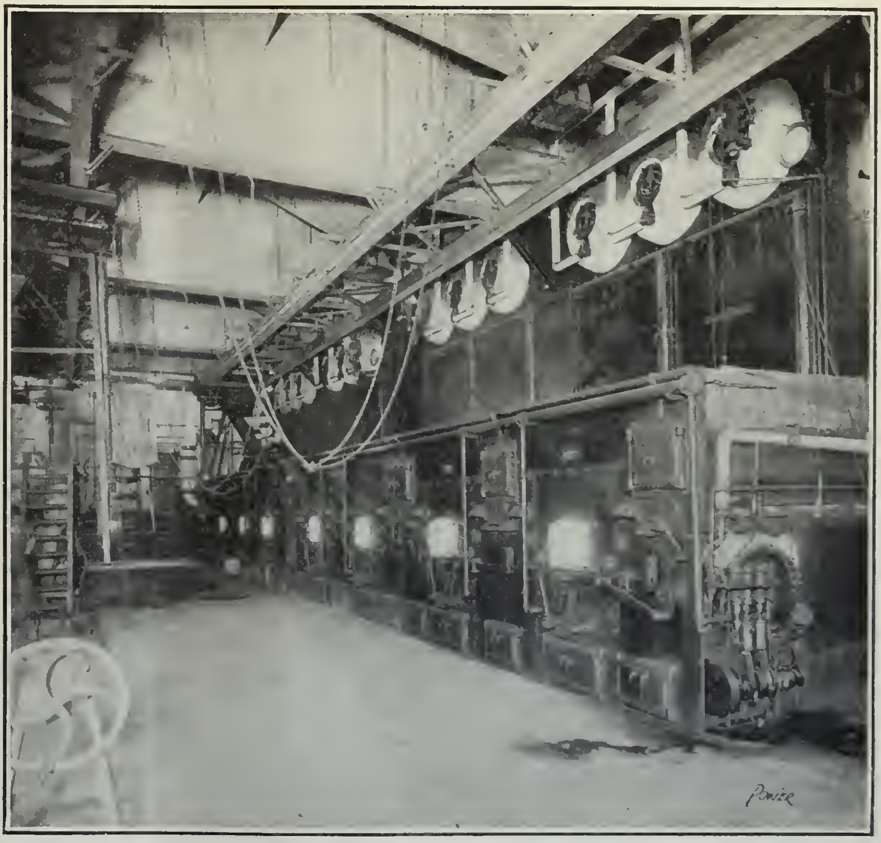


Fig. 3. View of the Boiler Room

All feed water is heated in a Hoppes open heater to 200 degrees and leaves the Green economizer at a temperature of from 245 to 265 degrees Fahrenheit. It is supplied to the boilers by two pumps: one a Cameron simplex steam pump, which is held in reserve; the other a D'Olier centrifugal pump direct coupled to a Terry turbine. These units are shown in Fig. 5, with the simplex pump in the foreground. The centrifugal pump in a 24-hour run averages approximately 10,000 cubic feet of water per hour, the night consumption being about one-third of the amount required during the day.

A main feed pipe to each battery of boilers is run above the steam drums, and branch pipes drop down and connect to the three drums of each boiler. The feed water is regulated by hand by means of an extension rod from the valve wheel extending down to within reach of the firemen.

Disposal of Ashes

Ash disposal in some plants is an item of considerable expense. Any arrangement that is efficient, and at the same time simple to operate, should be a success. These two features are found in the ash-handling system in this plant, as will be apparent from the view of the ash-conveyer tunnel, shown in Fig. 6. This tunnel extends the entire length of the boiler house, centrally between the two rows of boilers, to the coal tipple.

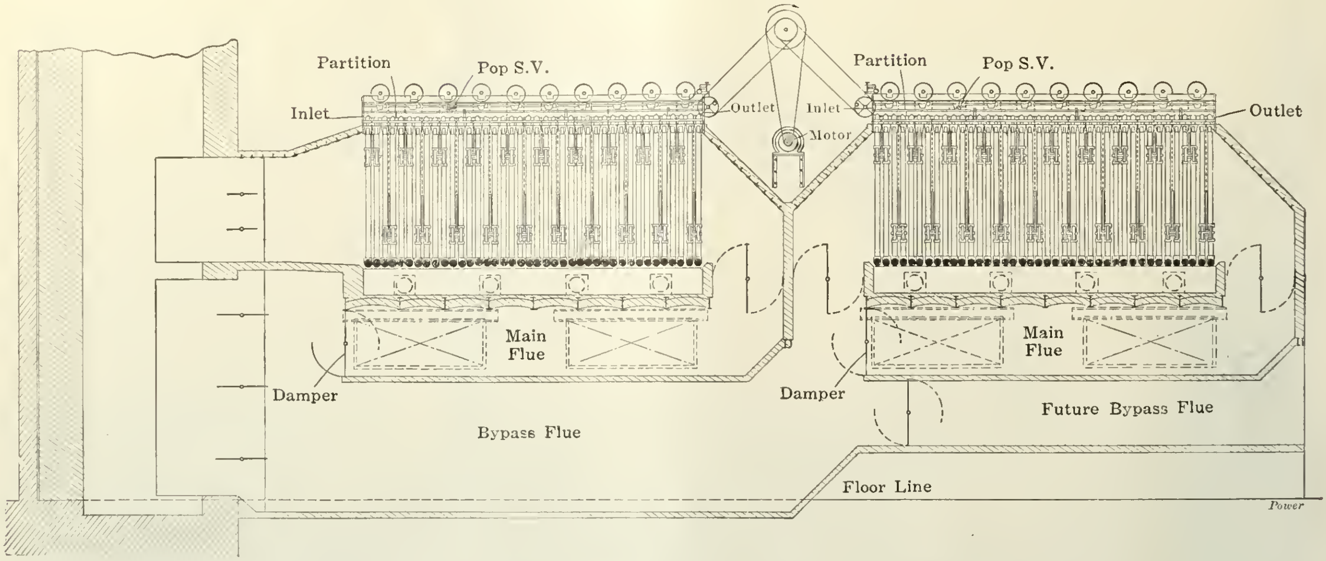


Fig. 4. Arrangement of Two Economizers, Showing Method of Bypassing the Gases

The ashpit under the grate of each boiler is constructed with a bottom sloping at an angle of 45 degrees. The outlet of each ashpit leads directly into the conveyer tunnel. In the foreground two of the ashpit doors are open and the chute dropped ready for the ash man to hoe the ashes out into the moving hopper buckets which convey it to the upper concrete tipple, shown in Fig. 2. Here the ashes are dumped and are loaded from time to time into a wagon placed under the delivery chute. When an ashpit has been cleaned the door is closed and the chute lifted and secured in position, as shown at the further end of Fig. 6. As the sides of the elongated hopper are lined with sheet iron, no difficulty is encountered from the ashes "hanging up." The same conveyer buckets are used for handling both coal and ashes.

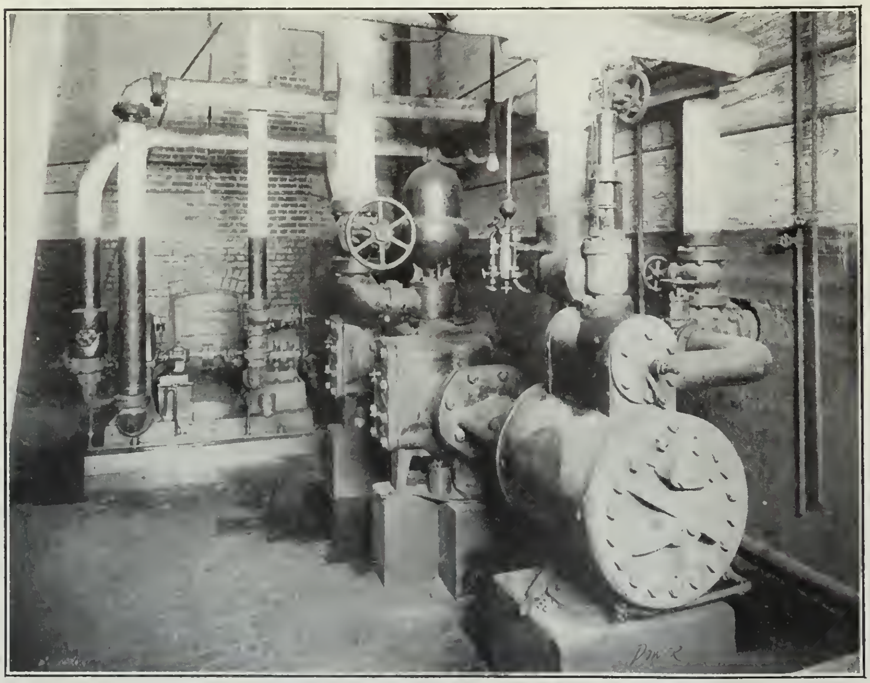


Fig. 5. Boiler-feed Pumps

Turbine Room

Stepping from the boiler room into the generating room, one is almost amazed at the space devoted to this part of the plant. Nothing is cramped, and every machine is set with order and precision. The various types of the same machine are grouped and arranged in a manner not easily improved upon.

Referring to Fig 1 again, it may be seen that the electrical generating units are grouped at one end of the generating room. The larger turbine is of 1000 kilowatts capacity and is used to furnish the electrical current for manufacturing purposes. The smaller turbine is of 500 kilowatts capacity and is used to carry the night load. Next to the iron supporting columns is the De Laval turbine of 200 kilowatts capacity. This unit is held as a reserve, but can be coupled to the 500-kilowatt unit in case the large turbine generator should become disabled. The two Allis-Chalmers turbines are run condensing, but the De Laval turbine is run noncondensing.

In the foreground a motor-driven direct-coupled exciter unit is shown, also a steam-driven direct-coupled unit used for the same purpose; the latter, however, is held in reserve. At the generator end of the turbines, not shown, is the switchboard from which the electrical output of the plant is controlled.

Over these machines is a 25-ton Niles motor-operated traveling crane for lifting the heavy parts of these units when necessary.

A near view of the refrigerating machines is shown in Fig. 8, also the air compressors and hydraulic pumps. There are two York refrigerating machines, each of 30-ton capacity and of standard make. One feature worth noting is the arrangement of the condenser coils. There are two sets of double pipe coils, but either one can be used with either machine, or both used with either of the machines. Steam is delivered to the steam cylinder from the vertical pipe which connects with a main steam pipe in the basement.

A motor- and one steam-driven air compressor furnish compressed air at a pressure of 70 pounds per square inch. The air is conveyed to the factory buildings through a pipe placed in a tunnel, which also accommodates the main feeder wires, hot- and cold-water pipes, live-steam and all return pipes. Weather conditions have no effect upon the distributing end of the plant. The belt-driven air compressor receives its power from a 50-horsepower Bullock induction motor located in the basement. The motor-driven unit is used during the summer months, and the steam-driven unit during the winter months when the exhaust steam can be utilized in heating the water used for heating the shops.

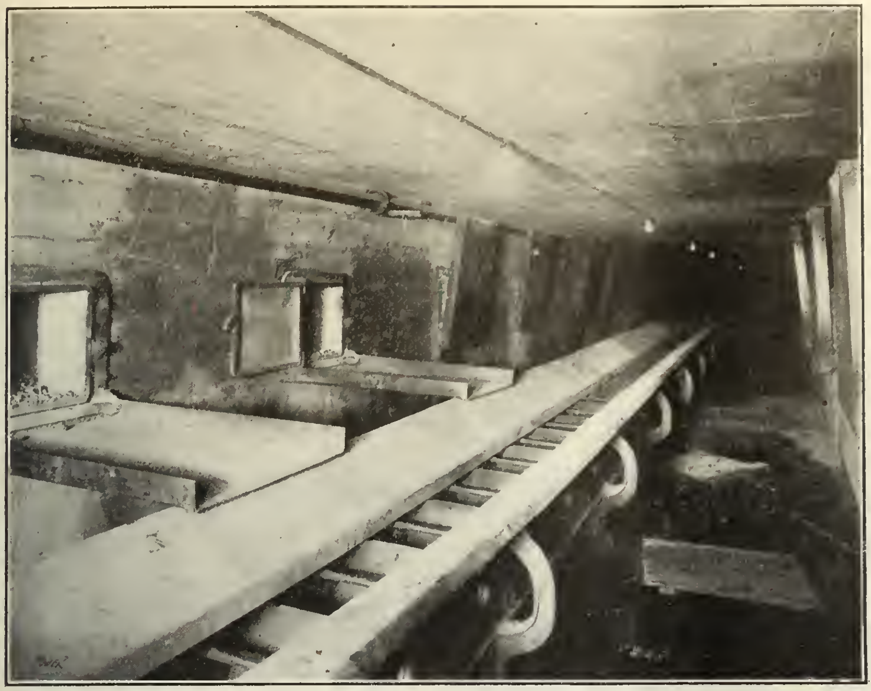


Fig. 6. Method of Disposing of Ashes

Homemade Accumulator

Between the air compressor and the side wall are two hydraulic pumps which work against a pressure of 2500 pounds per square inch. A homemade accumulator weight keeps the pressure in the delivery-pipe lines constant. The construction of this weight is shown in detail in Fig. 9. It weighs 95,000 pounds and is made of cement in which scrap iron and broken sections of concrete sidewalls are mixed. It is made in sections, two to each layer, with the exception of the bottom layer which is made solid. When constructing the weight a sheet of tar paper was placed between each layer and section, which allows the removal of any section without breaking the neighboring one. Eye bolts are set in each concrete slab below the top surface, which makes it an easy matter to lift them with the crane. These slabs are each 4 feet 3 inches wide, 8 feet 6 inches long and 2 feet thick. These, together with the solid, bottom slab, make a weight 8 feet 6 inches square and 8 feet 10 inches high, consisting of nine blocks of concrete. The concrete of each block was made of a one, four and seven mixture of cement, sand, scrap iron and old concrete. This weight cost in round numbers $250. Had it been made of cast iron at 1½ cents a pound (as quoted), the cost would have been $1425; therefore a saving of $1175 was made, and, although occupying a little more space, the concrete weight answers the purpose exactly as well as a more expensive cast-iron weight.

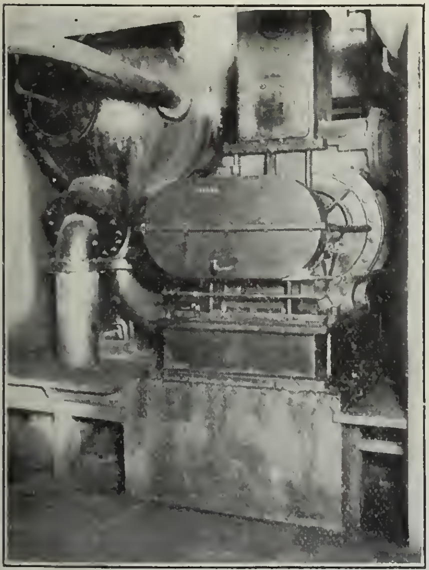


Fig. 7. Rotary Vacuum Pump and Jet Condenser

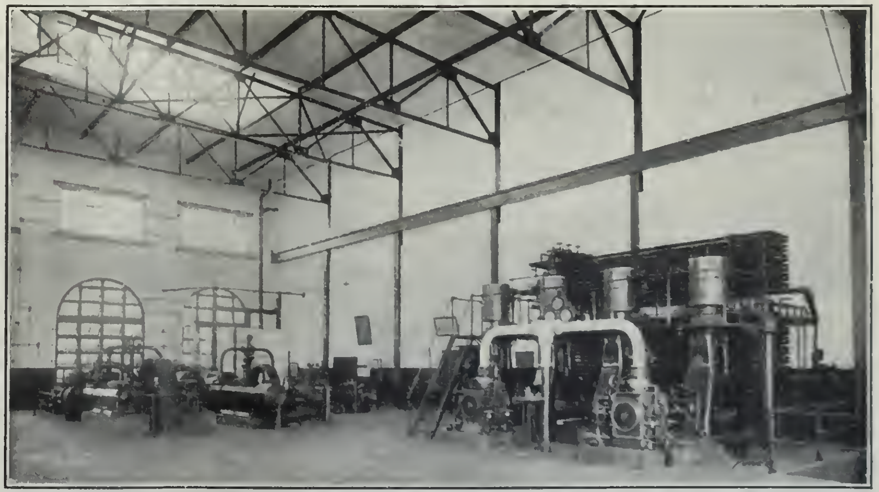


Fig. 8. Air Compressors, Hydraulic Pumps and Refrigerating Machines

The weight surrounds a central steel chamber in which the water is confined, and as the pressure increases, the weight is raised. The inlet connection from the pump is at the bottom of the column. A 6x6-inch wooden guide frame which has a cross piece fitted to the top steadies the weight. On each end of this cross piece three rollers are fitted. A wide-face pulley rolls on the edge of a strip of 3x⅜-inch tee iron; the other two rollers bear against the sides of the tee iron.

Attached to a top and bottom corner of the concrete weight are two angle irons. A cable is connected to the top angle iron by means of a turnbuckle. This cable extends down to a flanged pulley, and, after being wound twice around it, is extended to another flanged pulley attached to the throttle-valve stem of the steam pipe feeding the steam hydraulic pumps. The cable passes back to the first pulley and, passing over the top, is secured to the bottom angle iron. As the concrete weight rises and falls the valve is opened and closed.

Condensers

The condensers for the two condensing turbines, Fig. 7, are of the jet type, and were made by the Allis-Chalmers Company. The 1000-kilowatt unit condenser is equipped with a Connersville Blower Company's rotary cycloidal vacuum pump. It

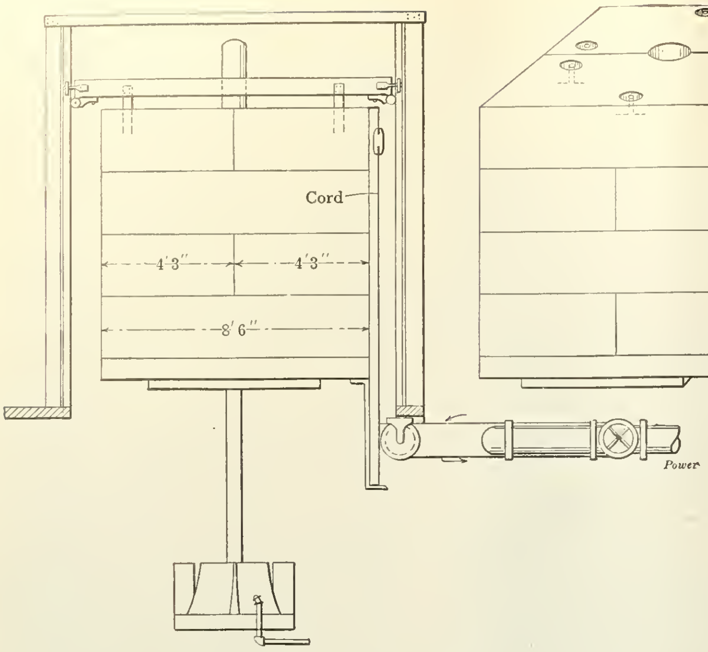


FIG. 9. DETAILS OF ACCUMULATOR WEIGHT

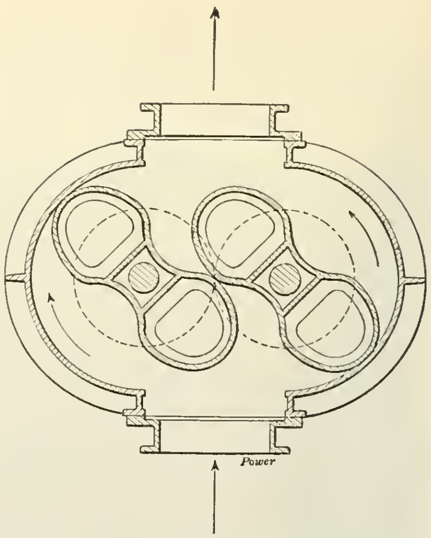


FIG. 11. CONSTRUCTION OF VACUUM PUMP

is driven by a piston-valve engine, direct connected, and runs at a speed of 80 revolutions per minute. The displacement is 35½ gallons per revolution. The arrangement of the vacuum pump on its foundation is shown in Fig. 10.

The 500-kilowatt turbine is fitted with a jet condenser; the same make of air pump is used as with the large machine. The pump is coupled to a piston-valve engine which runs at 95 revolutions per minute. The displacement of the pump is 13.10 gallons per revolution.

A sketch of the interior construction of the vacuum pump is shown in Fig. 11. The outline of the two so-called impellers of this pump is made up of cycloidal curves, which construction eliminates pockets and compression spaces. There is no metallic contact between the impellers or casing, as a slight clearance is allowed. The arrows indicate the direction of water flow.

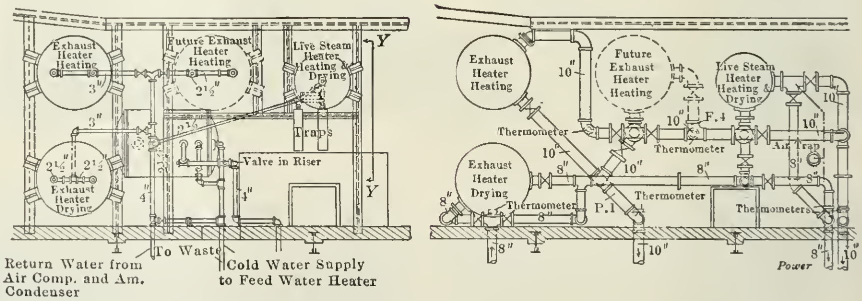


FIG. 12. TOP AND BOTTOM PLAN VIEWS OF HOT-WATER HEATING SYSTEM

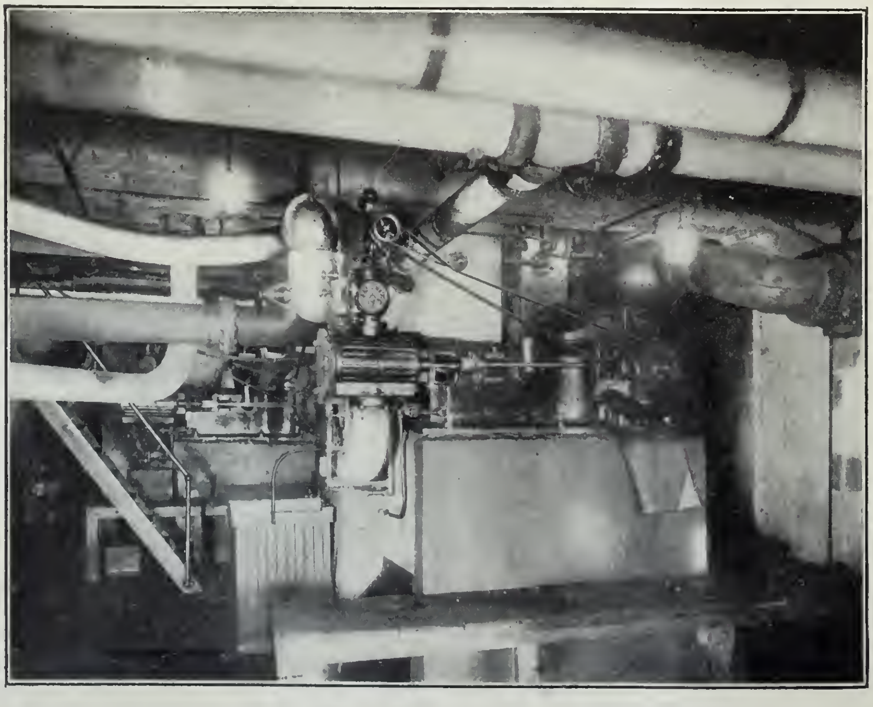


FIG. 10. ENGINE DRIVING VACUUM PUMPS

The water discharged from the vacuum pumps goes to a hotwell from which it is taken by a motor-driven centrifugal pump and delivered to a C. H. Wheeler cooling tower located in the yard. The circulating water flows from the cooling tower to a large cold well, and from the reservoir to the condensers by suction.

The water-cooling tower fans are operated by a motor, but a steam engine, located in the basement, is held as a reserve unit. A large fire pump is also installed in the basement which in case of fire to the factory buildings would doubtless do efficient work.

HOT-WATER HEATING SYSTEM

The hot-water exhaust-heating system was designed by Evans, Almirall & Co. Fig. 13 shows a side and end view of the two systems of heaters and their pipe connections. Fig. 12 shows a top and bottom plan view. There are two separate heating systems which are connected to a live-steam heater, as shown. The Hoppes feed-water heater is placed below the exhaust-steam heaters and the exhaust steam for the high-pressure De Laval turbine, and the steam auxiliaries escape to the atmosphere through the exhaust header, or can be bypassed to the exhaust-steam heater, as shown in the end elevation of Fig. 13.

POWER PLANT EQUIPMENT.

BUILDING		BRICK.
Boiler room	100 feet wide x 130 feet long	One firing aisle.
Generating room	100 feet wide x 118 feet long	Separated from boiler room by brick wall—concrete floor.
Boilers	12 B. & W	Final total.
At present installed	Six	In two rows.
Size	5000 square feet heating surface	Three steam drums.
Pressure	175 pounds	Saturated steam.
Superheat		None.
Grates	Mechanically operated	Murphy stokers.
Steam piping	6-inch manifold, 15-inch header	8" expansion bend placed horizontally over boilers.
Feed-water heater (1)	Open type	Hoppes
Boiler-feed pump (1)	14 and 9 by 18 inch	Horizontal simplex, Cameron.
" " " (1)	2½-inch, 4-stage	D'Olier centrifugal, turbine driven.
Hot-water heating system		
Pumps (3)	Centrifugal De Laval	De Laval turbine driven.
Hotwell pump	Centrifugal, D'Olier, 10"	Synchronous motor driven.
Hydraulic pump (1)	21 and 3⅓ by 24 inch	Knowles duplex.
" " (1)	19, 30 and 4½ by 24 inch	Worthington comp. duplex.
Fire pump (1)	20 and 12 by 16 inch	Knowles duplex.
Ammonia-condenser pump (1)	6 and 7½ by 6 inch	Worthington duplex.
Coal and ash handling		
Conveyer	Link-Belt bucket type	15-horsepower motor.
Capacity	30 tons per hour	Crusher motor, 15 horsepower.
Bunker (1)	Six compartments	Total capacity 1500 tons.
Foundations	Concrete	Turbines, pumps, etc.
Turbo-generators (1)	1000 kilowatts	Allis-Chalmers.
" " (1)	500 kilowatts	Allis-Chalmers.
" " (1)	200 kilowatts	De Laval (two 200-kilowatt generators).
Speed	1800, 3600, 10,500 r.p.m. respectively	De Laval generator, 900 r.p.m.
Generators	3-phase, 60-cycle, 480 volts	Bullock.
Condenser (2)	Jet, Allis-Chalmers	Rotary cycloidal vacuum pump.
Auxiliary turbines		
Hot-water circulating pump turbine (2)	4400 r.p.m	De Laval turbine.
" " " " " (1)	1700 r.p.m	Motor driven.
Turbine-driving boiler-feed pump (1)	2500 r.p.m	Terry turbine.
Exciters		
Motor driven (1)	D. C., 50 kilowatts 125 volts	Used as a general exciter unit, Crocker-Wheeler.
Steam driven (1)	D. C., 27 kilowatts, 125 volts	Emergency unit, General Electric.
Steam engines (1)	10x12 inch, speed 305 r.p.m	
" " (1)	11x16 inch, speed 90 r.p.m	Fan engine.
" " (2)	8x10 inch, speed 97 r.p.m	Vacuum pump.
" " (1)	12x12 inch, speed 80 r.p.m.	" "
Motor (1)	60 cycle, 3-phase	Crocker-Wheeler driving exciter.
" (1)	50 horsepower, 3-phase, 60-cycle, 850 r.p.m	Bullock driving air compressor.
" (1)	3 horsepower, 1800 r.p.m	Operates economizer.
" (1)	6 horsepower, 1130 r.p.m	" "
" (1)	15 horsepower, 680 r.p.m	Operates coal conveyer.
" (1)	15 horsepower, 680 r.p.m	Operates coal crusher.
Ammonia compressor (2)	Steam cylinder 11½x15 inch	York, 30 tons each.
	Ammonia cylinder 11x15 inch	
Air compressor (1)	12x14 inch, speed 115 r.p.m	Motor driven.
" compressor (1)	Steam cylinder 11 and 18 by 14 inch	Steam driven.
	Air cylinder 12x14 inches	
	125 r.p.m	
Crane (1)	25-ton electric	Niles.
" (1)	10-ton hand	"

The water is circulated through a series of pipes in each heater by a turbine-driven centrifugal pump. From the heaters the water is sent to the factory radiators through a single supply pipe and is returned to either of the three centrifugal pumps through a single return pipe. As the head of water in both the suction and delivery pipes is balanced, the work of the pump is confined to overcoming friction in the pipes. The water is circulated over and over again, and the loss is slight.

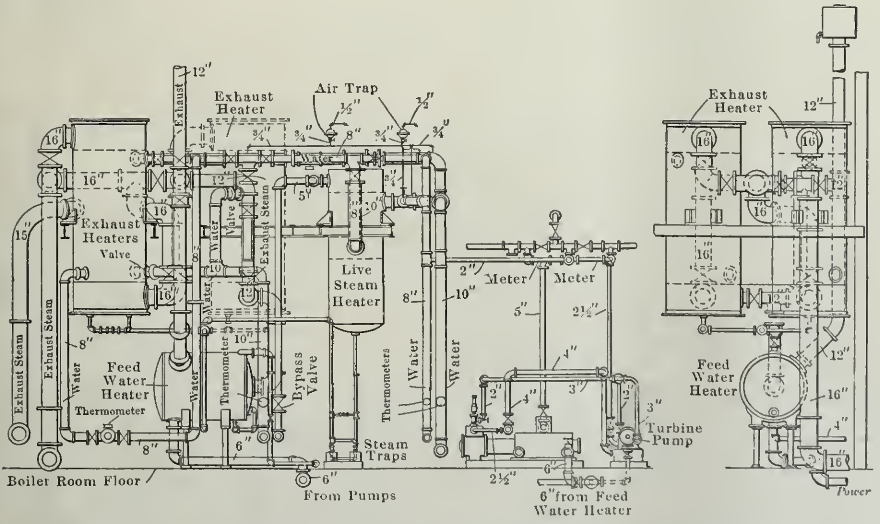


Fig. 13. Side and End Elevation of Hot-water Heating System

Exhaust steam is passed through the tubes of the heaters and the water is circulated on the outside.

Besides the two exhaust-steam heaters there is a live-steam heater, so that in case the exhaust steam does not heat the water to the temperature desired, live steam can be used. The live and the exhaust steam are never mixed, the two heaters being entirely separate. This permits of returning the live steam to the boiler without passing through a purifying device. The piping is so arranged that the water can be sent through either or both heaters. The water first passes through the Hoppes heater to the bottom exhaust heaters, and then passes out at the top to the heating system. The exhaust steam from the auxiliary engines first passes through either or both of the exhaust heaters to the Hoppes heater and then to the exhaust pipe and the atmosphere. Bypasses are arranged so that either the feed-water heater or the exhaust heaters may be cut out of service.

All steam, water and air pipes are placed in the basement. The main steam pipe is 15 inches in diameter and is fitted with van-stone joints. Long-radius bends are used on all live-steam piping throughout the plant. The table gives details regarding the size, type and use of the various machinery installed in the plant.

The operating force of this plant consists of ten men, which includes both day and night shifts.

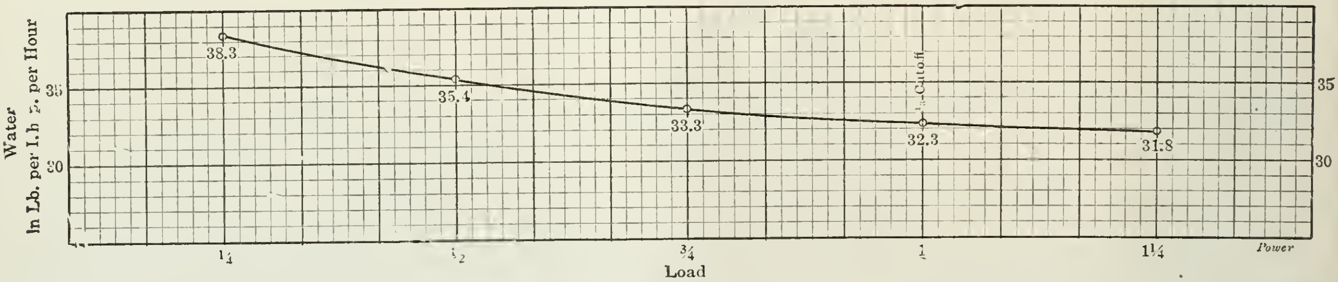
Water Rates for Simple Engines

By Wallace S. Fowler

By using the accompanying curve as a basis the water rates for various sizes of simple engines, both condensing and noncondensing, may be figured for different steam pressures and conditions of load.

While connected with a well known concern manufacturing small steam engines, the writer was assigned the task of procuring and tabulating data relative to the power, efficiency and water consumption of the various sizes of engines operating under different conditions of load and steam pressure, both condensing and noncondensing.

Accordingly, complete data were obtained covering sizes from 5x5 to 16x16 inches, with steam pressures ranging from 40 to 200 pounds and loads varying from 25 per cent. normal to 25 per cent. overload. The engines were all of the ordinary piston or slide-valve type, direct connected to generators, and full load was based at one-third cutoff. The problem now was to condense this mass of data into such form as would be readily available to the salesmen and engineers who were to use it. A 14x14-inch engine running noncondensing on 100 pounds steam pressure was chosen as a basis for constructing the fundamental curve shown in the cut, from which the performance of the other sizes might be figured. The water rate, expressed in pounds of water per indicated horsepower per hour, was spaced off as ordinates; the load, expressed in fractions of the normal load, was laid off as abscissas, and a smooth curve was drawn through the points of intersection. This done, cards showing the power, efficiency and water consumption of the standard sets, were made similar to the form shown in the table. These, together with the curve for figuring odd sizes, made a ready-reference library, which was convenient and reasonably correct.



CURVE OF WATER CONSUMPTION. SIMPLE ENGINES AT DIFFERENT LOADS, CONDENSING AND NONCONDENSING

The curve shows the water rate for a 14x14-inch engine running noncondensing on 100 pounds steam pressure. For condensing engines subtract 15 per cent. For smaller sizes add 3 per cent. for each inch decrease in the stroke. For larger sizes subtract 2½ per cent. for each inch increase in the stroke. If steam is above 100 pounds and noncondensing subtract ½ pound for each 10-pound increase. If steam is below 100 pounds and noncondensing add 1 pound for each 10-pound decrease. If steam is above 100 pounds and condensing subtract ¼ pound for each 10-pound increase. If steam is below 100 pounds and condensing add ½ pound for each 10-pound decrease.

Perhaps the greatest merit of a curve of this kind lies in the aid it renders to the rapid comparison in economy of an engine under different conditions, or of different engines under the same or different conditions. A practical example will serve best to illustrate this point. Suppose that a prospective purchaser requires an engine that will deliver 38 brake horsepower for ten hours per day. He has two engines under consideration, one a 7x7-inch running at 375 revolutions per minute, to be run at one-third cutoff on a steam pressure of 150 pounds gage; the other, a 9x9-inch running 300 revolutions per minute, also to be run at one-third cutoff, but on 90 pounds gage. Both are noncondensing, and either will deliver 38 horsepower. The curve will help him to decide by showing which engine will do the work with the least expenditure of steam. First, let it be determined how much steam the 7x7-inch engine will use. Starting with the point marked 32.3, which is the water rate at full load, and following carefully the rules given below the curve, the problem can be expressed as follows:

$$32.3 + (7 \times 0.03) - (5 \times \tfrac{1}{2}) = 36.6$$

pounds of water per indicated horsepower per hour. Assuming a mechanical efficiency of 90 per cent., this equals,

$$\frac{100 \times 36.6}{90} = 40.7$$

pounds of water per brake horsepower per hour.

Proceeding in the same manner for the 9x9-inch engine,

$$32.3 + (5 \times 0.03) + (1 \times 1) = 38.2$$

pounds of water per indicated horsepower per hour, and, as in the other case with a mechanical efficiency of 90 per cent., it equals,

$$\frac{100 \times 38.2}{90} = 42.5$$

pounds of water per indicated horsepower per hour.

Having thus determined the water consumption, the corresponding coal consumption can be found by substituting the proper values in the following expression:

$$C = \frac{W \times H}{V \times E}$$

where

C = Coal burned per brake horsepower per hour;

W = Water used per brake horsepower per hour;

WATER CONSUMPTION OF 14x14-INCH SIMPLE ENGINE.

120 Pounds Steam Pressure, Non-Condensing, 250 Revolutions per Minute, 100 Kilowatt Generator.

Load.	Output in Kilowatts.	Output in Effective Horsepower.	Brake Horsepower of Engine.	Indicated Horsepower of Engine.	Efficiency of Generator.	Efficiency of Engine.	Combined Efficiency.	Water per Kilowatt.	Water per Effective Horsepower-hour.	Water per Indicated Horsepower-hour.	Water per Brake Horsepower-hour.
1¼	125	167.5	185	196.2	90.5	94.3	85.4	48.3	36.2	30.8	32.6
1	100	134	149	160.2	90	93	83.7	50.2	37.4	31.3	33.7
¾	75	100.5	113	124.2	89	91	81	53.5	39.5	32.3	35.5
½	50	67	77	88.2	87	87.4	76	60.7	45.3	34.4	39.4
¼	25	33.5	42.7	53.9	78.5	79.2	62.2	80.5	60.1	37.3	47.2

H = Total heat in the steam above the temperature of the feed water;
V = Heating value of the coal;
E = Efficiency of the boiler.

In the foregoing problem assume that
V = 13,000 B.t.u.;
E = 65 per cent; and that the temperature of the feed water is 32 degrees Fahrenheit.

Then from the steam tables H is found to be 1193.5 B.t.u. for 150 pounds gage and 1182.9 B.t.u. for 90 pounds gage. Substituting these values for the 7x7-inch engine,

$$C = \frac{40.7 \times 1193.5}{13{,}000 \times 0.65} = 5.75$$

pounds of coal per brake horsepower per hour, and for the 9x9-inch engine,

$$C = \frac{42.5 \times 1182.9}{13{,}000 \times 0.65} = 5.95$$

pounds of coal per brake horsepower per hour.

By this it is shown that if the purchaser were to install the higher pressure engine his coal bill would be 3.36 per cent. less than it would be with the other engine. Having found the coal burned per brake horsepower per hour, it is a simple matter to compute the coal bill for the month or the entire year for that matter.

Problem in Expansion of Steam

By John French

> *Given the initial pressure in an engine cylinder and the ratio of expansion, an approximate solution for determining the pressure at the end of the stroke is presented.*

There have appeared at various times articles containing problems involving the expansion of steam. Most of the writers seem to have a rather hazy idea of just how to solve such problems.

The accurate solution is rather long and tedious, but there are approximate solutions which are fairly accurate.

Assume a problem in which dry saturated steam enters a cylinder at a pressure of 100 pounds gage and it is required to find the pressure of the steam at the end of the stroke, with an expansion ratio of three. If a temperature-entropy diagram is drawn, as shown in Fig. 1, it is a graphical history of the steam, where AC is the water line and DF the saturated steam line. If the steam could expand along the line DF, which it cannot do unless heat is added, it would be a very easy matter to find the pressure for any volume direct from the steam tables. The most probable expansion line will be approximately an adiabatic such as appears in the ordinary treatment of the Rankine and Carnot cycles and is shown as DE in Fig. 1, the complete Rankine or Carnot cycle being $BCDEB$.

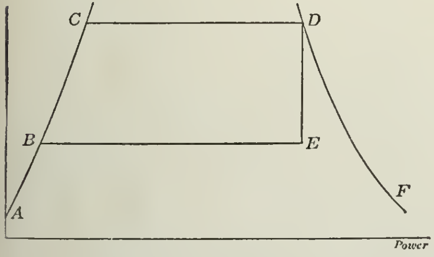


FIG. 1. TEMPERATURE-ENTROPY DIAGRAM

A crude approximate solution is obtained by assuming the law

$$p\,v = Constant,$$

and a more accurate approximate solution would be given by assuming

$$p\,v^{1.11} = Constant$$

If the problem given above is solved by the equation

$$p\,v = Constant,$$

then,

$$p_1 v_1 = p_2 v_2$$

$$115 \times 1 = p_2 \times 3$$

$$p_2 = \frac{115}{3} = 38.3 \text{ pounds absolute}$$

or 23.3 pounds gage, assuming for the sake of simplicity atmospheric pressure to be 15 pounds. Using

$$p\,v^{1.11} = Constant$$

the solution is

$$p_1 v_1^{1.11} = p_2 v_2^{1.11}$$

$$115 \times 1 = p_2 \times 3^{1.11}$$

and

$$p_2 = \frac{115}{3.389} = 33.9 \text{ pounds absolute}$$

or 18.9 pounds gage.

If an exact solution is made by the plotting of pressure and volume for constant entropy, the procedure is as follows:

From the result found by using the equation

$$p\,v^{1.11} = Constant$$

it is known that the pressure will be somewhere in the neighborhood of 34 pounds absolute. Assuming the steam to have expanded adiabatically until the pressure is 34 pounds, the first factor to be found is the quality at this pressure. Then the volume is found and if it is equal to three times the volume of the steam at the initial pressure, the required pressure has been ascertained. Using Reeve's steam tables it will be found that the volume of one pound of steam at 115 pounds absolute is 3.864 cubic feet so that the final volume will be 11.592 cubic feet.

The entropy of dry saturated steam at 115 pounds absolute is equal to the entropy of the water plus the entropy of vaporization for that pressure and the entropy of dry saturated steam at 34 pounds is equal to the entropy of the water plus the entropy of vaporization for that pressure. But at the final pressure the steam is not dry, so the entropy will be equal to the entropy of the water plus the entropy of vaporization multiplied by the quality.

Let E_1 and E_2 be the entropy for 115 and 34 pounds, and x the quality. Since the entropy at both pressures is equal,

$$E_{w_1} + E_{v_1} = E_{w_2} + x\,E_{v_2}$$

$$x = \frac{E_{w_1} + E_{v_1} - E_{w_2}}{E_{v_2}}$$

and

$$x = \frac{0.4864 + 1.0969 - 0.3784}{1.2999} = 92.66$$

The relative volume of dry saturated steam at 34 pounds is 12.11, so that the volume of the steam with a quality of 92.66 is 11.22 cubic feet. This is less than the volume required; hence it is necessary to repeat the same process for 33, 32 and 31 pounds. This can most easily be done by arranging a table as follows:

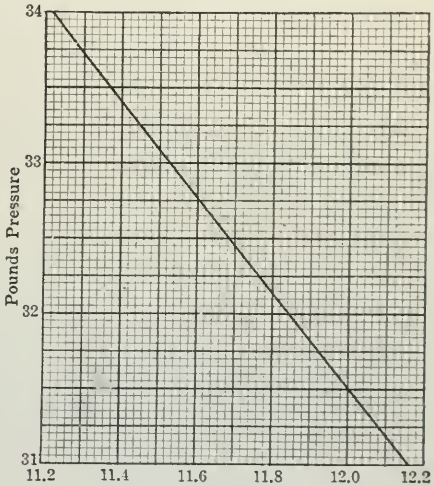


FIG. 2. PRESSURE-VOLUME CHART

Pressure.	Volume.	E_w	E_v	x	xV
34	12.11	0.3784	1.2999	92.66	11.22
33	12.16	0.3759	1.3048	92.55	11.53
32	12.82	0.3734	1.3098	92.37	11.84
31	13.20	0.3709	1.3149	92.19	12.17

With the values of $x\,V$ as abscissas and those of pressure as ordinates, the chart shown in Fig. 2 can be plotted. At a volume of 11.59 the pressure 32.8 pounds absolute or 17.8 pounds gage will be found, which is the exact pressure of the steam after expanding under the conditions stated in the assumption.

Operating vs. Consulting Engineer

By M. Kennett

An instance where the consulting engineer departed from the custom of heeding the suggestions of the chief, with the result that the layout proved so unsatisfactory as to require complete rearrangement of the equipment involved.

The majority of operating engineers are inclined to feel more or less jealous and resentful toward the consulting engineer who is called in to plan alterations or additions to their plants. This feeling is usually unjustified, as the consulting engineer is undoubtedly better qualified for this work than the operating engineer. Furthermore, my experience has been that the consulting engineer is usually ready to consider suggestions from the operating engineer, and in this way, by combining the technical knowledge of the former with the practical experience of the latter, a much more efficient layout is obtained than would be possible by relying upon either one alone. It was my misfortune, however, to come in contact with a striking exception to such a consulting engineer in connection with the installation of a new pump in an electric-power and pumping plant, in which I was employed some years ago.

This plant was built on the bank of a river, about 35 feet above the low-water level and some 400 feet back from the river; the overflow during periods of high water making it impractical to build the plant closer to the stream. The engine and boiler rooms were built on the surface, but owing to the elevation, the pumps were placed in an oval pit about 15 feet deep and located just in front of the boiler room, as shown in Fig. 1. In this pit were placed two compound direct-acting pumps, one of 1,000,000, and the other of 750,000 gallons capacity. During most of the year these pumps worked on a suction lift of from 10 to 12 feet, but during low water this was increased to about 20 feet in addition to the horizontal pull of about 400 feet, and at times considerable difficulty was experienced in operating them at this lift. Hence, when the increase in business demanded an additional pump, it was decided to place it about 5 feet lower than the old ones. As there was no room in the old pit, it was necessary to sink a new one just in front of it and connected to it by a flight of narrow steps; see Fig. 1.

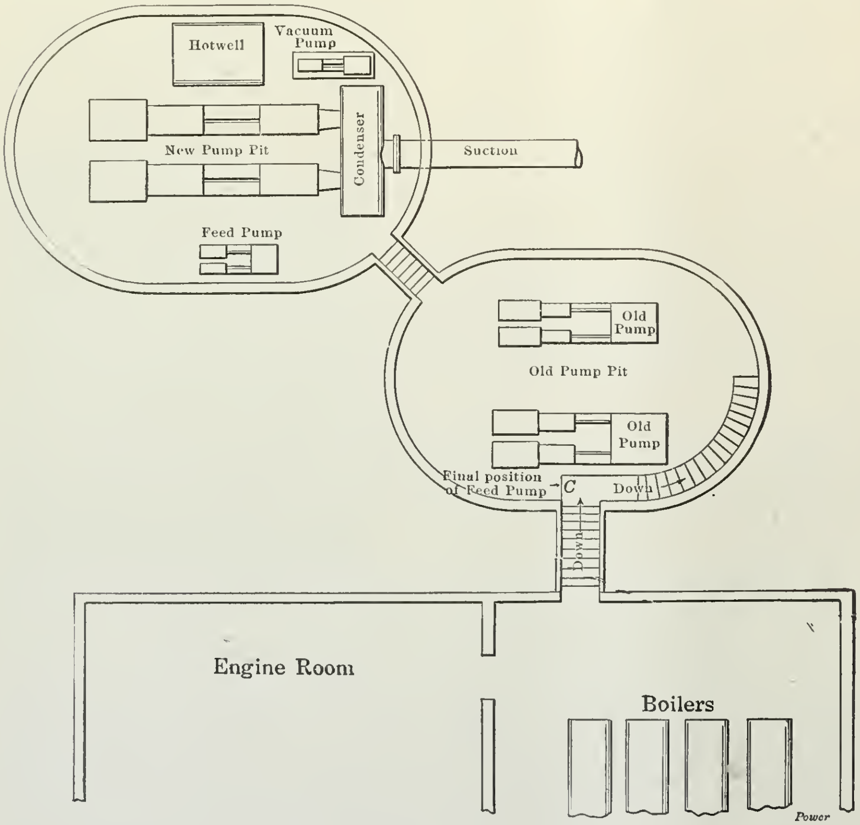


Fig. 1. Layout of Pump Pits

Although this new pit was directly in front of the engine room, no stairway from there was provided, as the consulting engineer claimed that it would be necessary only to go down to the new pump room two or three times a day. As will be seen from the plan, it could be reached only by a long route through the old pit, and the new pump could not be seen at all without going to the entrance of the new pit. When the engines were running, it was impossible even to hear the pump, and the only means of judging what was going on was by means of the gages in the engine room.

The new pump was of the compound, condensing type, having steam-jacketed cylinders and a surface condenser in the suction line, so that the entire volume of water flowing through the suction constituted the circulating water. The water end was of the outside-packed plunger type, and the pump was rated at 1,500,000 gallons capacity.

The vacuum pump was placed at the end of the condenser, and in order that the water might flow to it by gravity, it was located in a small pit, about 4 feet long, and 18 inches wide. This vacuum pump discharged into a hotwell which consisted of a cylindrical tank placed on its side on the floor. By means of a float valve the makeup water for the boilers was supplied to this hotwell, from which the feed for the entire plant was taken. The supposition was that when only this pump was running, the water in the hotwell would be too hot to be lifted by suction; consequently, the feed pump was located on the floor of the pump pit, on the side opposite the vacuum pump. This was the layout as planned by the consulting engineer. The chief engineer of the plant, who was a very capable man, found several objections to it, but was unsuccessful in his efforts to have the plans altered, and the installation was completed as outlined.

Some of the objections which the chief raised were that the feed pump was in a very inconvenient place, where it could not be seen by the fireman without going down two flights of steps. Although it could be regulated from the boiler room by a valve in the steam line, it is always more satisfactory to see a pump in order to note the changes in its speed while regulating it. Another objection was that all the seepage, which was considerable, together with the drips from the plungers and stuffing boxes, would naturally flow into the vacuum-pump pit which did not have any drainage and really acted as a sump, and as a result had to be emptied by a jet provided for the purpose. This pit usually filled in about two hours, and the grit soon cut the links and rods of the vacuum pump, and its cramped position made it very difficult to pack or repair the pump.

These were some of the most serious objections to the proposed plan, and to overcome them to some extent, the chief suggested getting the hotwell out of the way by elevating it about 6 feet, and placing the feed pump on the landing at the bottom of the first flight of steps leading from the boiler room, as at *C*, Fig. 1. This would have located it in plain view from the boiler room, and given it a suction lift of only 5 feet. Furthermore, the chief did not believe the water in the hotwell would ever become hot enough to cause any trouble on this lift.

These suggestions were promptly vetoed by the consulting engineer on the ground that to elevate the hotwell would cause the vacuum pump to operate against a head, thus imposing additional work on it, which might lower the total duty of the pump to less than the guarantee.

To overcome this argument, the chief proposed disconnecting the discharge pipe during the test and letting the vacuum pump discharge on to the floor. But in spite of all arguments to the contrary, the consulting engineer ruled that the hotwell must remain on the floor. Nor could he see any objections to the feed pump being located in the lower pit, and contended that it would not handle the water from the hotwell on a 5-foot lift. He also considered it unnecessary to provide stairs from the engine room to the new pump pit, although their desirability would seem to be apparent to anyone.

Finally, after many tribulations, the installation was completed and the consulting engineer took leave, much to the relief of all. After a few weeks, when things had settled down to regular operating conditions once more, the chief quietly set to work to rearrange things. First, he had the hotwell raised about 6 feet, and suspended from the overhead beams, and then placed the feed pump where he had proposed, on the landing at *C*, which proved to be a much more convenient location; and I have never heard of its giving the least trouble in lifting the water.

The cylinders of the main pump were steam jacketed, and as originally installed, were drained by means of a trap. The chief left the trap in position, but cut it out of service by means of a valve, and took steam for the vacuum pump through the jackets. This kept the steam in better circulation, and kept the jackets thoroughly drained at all times, and the condensation was never enough to affect the operation of the vacuum pump.

An amusing incident occurred when the duty test was made on this installation. The manufacturers, of course, had their representative on hand to look after their interests and the consulting engineer represented the owner. In order to ascertain the total head against which the pump was operating, that is, the hight from the surface of the river to the surface of the water in the standpipe, a vacuum gage was placed on the suction pipe at *A*, Fig. 2. As each inch of vacuum is equal to 0.49 pound, the total pressure represented by the reading of this vacuum gage was added to that shown by the pressure gage, and this total divided by 0.433 gave the total head in feet against which the pump was working. As provision had been made at *B* for a gage, the consulting engineer placed one at that point, but the builder's man objected to this, claiming that the vertical hight between these two gages should be included in the total head. In other words, as this distance was about 6 feet he claimed that for every pound of water raised, the pump was developing 6 foot-pounds of energy for which it was not receiving credit, and he insisted that the pressure gage be located on the same level as the vacuum gage, as shown by the dotted lines in Fig. 2.

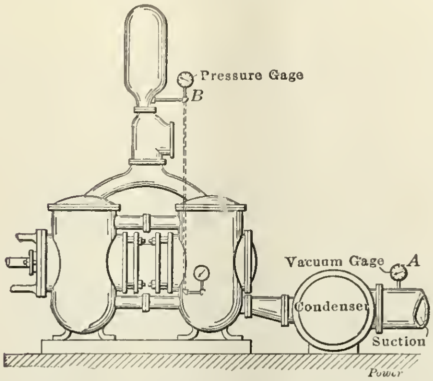


Fig. 2. Circulating Pump Showing Location of Gages

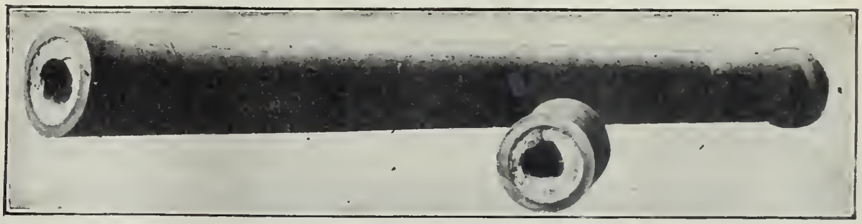
After considerable argument on this point it was evident that they could not agree, and each made the test with the gages where he thought proper, and the question was left to be decided later by a committee of consulting engineers. These men decided in favor of the builders, and under this decision the pump exceeded the guarantee by a considerable amount.

This test was more or less perfunctory, and no attempt was made to attain any great degree of accuracy. The pump was guaranteed to develop 55,000,000 foot-pounds of energy per 1000 pounds of steam at 100 pounds pressure at the throttle. However, no test was made to ascertain the quality of the steam, which must have been rather poor, as the pipe was about 100 feet long and no separator was provided, although the pipe was well protected by nonconducting covering. The quantity of water pumped was calculated by the displacement of the plungers, and the number of strokes as indicated by a counter, the length of the stroke being frequently checked during the run. An allowance of 5 per cent. was agreed upon for slip through the valves and packing. The discharge from the vacuum pump and from the trap which drained the cylinder jackets was carefully weighed and represented the total steam consumption. The head against which the pump was working was computed as described from the combined readings of the pressure gage on the discharge and the vacuum gage on the suction line. A short run of four or five hours was made, the consulting engineer and the representative of the builders each keeping their own notes. Naturally, there was quite a difference between the two results, which, as stated above, was finally decided in favor of the builders.

During the daylight run, the electrical load was very light, and considerable trouble was experienced in keeping the feed pump running slowly enough to maintain a constant water level in the boilers. When it was throttled down so as to run slowly enough, it frequently stopped altogether. A ¼-inch bypass with a globe valve in it was inserted between the discharge and the suction and the pump was then run at a conveniently slow speed, and the bypass was regulated according to the water level in the boilers. This was found more convenient than attempting to regulate the pump too closely.

Failure of Scaled Blowoff Pipe

One of the reasons a blowoff pipe burns out is because it is allowed to become partly filled with scale. The illustration shows a 2½-inch blowoff pipe which failed from this cause. It can readily be seen that with such an accumulation of scale, the pipe had but little protection against the furnace heat.



Scale in 2½-inch Blowoff Pipe

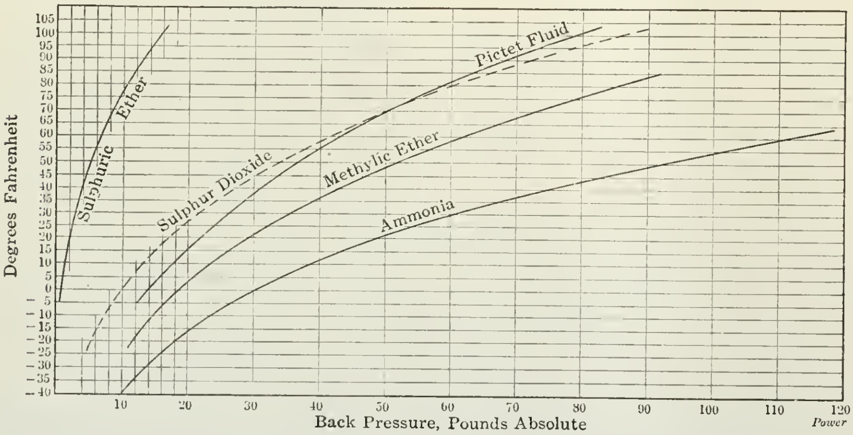
The pipe was lined with scale between the boiler and the blowoff pipe. The scale was of lime formation and exceedingly hard. A peculiar feature was that the scale seemed to vary in character, an end view showing different colored scale of varying thickness; the layers being easily distinguished. A blowoff pipe should be inspected just as frequently and as thoroughly as a boiler.

Pressures in Refrigerating System

By F. E. Matthews

The capacity of a refrigerating system may be increased by lowering the back pressure or by adding to the cooling surface. Some interesting data are given on temperatures and pressures and their relations.

It has already been pointed out that a given substance boils at different temperatures under different pressures; the boiling point being raised when the pressure is increased, and lowered when it is decreased. In the case of water, for example, which boils under atmospheric pressure at 212 degrees Fahrenheit, an increase in pressure to 70 pounds gage raises the boiling point to 316 degrees Fahrenheit, and a reduction in pressure to 29.74 inches vacuum lowers it to 32 degrees Fahrenheit, or to its freezing point. From this, since the law is a general one applying to all known liquefiable gases, it follows that to produce low temperatures the pressure on the refrigerating medium employed must be reduced to such a point that the corresponding boiling point will be a sufficient number of degrees below the temperatures to be produced to bring about the heat transfer through the expansion coils or other cooling surfaces. If, for example, it is desired to cool a cold-storage compartment at 10 degrees Fahrenheit, a back pressure of 24 pounds gage will be found too high to allow ammonia to boil at this temperature. At 23.64 pounds pressure it will boil at exactly 10 degrees Fahrenheit, but since this is the temperature of the surrounding air, there is no difference in temperature to bring about a heat flow and the boiling will not continue. When the pressure is reduced to 19.46 pounds gage, the ammonia will boil at 5 degrees Fahrenheit, and at this temperature there will be sufficient inflow of heat from the 10-degree surrounding air to cause quite appreciable refrigeration. A further reduction to 15.67 pounds gage lowers the temperature of the boiling ammonia to zero degree Fahrenheit and the increase in temperature difference from 5 to 10 degrees Fahrenheit will effect a rate of heat transfer just twice as great per square foot of pipe surface as was possible with half the difference. A still further reduction to 9.1 pounds gage pressure will allow the ammonia to boil at 10 degrees Fahrenheit, at which temperature the heat flow from the 10-degree room will be twice as great as it was at 15.67 pounds pressure and four times as great as it was at 19.46 pounds. In order to produce the same amount of cooling effect at 19.46 pounds pressure as was obtained at 9.1 pounds pressure, just four times as much pipe surface would have to be employed, and in order to do as much at 15.67 pounds, just twice the surface would be required.



VAPOR TENSIONS FOR VARIOUS MEDIA

If, instead of direct expansion, brine circulation is employed, it will be evident that for the same rate of heat flow, a lower pressure and temperature will be required in the latter case. Assuming, for example, that the rate of heat transmission per square foot per degree difference in temperature be the same between the ammonia and brine, and between the brine and air as it is between the ammonia and air (an assumption which is not wholly accurate but which will simplify the example), the heat transmission between the ammonia at 9.1 pounds pressure and the air at 10 degrees Fahrenheit would be only half as great in the case of the brine system as in the case of direct expansion. This because of the fact that 10 degrees difference in temperature must be allowed to cause a heat flow from the air to the brine and another 10 degrees for the flow from the brine to the ammonia. On this basis in order to produce the same heat flow, 9.1 pounds back pressure would have to be carried in the case of brine circulation against 15.67 pounds in the case of direct expansion.

Where the piping is installed and cannot be increased, there remains only one of the two variables. To increase its cooling capacity, therefore, lower back pressures must be employed.

The saving in first cost by the installation of scanty pipe surface always entails correspondingly lower back pressures, and is soon lost by increased operating expense due to decreased efficiency.

Where there is only one temperature to be produced in the cold-storage compartments a back pressure is usually carried such that the temperature corresponding to that pressure will be from 10 degrees or less on low-temperature work to 30 degrees or more on high-temperature work, below that of the cooler temperature. The lower the temperature to be produced, the lower the efficiency of the machine, and accordingly the more expense will be warranted in pipe area so as to increase transmission at a narrower range. Under average operating conditions the cost of the amount of expansion pipe required to allow this range in temperatures balances up fairly well with the loss in efficiency that would be encountered if less expansion piping were installed, and a lower back pressure carried.

Where several different temperatures are to be maintained with one back pressure, no fixed rule can be followed and each individual case must be figured out separately. If only a small percentage of the total cooling be low-temperature work, it is usually advisable to increase the surface and to reduce the temperature range between the liquid ammonia and the surrounding air in the lowest temperature compartment. In this case the use of an abnormal amount of pipe on a small percentage of the entire duty tends to increase the efficiency of the whole plant.

Some idea regarding the pressures that should be maintained in expansion coils when operating with different kinds of refrigerating media may be gained by reference to the curves, from which it may be seen that if an expansion temperature of zero degree Fahrenheit, such as would ordinarily be employed where the temperatures to be produced are from 10 to 20 degrees Fahrenheit, the expansion or "back pressures" required for these different refrigerating media are as follows: Ammonia, 30 pounds absolute; methyl ether, 19; pictet fluid, 14; sulphur dioxide, 10.

While no definite rules can be laid

down regarding the back pressures that should be carried even under average conditions, the pressures and temperatures given in Table 1 will be found to be fairly accurate.

TABLE 1. BACK PRESSURES AND TEMPERATURES.

Temperature of room, degrees Fahrenheit	5	10	15	20	28	32	36	40	50	60
Back pressure, pounds gage	7	10	12	15	22	25	27	30	35	40
Temperature of ammonia, degrees Fahrenheit	13	10	5	0	8	12	4	17	22	26

In general the engineer should endeavor to so manipulate his expansion valves as to carry the highest back pressure possible and still produce sufficient refrigeration in his coldest coolers. A second limit to possibilities in this direction is reached when no more ammonia feed can be put on the expansion coils without causing liquid ammonia to return to the compressor, causing it to pound and the rod stuffing boxes to leak. It may be mentioned incidentally that the back pressure can be carried materially higher, and the efficiency of the plant materially increased by keeping the expansion coils free from the insulating effect of ice on the outside and oil on the inside. The coating of ice should be kept as thin as possible at all times and opportunity should be taken to remove oil from the expansion coils once in every two or three seasons at most, and oftener if much oil is thought to have worked into the system.

When the temperatures get too low, it is better to slow down the machine rather than to reduce the refrigerating effect of the system by closing down expansion valves, reducing the back pressure and literally throwing away power because the refrigeration is not needed. The full importance of this truth is seldom recognized by either the supervising or operating engineer, and it is rarely that either strives for the last pound of increased back pressure half as diligently as for the last inch of vacuum in the steam condensers, although the former pressure is of far more importance than the latter in its effect on the general efficiency of the plant.

CONDENSER PRESSURE

Just as the back pressures have to be reduced in order to reduce the boiling point of the refrigerating medium when low temperatures are required, the condenser pressure always rises sufficiently to raise the boiling point of the medium when the temperature of the cooling water is raised. Table 2 gives the approximate condenser pressures that should result from the use of different quantities of cooling water of different temperatures on condensers of average proportions.

In every event the condenser pressure should be kept as low as possible and the back pressure as high as the temperatures to be produced will permit, narrow limits between such pressures being as important to the efficiency of a refrigerating system as wide ones are to that of a steam engine, in which the economy increases with the range between boiler pressure and condenser pressure.

TABLE 2. CONDENSER PRESSURES AND TEMPERATURES.

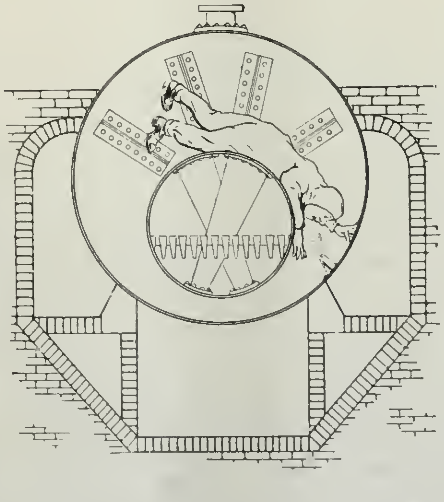
1 gallon per minute per ton per 24 hours—							
Temperature of cooling water, degrees Fahrenheit	60	65	70	75	80	85	90
Condenser pressure, pounds gage	183	200	220	235	255	280	300
Temperature of condensed liquid ammonia, degrees Fahrenheit	95	100	105	110	115	120	125
2 gallons per minute per ton per 24 hours—							
Condenser pressure, pounds gage	130	153	168	183	200	220	235
Temperature of condensed liquid ammonia, degrees Fahrenheit	77	85	90	93	100	105	110
3 gallons per minute per ton per 24 hours—							
Condenser pressure, pounds gage	125	140	155	170	185	200	215
Temperature of condensed liquid ammonia, degrees Fahrenheit	75	85	90	93	95	100	105

Remarkable Presence of Mind

By G. James Wells

Mr. Clark was a boiler expert, and what he did not know about a steam boiler was not worth anyone's attention. His favorite type was that known as the "Lancashire," and if an acquaintance wished to annoy him he could easily do so by suggesting that any water-tube boiler was superior to the Lancashire type.

Now, Mr. Clark, undoubtedly, was competent to express an opinion upon the merits or demerits of a boiler of any type, for he had spent his whole life around boilers, beginning as a riveter and plater and later becoming a charge hand, and proving so reliable that he was frequently chosen for work away from the shop. When away, he was obliged to send daily reports to the office, and in this way he soon learned to express himself in fair language, and having studied the board of trade reports upon boiler accidents, he was made an inspector.



As may be imagined, he was now on the road to further advancement and upon leaving the shop became a recognized member of the office staff. So far he had had no serious setback, and the indications were that he would prove eminently successful in his new capacity as inspector. His thorough knowledge of the trade as an expert workman, coupled with his natural shrewdness in detecting signs of deterioration, made an ideal combination for an inspector.

A customer having some boilers of the Lancashire type, one of which an insurance company had reported upon as requiring repairs, appealed to Mr. Clark's firm for advice and an estimate for doing the necessary work. This led to his receiving instructions to make a thorough examination and report at once. Accordingly a start was made for the factory, and upon reaching the boiler room he made a superficial examination, externally, and requested that the flues be thoroughly cleaned. He also marked certain portions of the brickwork for removal which his experience had taught him might reveal wasting of the plates. While this was being done, he entered the boiler and began a systematic examination of every plate, rivet and joint. One of the effects of heat upon boiler plates is to expand them, the hottest portions expanding most; hence deformations are frequently found which are due to the unequal expansion of the shell and flues. Resulting strains are produced, which give rise to bending at the flanges where the end plates are attached to the shell and flues. This movement takes place when the temperature of the boiler is changing, and is termed "breathing," and in time the plates commence to crack. Water entering the crack may cause corrosion and assist in increasing the depth of the crack. For this defect Mr. Clark was naturally on the lookout, and in order to be certain that he had seen all round each seam of the flues examined, he marked with a piece of chalk the limits of his view from any position. Thus he viewed the under parts of the flue seams by wriggling about on his back beneath the flues, afterward completing his inspection by lying on the top of the flue and reaching over until he could see his mark.

The most convenient light to use on such occasions is the old -fashioned tallow candle, which easily supports itself by adhesion when squeezed against some rough surface. Now it is not always easy to determine whether an apparent crack is really a crack in the plate or only a crack in the deposit upon the plate, and due to a little uncertainty upon this point Mr. Clark allowed himself to slip down head first between the flue and shell, as far as possible, in order to see better. However, he did not notice the position of the candle until suddenly his nose was uncomfortably close to the flame. His position was such that he could not get either hand to shift the candle, for he needed both to support himself from going farther down; so he began shouting for assistance. He yelled loudly and continuously, but to make himself heard and understood from the interior of the boiler, when a candle was burning his nose was not easy. Finally one of the men who heard the calls for help put his head into the manhole opening and asked, "What do you want?"

Mr. Clark, who was at no time very patient, on this occasion lost all that he possessed, and replied in suitable language with a demand that the man pull him back as the candle was burning him. After an instant's reflection the young fellow instead of instantly dropping into the boiler to assist the man inside called out, "Can't you blow out the candle?" The candle was instantly blown out and the crisis ended, after which Mr. Clark began wriggling his way back to the top of the flues, meanwhile wondering why he had not thought of blowing out the candle in the first place, also how it happened that when shouting he had not put out the candle.

Unfortunately, his nose was badly burned and occasioned him a great deal of physical pain, with the additional annoyance of explaining to his friends the origin of the trouble. His friends, although quite sympathetic, could not hide their appreciation of the humorous side of the incident, so he decided to avoid publicity by retiring until his nose had recovered its normal appearance.

Theory of Chimney Draft

By E. J. Kunze

Expansion due to heat, the force of gravity and pressure due to weight of the atmosphere are factors which produce the phenomenon of natural draft.

The term draft is sometimes used to indicate the volume or weight of gases passing through a fire. It would be better, when speaking of draft, to refer to a difference in pressure between two different points in the passage of a gas; but it is more convenient to indicate the difference in pressure between the external air and the gases at the point of measurement and draft is, therefore, usually so considered.

There are at least three explanations given for the phenomenon of chimney draft. It has been said that it was caused by heat, by pressure and by gravitation. Of recent years the latter has been the one most generally accepted. Parsons[*] says, "The problem of chimney draft is really one of graviation and not one of thermodynamics." While it is true that the action of gravity plays a part in the establishment of draft, it is equally true that there can be no difference in the specific gravity of any flue gases due to heat unless thermodynamic actions take place. It will be seen, therefore, that each of the three above-mentioned agents plays its part.

The tendency of nature is always toward equalization, reducing the abnormal to the normal; this is true of temperatures, of pressures and of forces of any kind. In the case of draft, while temperature is indeed equalized, no current is set up directly by the interchange of heat. The hotter gases lose heat to the colder gases but this does not cause the fluid to move. In the case of hot gases, a very small amount of heat is transmitted by conduction, the major portion being transmitted by convection due to the movement of the gases. Heat transmitted by radiation cannot, directly, set up a current because heat is so transmitted through both a denser and a rarer medium, and even through a vacuum, although it may not be sensed unless some material substance is set in vibration. If this were not so the heat energy of the sun would have been spent long before it reached the earth.

Although a drawing action may be implied in the term draft, there can be no such action in dealing with air or any other gas. A pushing rather than a pulling action exists. The gas moves from a place of higher pressure to one of lower pressure.

According to the kinetic theory of heat, the movements of the molecules of a gas when heated become more violent; hence, due to the mutual blows imparted to one another, the molecules are more and more separated from each other and in this way the density in different parts is changed and the specific gravity is lessened. The vibrations of the molecules do not cause the molecules set in vibration to be transferred from one position to another, the molecule is merely more or less agitated without causing it to lose its approximate position and the vibration, not the molecule, is transmitted along the medium by the action of the heat. But these vibrations cause the molecules affected to be repulsed by each other and hence a decrease in density occurs. The inequality thus set up causes a flow of the liquid or gas from the denser to the rarer portion and the denser gas pushes the rarer gas along until the latter reaches a place where the pressure is equal to its own.

In the case of chimney draft, the gases inside the stack are expanded due to the heat. Hot air weighs less than cooler air under the same pressure. The column of air inside the chimney weighs less, therefore, than an equal column of outside air. The pressure of the gas at the base of the stack will be less than the pressure of the outside air at that point. Suppose the gas were to expand fully before reaching the top of the stack, then the pressure of the gas would be equal to that of the air at the top of the stack. If the lower part of the chimney were to have communication with the outside air at that point, the heated gas would rise because of the difference in pressure of the air at the two ends of the chimney. This follows the law of gravity and for this reason, draft in a vertical flue is better than it would be under otherwise like conditions in a horizontal flue. The tendency is for the pressure to equalize, and it may seem most natural for the pressure of the outside air to be communicated to the inside of the chimney through the top but this is not the case.

The expansion of a columm of gas or air due to heat must take place in the direction of least resistance which, in the case of flue gases is, obviously, toward the top of the chimney. The pressure of the gas at the top of the chimney will be less than the pressure of the outside air at that point; first, because the gases have not been fully expanded, as a rule, by the time they have reached the top; secondly, the momentum of the moving gases increases the pressure in the direction of motion. The air, therefore, could not enter at the top. Hence, in order to tend toward equalization of pressure at the base of the chimney, the air must enter through the grate and cause the phenomenon of draft. Of course, the pressure at the base of the stack is not exactly that of the outside air at that point because of the loss in passing through the bed of fuel and ash. The higher the chimney, other things being equal, the higher will be the velocity of the moving gases and the lower will be the air pressure at the top, hence, the greater the draft.

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[*] "Steam Boilers," page 125.

The explanation of artificial draft follows a similar process of reasoning: With induced draft the pressure between the fire and the stack is reduced by means of a fan or jet of steam, forming a partial vacuum, at the same time a current is set up toward the top of the chimney, the gases being thus forced out by pressure. The air, therefore, must, as before, enter through the fire. In the case of forced draft, the air is forced in through the fire by means of a blower and in this case the combination of pressure and flow readily follows the explanation of draft given.

Therefore, draft is due to difference in pressure but, in the case of natural draft, this difference in pressure is produced by the expansion of the heated gases, the ultimate direction of the motion being vertical because of the action of gravity, the rarer gas being forced up by the denser gas (air) below it.

Engines with a Minimum of Clearance

With the valves set at an angle to the axis of the cylinder similar to air-compressor construction, with the ends of the cylinder rounded off and the piston made to conform, a clearance of 1-2 of 1 per cent. is made possible.

In our issue of August 30 we referred to an engine at the Brussels Exposition which has 0.75 of 1 per cent. clearance. We are able with the accompanying engravings to show how this is accomplished. The valves instead of being put in at right angles to the axis of the cylinder are set at an angle of about 30 degrees, as are the valves of the Linde and other air compressors where the suppression of the clearance is of so much moment. The end of the cylinder is either rounded or flattened off to meet this conformation, as shown in the engraving, so that the clearance volume is reduced to the very small space between the surfaces of the cylinder and the piston and the volume of the port *A*, Fig. 1. amounting to no more than 0.75 of 1 per cent. in the smaller engines and to ½ of 1 per cent. in the larger.

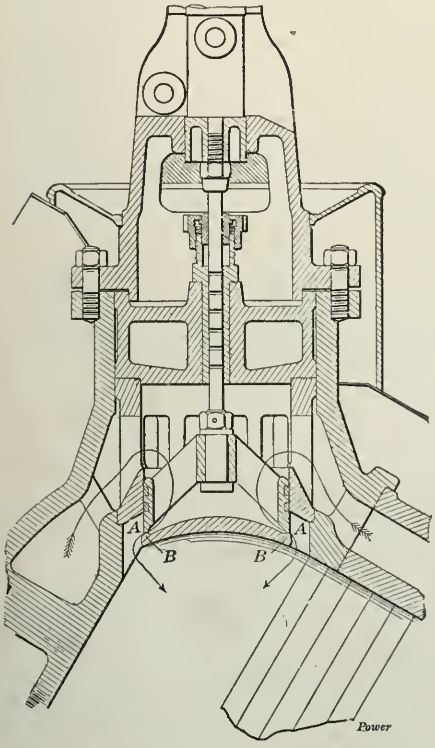


Fig. 1. Single-seat Valve of Walschaerts' "Giant" Engine

The admission valve is formed of a hollow cylinder with packing rings such as are used for certain piston valves. It does not close the port by passing the edge of it as does a piston valve, however, but seats positively with its lower edge at *B B*, Fig. 1, the steam following the course indicated by the arrows. The covering portion, not having to uncover an admission port, may have a large bearing surface with generously proportioned rings, whereas in the piston valve the parts upon which tightness depends are reduced and limited by the conditions required for opening and closing the port. The rings have a single joint and are not exposed to the irregular wear caused by passing over the bridges or girders of the ports. The difficulties and disadvantages of trying to make two seats simultaneously upon the double poppet are avoided.

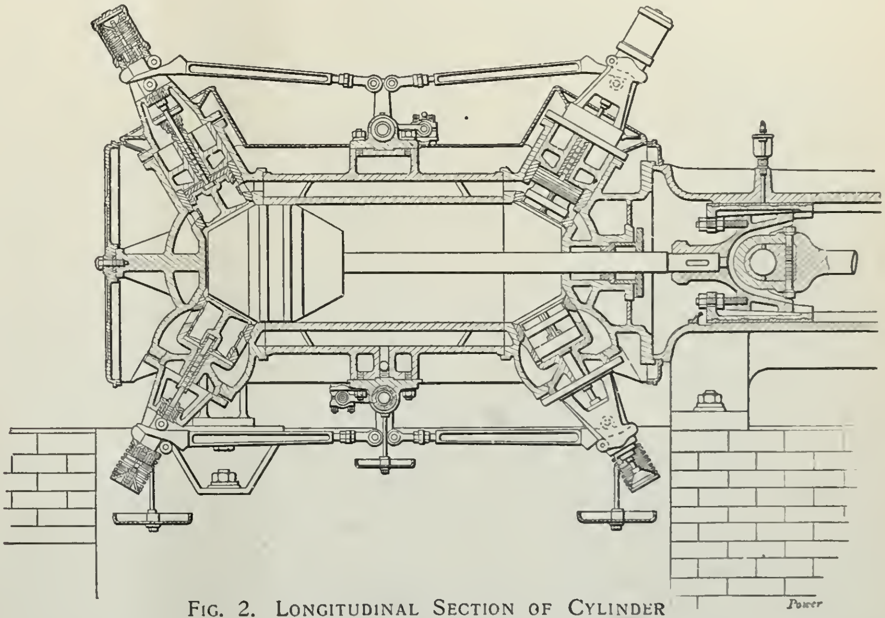


Fig. 2. Longitudinal Section of Cylinder

The engine is made in this form by Ernest Mennig, successor to Walschaerts, the inventor of the famous valve gear of that name, at St. Gilles, Brussels. He has used single-seated valves for a dozen of years successfully, placing them horizontally above and below the cylinder, but wishing to place them in the cylinder heads was unable with the horizontal arrangement to obtain sufficient area without the use of more than one valve for each position. It was to overcome this difficulty that they were placed in the oblique position shown in the engravings. The valves are well guided, and there appears to be no reason why they should not work as well in this as in either the horizontal or vertical position.

Although the surface initially exposed to the steam is somewhat increased by the hemispherical or conical shape of the piston head and surrounding wall, these surfaces are rendered as harmless as possible by being highly polished and steam jacketed. The admission is controlled by a governor placed on the same shaft as the eccentric and is effected by the simple cam levers shown in the longitudinal view of the cylinder.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

The Parallel Operation of Compound-wound Alternators

By R. L. Mossman

Compound-wound alternators operated in parallel generally cause more trouble with cross currents than machines that are separately excited only. In one case that I know of this difficulty was so serious that the compound windings had to be cut out of circuit entirely. Two compound-wound alternators were driven by engines having shaft governors that could not be adjusted while the engines were running. The generators were 150-kilowatt revolving armature machines, giving two-phase currents at 2200 volts, and were direct connected to their respective engines. The machines could be put in parallel without any trouble, but when the load came on the cross currents would build up so that one machine would have to be pulled out, and as there were times when one could not carry the load, it was decided to change the winding and make the machines separately excited.

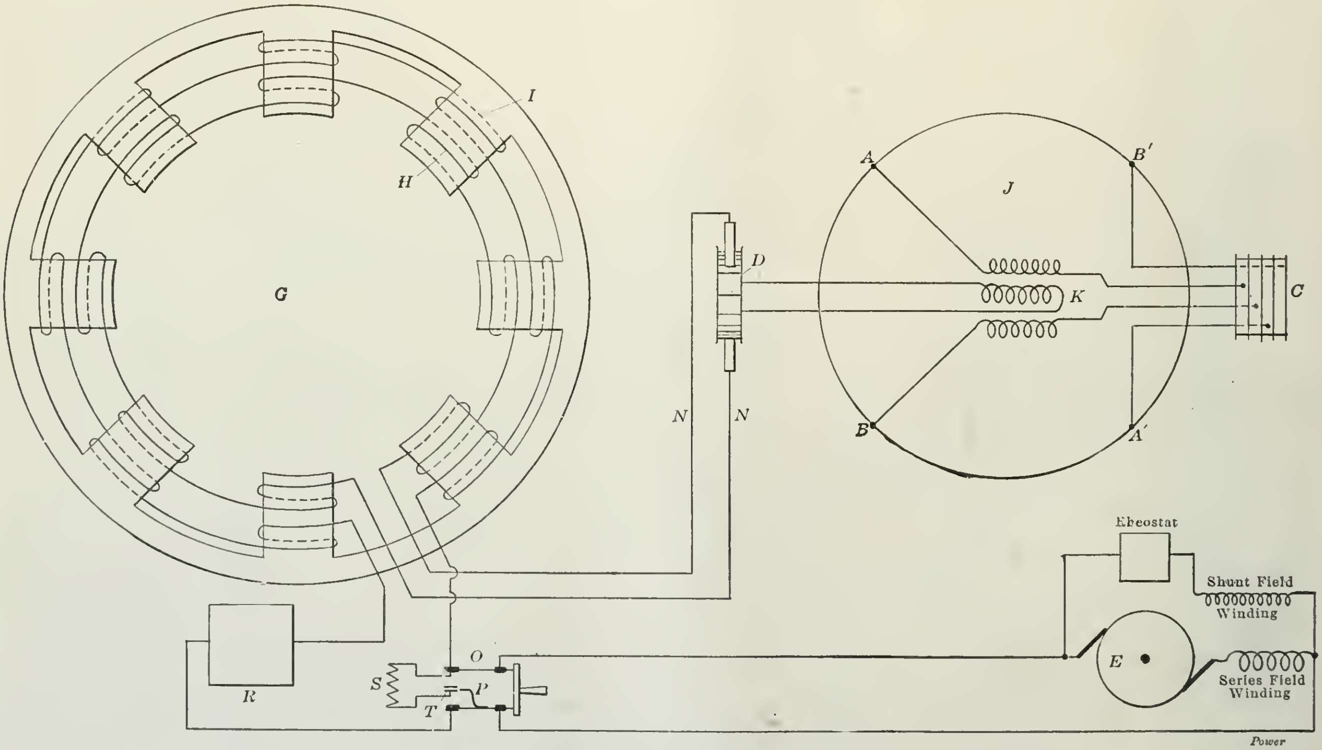


Fig. 1. Diagram of Circuits and Connections of Compounded Alternator

For the benefit of engineers who have not had experience with this type of machine, Fig. 1 is shown; this is a diagram of the circuits of a compound-wound two-phase alternator. The generator field magnet is shown at *G*; *H* represents the auxiliary field winding and *I* the main field winding supplied by the exciter. The armature is shown separately at *J* to make the diagram clearer; *A*, *A'*, *B* and *B'* are the terminals of the two armature circuits, carried to the collector rings *C* and a transformer *K* which is mounted in the armature spider and revolves with it. The secondary winding of the transformer is connected to a two-part rectifying commutator *D*, which changes the alternating current into pulsating direct current in the circuit *N N*. The primary windings of the transformer are in series with the leads *A* and *B*, which are ultimately connected to their respective collector rings at *C*.

The leads *N N* from the brushes of the rectifying commutator *D* connect the auxiliary field winding to the rectifier. The main field winding *I* is connected through the generator field rheostat *R* to the field switch *O*, and thence to the exciter *E*. The type of field switch shown is generally used to protect the field winding when the switch is opened. When a field circuit is suddenly opened there is a momentary inductive kick from the field coils similar to that of an induction coil, the voltage of which is sometimes high enough to puncture the insulation of the field winding. With the kind of switch here shown, when it is opened the winding is short-circuited by the auxiliary blade *P* entering the clips *T*, which are connected to one main blade of the switch through a resistance *S*, thereby preventing the sudden rise of voltage in the field coils.

The compounding effect of the auxiliary field winding in this machine is the same as that of the series winding of a compound-wound direct-current machine. When the load increases, the increased flow of current through the primary windings of the transformer *K* causes an increased flow in the secondary winding and, through the commutator *D*, in the direct current which flows through the leads *N N* to the auxiliary winding, increasing the strength of the field magnet and holding up the voltage. When the load is reduced the opposite effect is produced.

As the exciters were loaded only to about 60 or 70 per cent. of their capacity, the brushes on the commutator *D* were short-circuited, as shown at *W*, Fig. 2, thereby cutting out the auxiliary series field winding and leaving the field magnets to be magnetized only by the current from the exciters. The machines were run in parallel for several days in this condition, in order to note if any cross currents developed. As they operated without any difficulty in this condition the series field winding *H* was disconnected from the commutator *D* and connected in series with the main winding *I*, as shown in Fig. 3. The leads *N N* from the brushes were removed, also the brushes, brush-holders and "jumper" *W*, Fig. 3. This made it necessary to short-circuit the segments of the commutator *D*, as shown at *M*, in order to short-circuit the transformer *K* and prevent it from opposing the flow of current in the armature leads *A* and *B*. After this change was made no cross current showed up except such as was caused by poor regulation by the engine governors.

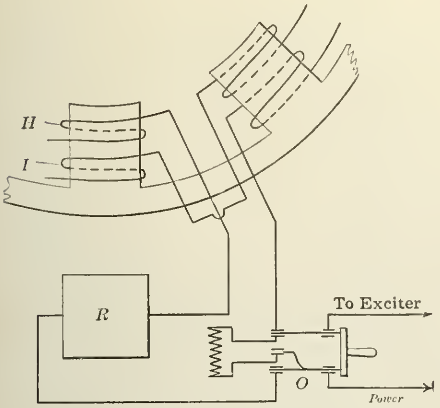


FIG. 2. FIELD WINDINGS COMBINED

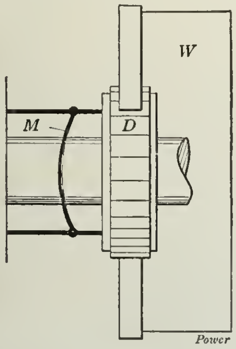


FIG. 3. SHORT-CIRCUITED RECTIFIER

In connecting the two field windings in series it was necessary to make sure that the windings would not oppose each other in their magnetizing effect. Of course, the auxiliary winding could have been left out of circuit entirely, but by connecting it in series with the main winding its magnetizing effect was utilized and helped out the exciters.

Care and Maintenance of Induction Motors

(Concluded)

By R. H. Fenkhausen

Controllers, autostarters, oil switches and circuit-breakers require little attention beyond an occasional adjustment of the fingers and contact clips which can be attended to by the motor inspector on his weekly visits. There is, however, a tendency among machine operators to tinker with the controlling apparatus whenever the motor does not run to suit them, and this tendency if not checked will result in excessive maintenance expense. Fig. 10 shows a simple device by which a controller or starter front may be locked to prevent tampering by unauthorized persons. It consists merely of a strap-iron bracket inside the cover, fitted with a lock staple protruding through a slot in the cover. The adjusting screw on a circuit-breaker may be locked in a somewhat similar manner. Fig. 11 shows the method used on the I-T-E circuit-breaker which may be adapted to other makes by exercising a little ingenuity.

A stock of repair parts for all motors and controllers should be kept on hand, as the time lost in waiting for a new part to be made will mean a loss of many times the cost of the part, and in addition it will be found that small repair parts made in the shop cost from ten to twenty times the price asked by the manufacturer, who can usually, by one stroke of a special die, form a repair part which even a skilled mechanic would require several hours to make by hand.

The following spares are recommended for standard motors: Twelve stator coils of each type used, 2 spare bearing sleeves of each size, fish paper and oiled linen for coils.

Wound-rotor induction motors require in addition: One set of gauze insertion carbon brushes of each size, 1 set of tension springs for brush-holders, 100 pigtails for brushes.

For autostarters and controllers: Ten gallons of transformer oil; 1 set of drum segments, 12 contact fingers, for drum type; 1 set of blades, 1 set of contact clips, for switch type; 1 operating handle for each type.

Spare autotransformer coils need not be carried because there is usually a spare starter available and it can be substituted in less time than would be required to change the coils, especially if connectors are used as recommended in a former article.

Almost all large plants have spare motors for special purposes and to insure against delay from breakdowns, and as these motors receive hard usage and are frequently moved, they are more liable to damage than motors in regular service. The principal source of trouble comes from neglect to empty the oil wells before moving the motor; as the motors are seldom kept level in transit, the oil is spilled all over the windings. Instead of removing the drain plug and drawing the oil off into an old can, where it will gather dust, as is the usual practice, the devices shown in Figs. 12 and 13 may be used and will save time and oil. The drain plugs on all spare motors and portable sets are removed and pet cocks substituted. The male end of a $\frac{3}{8}$-inch ground union is screwed on to each pet cock. The syringe shown in Fig. 13 may be made up from standard fittings and a cup leather fitted to the piston; the female end and union nut of a $\frac{3}{8}$-inch ground are attached to the nozzle of the syringe. As many more male ends are required than female, we made up ours in the shop to avoid breaking up complete unions. Before removing a motor a syringe is attached to the pet cock of each bearing, the cocks opened and the oil drawn out of the wells. When the motor is set at its new location the oil is forced back into the wells by means of the syringes, the cocks closed and the syringes uncoupled. Portable blower and

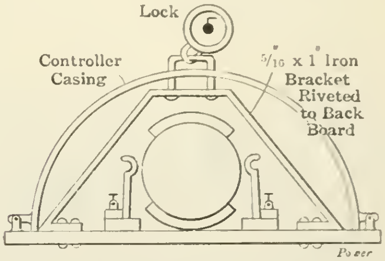


FIG. 10. CONTROLLER-COVER LOCK

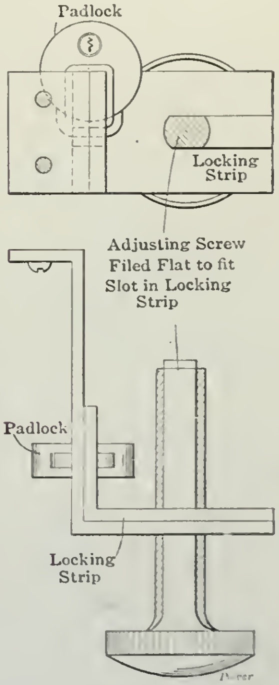


FIG. 11. LOCKING ADJUSTING SCREW ON CIRCUIT-BREAKER

pumping sets for use aboard ship, where they are handled many times a day and hoisted from deck to deck, have syringes permanently attached to each motor.

In addition to the weekly cleaning, motors should be blown out with an air hose every sixty days. To enable the air to dislodge all greasy dirt a special nozzle similar to Fig. 14 should be used; this increases the force of the blast, and the shape of the orifice gives a thin, flat jet that literally "slices" off the grease and dirt and does particularly effective work in the airgap and between the coils and end connections.

When changing motors from a floor to either wall or ceiling suspension, it is, of course, necessary to shift the heads through 90 degrees or 180 degrees in order to bring the oil wells into the proper position. When the heads are of the pattern shown in Fig. 2, with oil-well covers embracing an arc of 90 degrees from the vertical center line, there is danger of a careless employee replacing the heads with the oil wells on the side instead of on the bottom. On the smaller sizes of motors up to 5 horsepower many manufacturers make the heads perfectly symmetrical and the oil-well cover is the only indication of top and bottom. A little consideration will show that displacement of the head through an angle of 90 degrees will leave the cover in the same relative position, although on the other side of the center line. The only sure way is to look at the oil rings.

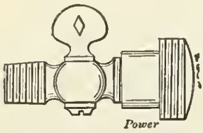


FIG. 12.

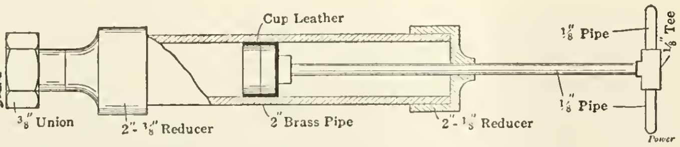


FIG. 13. PUMP FOR REMOVING OIL

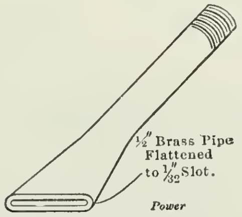


FIG. 14. A GOOD AIR NOZZLE

Most motors of 5 horsepower and under have heads bolted on with small hexagon-head cap screws often located in such close quarters that it is difficult to insert or remove them without using a socket wrench. As wrenches of this type are not apt to be at hand when needed in a hurry, a great saving of time and patience will result if the heads of the cap screws be slotted with a hacksaw. The cap screws may then be driven home with great ease by means of a brace and a screwdriver bit.

A Simple Solution of a Complex Resistance Problem

BY J. M. ROW

The accompanying problem in divided circuits is generally considered to be very difficult to solve without algebra, but I have found that it may be worked out by the use of nothing higher than simple arithmetic.

In the diagram *A*—*B*, *A*—*C* *B*—*C*, *B*—*D* and *C*—*D* represent five wires having a resistance of 4, 11, 3, 7 and 2 ohms, respectively. It is desired to find the joint resistance from *A* to *D*.

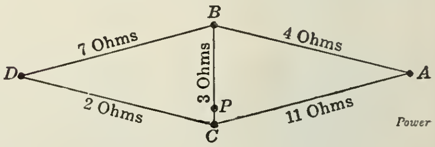
The first thing is to determine whether or not any current will find its way through *B*—*C* when passing from *A* to *D*; and, if so, to determine in which direction it will flow. This may be ascertained by a simple statement of proportion thus: if $\frac{4}{11} = \frac{4+7}{11+2}$ then no current will flow through *B C* and the solution is very simple, being the same as though the wire from *B* to *C* had been removed. But if $\frac{4}{11}$ is greater than $\frac{4+7}{11+2}$ the current will flow from *C* to *B*, while if it is less the current will flow from *B* to *C*. This is all based on the assumption that *A* is connected to the positive terminal of the source of power. It is evident in the case assumed that current will flow in the cross wire from *B* to *C*.

Now assume a potential from *A* to *D* sufficient to cause one ampere to flow from *B* to *C*. It is evident, then, that one ampere comes from *A* to *B*, then goes to *C* and thence to *D*. Of course, more current will flow from *A* to *D* than this one ampere, but the remainder of the current may be neglected for the present. Since one ampere is flowing from *B* to *C* it follows that the potential at *B* is 3 volts higher than at *C*. Now suppose this wire to be cut at some point, as at *P*. This will interrupt the flow of that one ampere from *A* to *B*, to *C* and to *D*. Now that one ampere from *A* to *B* caused a drop of 4 volts which is now available for sending current from *A* past *B* to *D*, and it will produce $\frac{4}{11}$ amperes. It is evident that we have lost one ampere and gained $\frac{4}{11}$ ampere; the net result is a loss of $\frac{7}{11}$ ampere in that branch. This decreases the drop from *A* to *B* by $\frac{4 \times 7}{11} = \frac{28}{11}$ volts. That is to say, the potential at *B* is $\frac{28}{11}$ volts higher than when the circuit is closed from *B* to *C*.

In the lower branch, when the one ampere was interrupted which flowed from *C* to *D*, it decreased the drop by 2 volts, which became available for sending current from *A* through *C* to *D*, and will produce $\frac{2}{13}$ ampere; this, flowing from *A* to *C* will increase the drop in that branch by $\frac{22}{13}$ volts. This leaves the potential of *C* $\frac{22}{13}$ volts lower than when the circuit was closed at *P*. Now *B* was 3 volts above *C* and increased $\frac{28}{11}$ volts, while *C* decreased $\frac{22}{13}$ volts. This makes a potential difference across *B C* of $3 + \frac{28}{11} + \frac{22}{13} = \frac{1035}{143}$ volts when *B C* is cut and the voltage from *A* to *D* is just sufficient to cause one ampere to flow from *B* to *C* if the circuit were closed.

With a potential of one volt from *A* to *D* and the circuit open at *P*, the drop from *A* to *B* is $\frac{4}{11}$ volt, while from *A* to *C* it is $\frac{11}{13}$ volt. The potential difference from *B* to *C*, then, is $\frac{11}{13} - \frac{4}{11} = \frac{79}{143}$ volt when the potential form *A* to *D* is one volt; while with some unknown voltage from *A* to *D* which was sufficient to cause one ampere to flow from *B* to *C* a potential difference of $\frac{1035}{143}$ volts was maintained from *B* to *C*. This unknown potential may be found now by dividing $\frac{1035}{143}$ by $\frac{79}{143}$, and it equals $\frac{1035}{79}$.

With the wire *B*—*C* still open the current through the upper branch of the diamond is $\frac{1035}{79} \div 11 = \frac{1035}{869}$ amperes and that through the lower branch is $\frac{1035}{79} \div 13 = \frac{1035}{1027}$ amperes. It was previously ascertained that opening the cross wire decreased the current from *A* to *B* by $\frac{7}{11}$ ampere and increased the current from *A* to *C* by $\frac{2}{13}$ ampere. Then with the circuit closed from *B* to *C* the current from *A* to *B* is $\frac{1035}{869} + \frac{7}{11} = \frac{1588}{869}$ amperes; while from *A* to *C* it is $\frac{1035}{1027} - \frac{2}{13} = \frac{877}{1027}$ ampere. The total current flowing from *A* to *D* is $\frac{1588}{869} + \frac{877}{1027} = \frac{2{,}400{,}389}{892{,}463}$ amperes, and as this is produced by a voltage of $\frac{1035}{79}$ the joint resistance from *A* to *D* must be $\frac{1035}{79} \div \frac{2{,}400{,}389}{892{,}463} = 4.889$ ohms.



PROBLEM IN JOINT CIRCUITS

Gas Power Department

Modification of the Humphrey Pump*

By Herbert A. Humphrey

This paper deals with the subject of Humphrey pumps from the broad point of view, omitting all details as to types of valves and valve gear except in one instance just to illustrate the kind of device which has been found suitable. Also, to make the paper more complete in itself, some of the old ground is covered again.

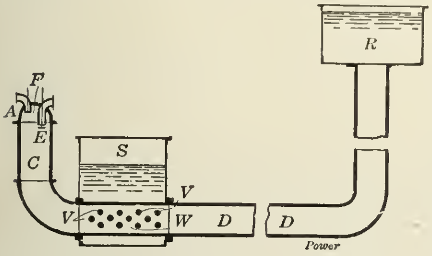


Fig. 1. The Simple Pump

Take the simplest case of Humphrey pump, as illustrated in Fig. 1. Water has to be raised from a supply tank *S* to an elevated tank *R*. The water-valve chamber *W* is equipped with valves *V* opening inward. In the top of the combustion chamber *C* are an inlet valve *A* and an exhaust valve *E*. A simple interlocking gear is arranged between these two valves so that when the valve *A* opens and closes it locks itself shut and releases the valve *E*, and when the valve *E* opens and closes it locks itself shut and releases the valve *A*. Consequently, when suction occurs in the chamber these valves open in turn.

Imagine a charge of gas and air to be compressed in the top of the chamber *C*, and fired by a spark plug which projects through the top casting. All the valves are shut when the explosion occurs, and the increase in pressure drives the water downward in the pump and sets the whole column of water in the discharge pipe *D* in motion. The column of water attains kinetic energy while work is being done upon it by the expanding gases, so that when these gases reach atmospheric pressure the column of water may be moving with considerable velocity. The motion of this column of water cannot be suddenly arrested, hence the pressure in *C* falls below that of the atmosphere, the exhaust valve *E* opens and also the water valves *V*. Water rushes in through the water valves mostly to follow the moving column in the pipe *D*, but partly to rise in the chamber *C* in an effort to reach the same level inside the chamber as exists in the suction tank.

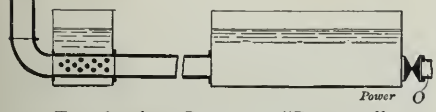


Fig. 2. Air Cushion "Return"

When the kinetic energy of the moving column has expended itself by forcing water into the elevated tank, it comes to rest, and there being nothing to prevent a return flow, the column starts to move back toward the pump and gains velocity until the water reaches the level of the exhaust valve, which it shuts by impact.

> **Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.**

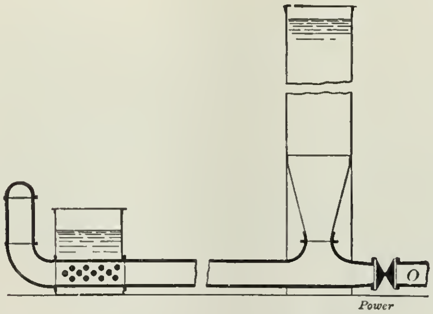


Fig. 3. Standpipe Return

A certain quantity of burnt products is now imprisoned in the cushion space *F*, and the energy of the moving column is expended in compressing this gas cushion to a greater pressure than that due to the static head of the water in the tank *R*. Hence a second outward movement of the column results, and when the water reaches the level of the valve *E* the pressure in the space *F* is again atmospheric, and further movement of the water opens the valve *A* against a light spring and draws in a fresh charge of gas and air. If there were no friction the water would fall to the same level as that from which the last upward motion started, but the amount of combustible charge drawn in is slightly less than this movement would represent. Once more the column of water returns under the elevated tank pressure and compresses the charge of gas and air, which is then ignited to start a fresh cycle of operations.

The action of the pump is not altered if, instead of delivering into an elevated tank, it discharges into an air vessel, as shown in Fig. 2, or into an open-top stand pipe or tower, as shown in Fig. 3, and both of these arrangements are useful if a continuous flow from the outlet *O* is desired. In many cases the pump can be placed directly in a concrete pit which serves also as a foundation, and the discharge can be by an inclined pipe, as shown in Fig. 4, where the water-valve box is shown vertical.

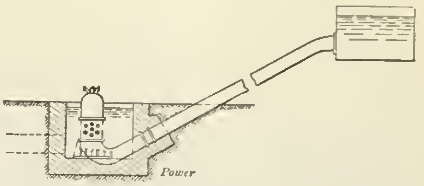


Fig. 4. Vertical Water-valve Box

In the simple form of pump the degree of compression of the combustible charge prior to ignition depends on the hight to which the water is raised, and exceeds the static equivalent of the head. This can be made clear by reference to Fig. 5, where water returning under a pressure *A B*, and moving through a stroke *A D*, does work in compressing the gas equal to *A B C D A*. The gas is compressed from *A* to *E*, and the work of compression is *A E D A;* consequently the shaded area above the line *C B* must be equal to the shaded area below it, and *E*, the maximum compression pressure, is much above *C*. This result is obvious if one remembers that the kinetic energy acquired by the liquid column on its return flow is utilized in compressing the combustible gas while the compression brings the column to rest.

The same considerations enter into the question of the cushion pressure attained, but here we are dealing with the compression of a volume of gaseous fluid which occupies the clearance space only, and the stroke of the water column is greater in proportion, because for the first part of the stroke exhaust products are being expelled and no compression occurs. Without going into the mathematics of the subject, it is sufficient to say that

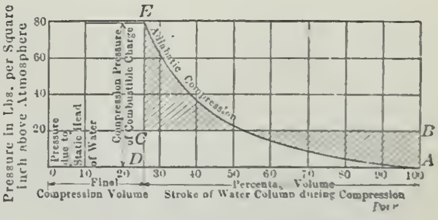


Fig. 5. Work Diagram

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*Abstract from a paper read before the Manchester (Eng.) Association of Engineers.

the cushion pressure rises rapidly as the hight to which the water is lifted increases, and at the maximum lift of about 40 feet to which the simplest type of pump is limited, it may exceed the explosion pressure when using producer gas, or may approximately equal the explosion pressure when working with town gas or gasolene.

No simple means of indicating the work done by the expanding gases has been found, so that the indicated horsepower of a Humphrey pump cannot be directly ascertained, but indicator diagrams showing pressures on a time basis are interesting. Such an indicator diagram is shown in Fig. 6; this was taken from a 45 water horsepower four-stroke pump, lifting to 39.4-foot head and working on town gas. The figures relating to this diagram are:

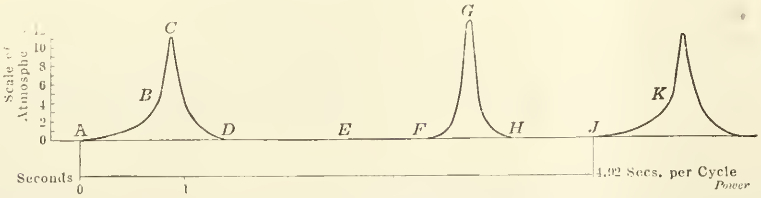


Fig. 6. Time-pressure Diagram

Compression pressure at ignition, 56.5 pounds per square inch.

Explosion pressure, 163 pounds per square inch.

Cushion pressure, 185 pounds per square inch.

The pump was delivering 306 gallons per working stroke.

The diagram may be explained as follows:
Compression of charge from A to B.
Explosion from B to C.
Expansion from C to D.
Taking in water and scavenging air and partly exhausting, D to E.
Return stroke completing exhaust till valve shuts, E to F.
Cushioning, F to G.
Cushion expansion, G to H.
Intake of new charge, H to J.
Next compression of charge, J to K.

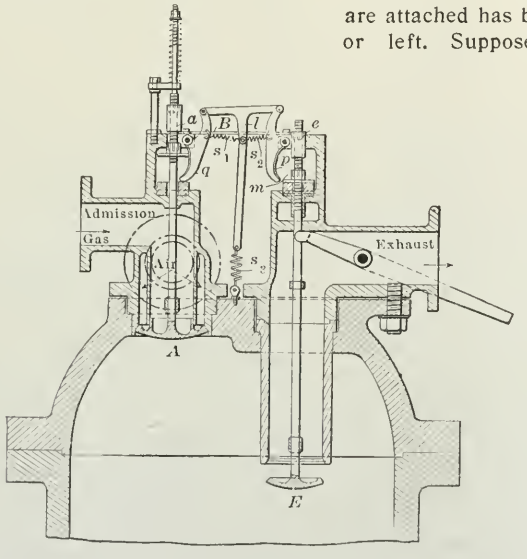


Fig. 7.

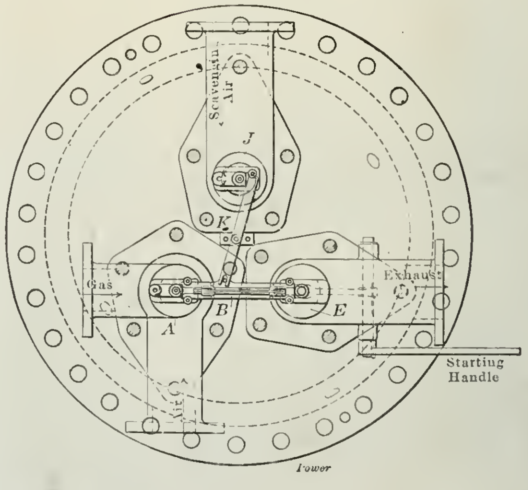


Fig. 8.

Valve Mechanism of the Standard Simple Pump

The stiff spring used does not permit the suction to be shown on this diagram, but the partial vacuum at the end of cushion expansion may momentarily be as much as four pounds per square inch, and the water suction may at first be equal to two or three feet of water, both dying down to atmospheric pressure during the respective strokes.

The author has dealt at some length with the four-stroke cycle pump, because its construction is simple, and it is the key to all the others. The four unequal strokes which result from the free oscillations of the water column give a cycle which is superior to the Otto cycle, and is in fact the theoretically perfect heat cycle described in textbooks.

Figs. 7 and 8 illustrate the valve gear of the pump which was shown in operation at the Brussels International Exhibition. The bolt *B* sliding horizontally must lock either the admission valve *A* or the exhaust valve *E* by engaging under one of the collars *a* and *e*, which are fixed on the stems of their respective valves. The bolt is urged right or left, according to whether the spring s_1 or s_2 is pulling the hardest, and this again depends on whether the link *l* to which the springs are attached has been shifted to the right or left. Suppose the exhaust valve opened last, then its washer *m* by pressing against the cam arm *p* will have moved the system *p l q* so that it leans to the right, in which position it is retained by the tension of the spring s_3. This puts tension on the spring s_1 and loosens the spring s_2; the bolt *B*, therefore, tries to move to the right, but until the exhaust valve shuts it can only press against the collar *e*. However, when the valve *E* comes back to its seat the bolt instantly locks under *e*, holding the valve *E* shut and releasing the valve *A*, so that the next time suction occurs in the combustion chamber, *A* only can open. Precisely the same kind of action occurs when *A* shuts; it becomes locked and *E* is released again. Thus the valves *A* and *E* are automatically allowed to act alternately, the difference between them being that *E* remains opens till shut by the rising water but *A* shuts under the action of its supporting spring as soon as the suction in the chamber permits the spring to lift the valve to its seat.

The scavenging valve *J*, shown in the plan of the combustion head, Fig. 8, operates at the end of each expansion stroke, its locking and release periods corresponding to those of the exhaust valve; they are made simultaneous by a lever pivoted at *K*, and operated by a pin on the bolt *B*. If the water could rush in fast enough when the pressure falls to atmospheric, there would be no scavenging action, but the incoming water has to be accelerated, and that gives rise to sufficient suction to effect the desired scavenging. In the exhaust outlet there is a light nonreturn valve, to prevent burnt products being drawn back into the chamber.

Starting the Pump

If a pump when working is stopped by switching off the ignition current it will continue its cycle up to the point of compressing the new charge, and then pull up, after a few oscillations of the water column, with the fresh charge ready for ignition, to again start the pump when the current is switched on. In starting up from "all cold" a single charge of air is introduced against the head of water by a hand pump or small compressor, and the exhaust valve is opened by a hand lever to allow the air to escape. This starts the oscillation of the column which automatically shuts the exhaust valve, cushions, draws in a new charge, compresses and fires it, and so the pump gets away under load.

HIGH COMPRESSION WITH LOW CUSHION PRESSURES

Without much sacrifice of simplicity the cushion space may be kept separate from the combustion chamber, and can then be made relatively large because the products or air in the cushion do not then mix with the new charge. This forms one way of keeping the cushion pressure low. If an internal bell be used to provide a separate cushion of burnt products, a slight modification of the gear is required to control the three valves. Instead of a scavenging valve, there would be a valve *J* opening into the top of the internal bell, as shown in Fig. 9, and on its spindle a loose nonreturn valve *Q*. The valve *J* is released to fall at the same time the admission valve opens, but the valve *Q* prevents any inlet into the top of the bell.

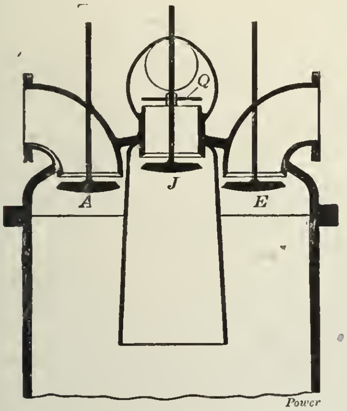


FIG. 9. HIGH-COMPRESSION PUMP CHAMBER

The action is as follows: When explosion occurs the bell is full of water and all valves are shut. Expansion below the open bottom of the bell allows the water to escape from the bell, which becomes filled with burnt products. On the first instroke of the water the exhaust valve *E* is open as usual and allows the products to escape, but the valve *J* is shut, so that the burnt products in the bell are trapped, and suffer compression by the rising water, which clears away all the other burnt products outside the bell. The compressed cushion in the bell now expands and the second outstroke occurs, during which the suction opens the admission valve *A* to let in the fresh charge as usual, and the valve *J* also opens.

The second outstroke, being shorter than the first, does not lower the water to the bottom of the bell; therefore the pure combustible charge outside the bell does not mix with the products inside. Lastly, the second instroke occurs, when the water rising in the bell forces the exhaust gases out past the nonreturn valve *Q* until the water shuts the valve *J*; the return column of water, having gained velocity depending on the volume of gases discharged from the bell, compresses the pure combustible charge outside the bell. Since the volume of the bell is not limited, the water column may attain a high velocity and so give a high compression pressure, insuring good efficiency.

New High Pressure Pumping Station

By A. D. BLAKE

The new high-pressure pumping station which has just been completed in connection with Philadelphia's high-pressure fire system, places that city in a position to successfully cope with most any exigency that might arise. The new station is located at Seventh and Lehigh streets and furnishes fire protection to the Kensington and Richmond manufacturing districts. The water supply is taken from the Fair Hill reservoir as the station is some distance from the river.

In many respects the new station is similar to the Delaware avenue station, which has been in successful operation since 1904, but contains many improvements over the earlier installation. Like the Delaware avenue station, gas engines are employed to drive the pumps, the gas supply being taken from the city mains. In choosing gas engines in preference to electric motors for driving the pumps it was claimed that the gas supply in Philadelphia is more dependable than the electric service and, moreover, with the former the city is obliged to pay for only the actual amount of gas that is consumed, whereas, if electricity had been selected, the city would have had to guarantee and pay for a definite minimum amount whether consumed or not.

Fig. 1 shows an exterior view of the station, the appearance of which reflects much credit upon the designers and forms an architectural addition to that section of the city.

The capacity of the plant is 12,000 gallons of water per minute delivered at a working pressure of 300 pounds per square inch to the high-pressure mains which are entirely separate from the city's low-pressure mains. The equipment consists of ten 300-horsepower, 18x22-inch, three-cylinder, four-stroke, Westinghouse, vertical gas engines running at 200 revolutions per minute, each connected through a single 5 to 1 reduction gear to an 11½x12-inch vertical, triplex, double-acting Deane pump. These units are arranged along the main floor in two rows, as shown in Fig. 2. There are no clutches between the engines and pumps, and the former are started by compressed air. The pumps are started under friction load, the water being bypassed to the suction; when the desired speed has been reached, the motor-driven bypass valve is automatically closed, and the pump is ready for service. A relief valve attached to each pump diverts the water back into the suction line when the pressure rises above 300 pounds.

The pumps are of massive construction and when running are unusually free from vibration and noise. The crank shafts are of a single piece of heavy forged steel with the cranks set at 120 degrees. The connecting rods are heavy steel forgings of the marine type with both ends adjustable. The valve chests and cylinders are cast separately so as to localize any damage that might occur and facilitate quick repairs. Both the suction and the discharge valves are located above the highest position of the piston, which insures the pump being primed at all times.

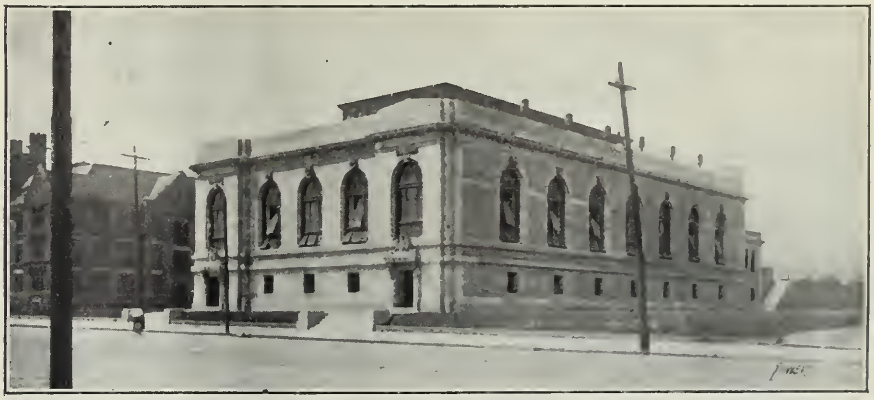


FIG. 1. EXTERIOR VIEW OF STATION

In addition to the ten main units there is a 125-horsepower gas engine of a design similar to the larger ones. This is belt connected to a 20-kilowatt, 110-volt, direct-current Westinghouse generator and to a 9½x5x10-inch Hall tandem air compressor. The generator supplies current for operating the valves on the discharge lines, for the ignition spark in the gas-engine cylinders and for station service. The compressor supplies air at 200 pounds pressure to ten cylindrical tanks, connected in parallel, which furnish compressed air for starting the gas engines. The admission of this air is controlled by a lever-operated valve located at each individual engine. Hence, in order to put a unit into service it is necessary for the operator to go over to that unit.

By an arrangement of cam-operated

valves the compressed air is admitted only to the cylinder whose piston happens to be up.

Precautions have been taken against failure of the compressed-air supply or of the direct current, due to a breakdown of the smaller gas engine, by the installation of a motor-generator set, running off the two-phase city-power circuit. This set supplies direct current in duplicate of the generator previously mentioned, and, in addition, the motor is belted to an air compressor which is an exact duplicate of the compressor belted to the gas engine. In order to carry the extra load of the compressor, the motor is rated at 50 horsepower, although the generator has a capacity of only 20 kilowatts.

To further insure ignition current, each unit is equipped with a ½-horsepower motor-generator set running off the city circuit, and a storage battery, the battery being charged by the small motor-generator set. This makes four sources of supply for ignition current: the generator driven by the gas engine, the large motor-generator set, the small motor-generator sets and the storage batteries, either one of which may be switched into circuit by a small panel-board located at each unit. Each gas-engine cylinder is equipped with duplicate spark plugs, either of which may be used; this obviates a cylinder being put out of service should the spark plug fail while the engine is running.

All the piping is located in the basement, which is light and affords plenty of head room. The water supply is brought from the reservoir in two 24-inch pipes, each of which supplies five of the main pumps. These pipes terminate in a U-connection into which is placed a valve; hence, should a break occur in one of these lines, it may be shut off by a valve located at its entrance to the building, and by opening the cross connection, all the pumps may be fed by the other line.

The discharge consists of two 20-inch branches, one running east and the other west from the station. As in the case of the suction lines each of these discharge lines is fed by five of the main pumps and by means of a cross connection and motor-driven valves the discharge from the whole ten pumps may be thrown into either main. Each discharge main is connected to an air chamber which takes up any shocks in the line. A bypass from each supply pipe to the corresponding discharge main, which is fitted with a check and gate valve, insures the high-pressure mains being full of water at all times. This prevents any undue shock which would result if the mains were empty and the water suddenly turned into them at a high pressure.

By means of indicating lamps on the switchboard the operator is able to tell when the motor-driven valve on each branch from a pump to the discharge main is wide open. If the lamp indicates that this valve is wide open and the pressure is not up to the required amount, then it is obvious that another unit must be thrown in.

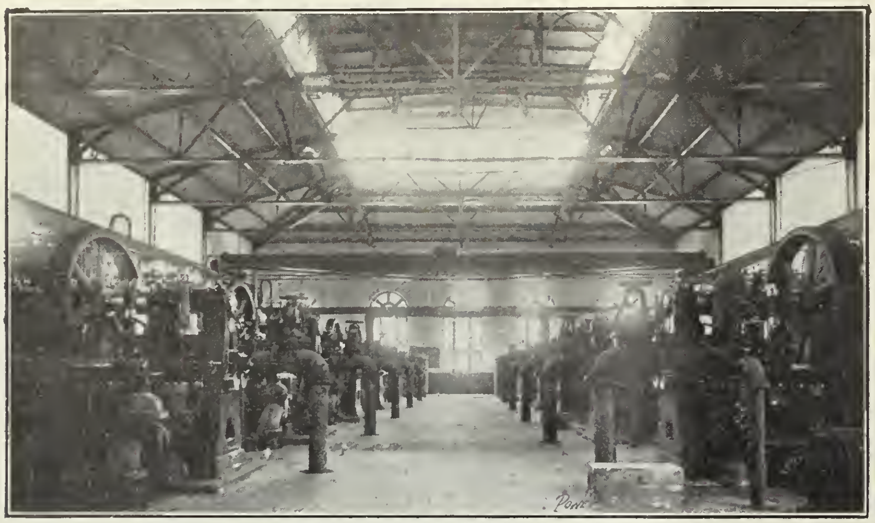


Fig. 2, Interior of Station Showing Main Units

The jacket water for the cylinders may be taken from the city mains, from the reservoir or from the high-pressure discharge mains. In the latter alternative the water passes through a reducing valve which reduces its pressure from 50 to 70 pounds. Each system of piping in the station is painted with a distinctive color so as to obviate any confusion.

By duplicating many of the more important pieces of apparatus and by providing various sources of supply for the electric current, the chances of the system being put out of service, through failure of any part of the equipment, has been reduced to a minimum. It has been demonstrated that full pressure can be placed upon the system within forty seconds from the time the alarm is received.

Explosion in a Blast Furnace Power Plant

By Osborn Monnett

An explosion of blast-furnace gas occurred about 11 a.m. on Sunday, November 13, at the Youngstown, Ohio, works of the Carnegie Steel Company, demolishing a gas holder said to have had a capacity of 45,000 cubic feet, partly destroying several mill buildings, tearing a hole in the tile roof of the blowing-engine house, and breaking windows in houses and business buildings for blocks around. A number of employees had narrow escapes from death and it is probably owing only to the explosion happening on Sunday that some fatalities were not recorded.

The gas, of approximately 80 B.t.u. per cubic foot, used at this plant for power purposes is taken from furnaces Nos. 5 and 6, each of 600 tons capacity, passed through the usual complement of washers to the engines. The blowing-engine equipment consists of four 3500-horsepower gas-driven units, and there are two gas-driven electric generators each of 2000 kilowatts capacity.

The gas was stored in a holder with a series of deflectors, for the purpose of mixing the gas and insuring a more nearly uniform quality at the engines.

For some time, owing to lack of work, it has been the practice to shut down furnaces Nos. 5 and 6 on Friday of each week, allowing them to stand with fires banked until Sunday, when they were started up again in time for the Sunday night shift. This experiment, it is understood, has been watched with interest by steel men, not all of them being agreed as to the feasibility of the plan.

Standing idle for 48 hours each week gave opportunity for air to percolate into the gas system in various ways, and it has been usual, upon resuming operations, to thoroughly bleed the mains before starting the gas engines, in order to purge the system of any air it might contain. On this occasion, however, it seems that this precaution was either forgotten or neglected and that, consequently, there was still a mixture in the mains capable of propagating a flame. The engines, it is reported, were just being started and only a few power strokes had occurred, when a roar was heard in the gas mains, being instantly transferred to the holder, which was located just outside the blowing-engine house.

The incident emphasizes the need for the greatest care in the operation of such installations and suggests the desirability of developing some form of flame retarder for the intake mains of large gas engines similar to those now being used in the transfer passages of small two-stroke-cycle gasolene engines.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

A Simple Filing Method

To the operating engineer in any plant the problem of taking care of catalogs and valuable printed matter is a big one. He is usually pretty tired when the time comes for him to wash his hands and sit down to a job of indexing, so the indexing is not done. To properly index fragmentary printed matter is a task which even the best equipped man looks at with awe and distaste; hence any system of filing for the man in the engine room must be one that requires no indexing; it must be like the open shelf in the modern library. A few years ago the reader who entered a library had to look through an immense book catalog or a card index in a room apart from the books, access to the books being denied to everyone except the employees. The present practice, however, in the greater number of libraries is to allow the readers to look over the shelves and select their own books. I have found the following method of keeping track of the printed matter, sent so freely by manufacturers, to be very successful.

Cloth- and leather-bound books are set on edge on a shelf in the engine room. The shelf should be kept for the books alone and if men are permitted to get into the habit of using it as a "catch all" for waste, gloves, etc., they soon get in the habit of caring nothing for the books. For printed matter not bound in cloth or heavy bindings, obtain a box 14 inches wide, 10 inches deep and of any length desired. In the bottom of the box bore ¼-inch holes at intervals of 1 inch the entire length on the center line. Out of heavy cardboard, or thin wood, cut a number of sheets 9 inches wide and 13 inches long, with a projection ½ inch high and 2 inches long on top arranged like the guide cards of the ordinary card-index box. Each projection should be past the one in front of it so that for the 14-inch length there will be seven sheets to fill the space. Take a piece of wood or cardboard 9 inches wide by 13 inches long and nail one of the long edges to a piece of board 13 inches long and 4 inches wide. At the end put triangular pieces as braces, so that when the large sheet is upright on the short piece it will stand and be able to resist some pressure. In the bottom piece put three pegs a little less than ¼ inch in diameter an inch apart so that they will fit into the holes in the bottom of the box; this is the "follower."

On the projections of the sheets paste small labels, and mark them, "engines," "boilers," "stokers," etc. Place the follower in place and put the guide sheets in front of it, as nearly in alphabetical order as possible. Place the catalogs in front of the sheet having the label indicating where the catalog belongs. This box will take a 10x12-inch catalog and all intermediate sizes down to 6x9 inches.

When a catalog contains matter on several subjects it can be separated into sections and each section filed where it belongs. When this is done a memorandum should be made on the facing sheets, telling where the other sections are placed. Another way is to have in each compartment a thin book containing a cross-reference index; for example, John Smith & Son issue a catalog dealing with engines, boilers and air compressors. The catalog is filed under "engines" and the word is written in red pencil on the cover so that it will be returned to the proper division if ever taken out of the box. In the divisions reserved respectively for boilers and air compressors will be written in the cross-reference books, "See catalog of John Smith & Son, filed with engines."

The engineer often reads "kinks" and interesting articles in the papers he receives. They are forgotten or cut out and lost because no systematic method of filing is used; for such matter the old-fashioned scrapbook idea is good but this should not be bulky and it is better to have several, one for each subject, than to have one. By removing the alternate leaves the clippings do not cause the books to bulge very much. File these books in the compartments in the same way the catalogs are filed. When the clippings occupy a full page or are partly on one page and partly on another they cannot be pasted but can be filed away in folders just as letters are filed in vertical filing cases.

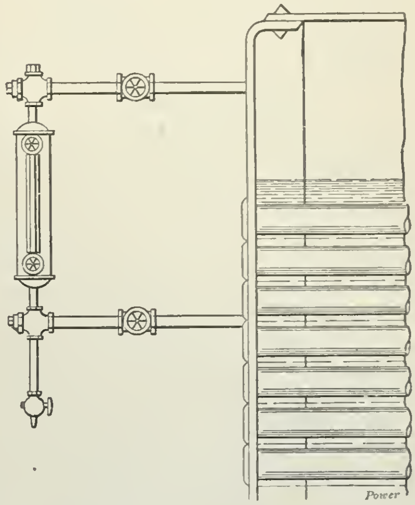
This filing system requires practically no work on the part of the engineer, for the largest part of the work is done when the box and its guides are made. The cover can be lifted and a catalog or clipping put in the box in a very short time and very few require any cross references.

ERNEST MCCULLOUGH.

Chicago, Ill.

Water Gages

There is no appliance used in connection with boilers that needs more attention than the water column. It is true that the safety valve is of great importance in preventing overpressure; but, if a water column, which is depended upon to show a correct and safe level of water in the boiler, fails and misleads, the situation becomes very dangerous. Some columns are not placed upon the boilers so that they will show the correct water level, and this is really more dangerous than a column that is out of repair, as in the former case the water tender is depending upon a certain water level and may be running much below it.



VALVES PLACED IN CONNECTION TO WATER COLUMN

It is not an uncommon thing to find a set of columns placed upon a boiler without cutoff valves between the boiler and the columns. Some engineers claim that no valve should be placed between the boiler and the column, and this theory has some merit, but it seems to be an unsettled question. The only point that can be urged against it is the fact that the valve could drop upon the seat and close off communication with the steam or water space, but this can happen only when the valve stem is in a vertical position, not when in a horizontal position. There is little chance of the valve becoming detached from the stem and closing off the steam or water passage, but one of the main troubles I have found in connection with the use of globe valves is the closing off of the valve by the formation of scale. To avoid this trouble, however, a small gate valve may be used in a horizontal position. There is one great advantage in having the column so that it may be cut off from the boiler, and that is to make repairs or clean the column while the boiler is hot.

The accompanying sketch shows a method of attaching water columns that has been found satisfactory. By placing the two crosses as shown and by using the gate valves the two plugs *C* and *D* can be removed and a rod run through the pipes to remove any obstructions. A blowoff pipe should always be inserted so that a circulation may be started at any time in the column. Care should be taken in setting the column body with reference to the water line of the boiler. The water line on horizontal return-tubular boilers should be at least three inches above the upper tubes and the gage line should be carefully set to this. Columns are frequently set where this is not carefully observed, in which case the column does not truly indicate the water level in the boiler and this is worse than not having a column.

C. R. McGahey.

Lynchburg, Va.

Accepting Advice

Some engineers seem to regard the supply salesmen as a sort of unnecessary evil and almost invariably treat them as such when they are given a friendly call by these "knights of the grip." Although I find it necessary to "get busy" sometimes, I have never yet seen the occasion or the salesman with whom I could not spare the time to sit down and "talk it over," even though quite sure that I needed nothing in his line. Invariably he has something to say which will be of interest and I have often received pointers which proved quite valuable in my work.

There are engineers who seem to be possessed with the idea that when anyone volunteers information that they are considered as a dummy or else the other man just wants to show how smart he is. It always seemed to me that such men held a spite for themselves. I do not mean that a man should believe everything he is told or reads, but if he has any powers of reasoning he can decide for himself the truth or falsity of the assertions. A person should not expect to put into practice only original ideas, for if such were the case, they would have to live to a great age in order to accomplish anything worth while.

I have been a reader of Power for a number of years and almost invariably upon picking up a new issue turn first to the advertisements to see if there is anything new in the market.

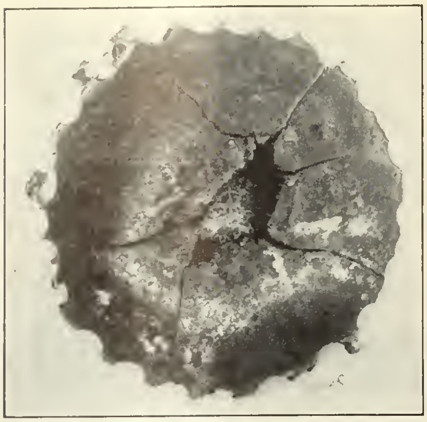
I do not think that engineers are any more obstinate about being shown the errors of their ways than men of other professions, but engineers notice it more because they come in contact with men of their own profession more than with men of other callings.

A. Ransom.

Wheeling, W. Va.

Laminated Boiler Plate

The accompanying view shows a piece of blistered plate that was cut from the front sheet of a 6x18-foot return-tubular boiler. The plate was ½ inch thick, originally, but after the laminations set in it was split into practically two plates, one ⅛ inch thick and the other ⅜ inch; the thicker being the one through which the hole was burned. The boiler had been in service two years and, although designed for 150 pounds pressure, it was being operated at only 100 pounds.



Hole Burned in Lamination

A. F. Rogers.

Negaunee, Mich.

Beading Boiler Flue

I would like to have the opinion of Power readers as to which is the better way to bead flues in a return-tubular boiler. Will they hold better and last longer if they are beaded down flat and tight against the head or should the small space be left which is the natural consequence of light beading when the outside edge of the bead is tight against the head?

L. Earle Brown.

Roundup, Mont.

A Solid Gasket

A piston pump which was used to raise water to a tank situated a few hundred feet up the side of a hill was left to the care of a roustabout in the mill, and occasionally the superintendent would stroll down to look it over. One day some imaginary trouble caused them to remove the head of the steam cylinder and when the head was replaced the trouble commenced. Steam was admitted to the cylinder and it made one stroke. The pump was taken apart but to no avail. Finally the roustabout discovered that by throwing over the reverse lever the pump would make two strokes, so in this way the tank was kept full for two or three days.

The superintendent did not care to expose his ignorance but finally decided it would be better to consult a pump expert than to run the pump by hand. When the latter arrived he requested that the cylinder head be removed. "Now, then," he said, "take your knife and make a hole in the gasket at that indented spot." When these orders had been executed the cylinder head was replaced, steam was admitted to the cylinder and the pump ran without a hitch, much to the astonishment of the two bystanders.

When the new gasket had been placed on the cylinder, holes had not been made to allow for the escape of the steam from that end of the steam chest.

Earl E. Webster.

Maxton, Ariz.

Liquid Discharging Device

The accompanying illustrations show a device for ejecting oil or other liquids from a barrel, which I have recently had patented. A metal plug *B* which is threaded and tapered to fit any sized hole is screwed into the bunghole of the barrel to be emptied. Through this plug a delivery tube *D* extends to within a short distance of the bottom of the barrel, and an inlet pipe *F* for compressed air or steam projects into the annular space between the plug and the delivery tube. When the apparatus has been adjusted, pressure is applied to the surface of the liquid in the barrel through pipe *F* and the liquid is ejected through the outlet pipe.

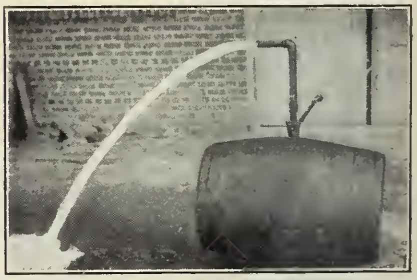


Fig. 1. Device in Operation

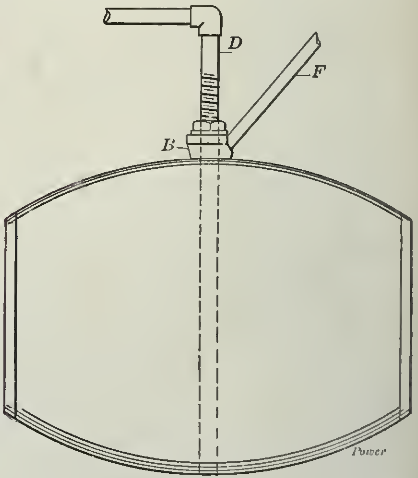


Fig. 2. Pipe Projecting into Barrel

Earl Pagett.

Coffeyville, Kan.

Size of Indicator Diagrams

I have noted for a number of years that indicator diagrams have been taken almost entirely with the standard length of approximately 4 inches, regardless of the hight of the diagram, and in some cases even longer diagrams have been taken. It is my opinion that the length should have some definite relation to the hight. Diagrams are taken for the purpose of setting the valves or determining the power which is generated by the engine. If they are taken for the former purpose, it is essential for best results that the important events of the stroke be sharply indicated. A long, narrow diagram will result in all of the important events being long drawn out, and make it difficult to determine at what point they occur. The same is true with a short high diagram. On the other hand, one which is nearly square will show up the various events of the stroke to a marked extent, and make it possible to determine accurately where each event takes place. Those taken for the purpose of determining the power require only such a length as will allow of accurate measurement and a sufficient area to be determined within the limits of accuracy of the length measurement. Any greater length than this is valueless. If a length can be determined within 1 per cent., it is long enough, and the area can be determined readily within this length if the diagram exceeds one inch in area. Also, for high-speed operation, the small diagram reduces to a very considerable extent the possible error due to pencil friction, inertia of the drum and other errors incident to indicator work.

Henry D. Jackson.

Boston, Mass.

Duplex Pump Trouble

A somewhat puzzling failure of a new duplex pump to work satisfactorily was due to the piston-rod nut in the steam cylinder becoming loose, thereby permitting steam to pass under the packing ring, between the piston head and the follower.

This pump had been in service only a few days when it was noted that one side began to lag in its stroke toward the steam-cylinder end, acting at first as though the discharge valve had become jammed, or the steam valve had become deranged. After examining these parts and finding them in good condition, the steam-cylinder head was removed, and on applying a wrench to the piston-rod nuts one of them was found to have loosened less than half a turn. This was tightened, the head was replaced, and the pump worked in a satisfactory manner.

This experience goes to show that packing rings should be a good working fit in pistons; otherwise an excessive waste of steam will occur without anyone suspecting such to be the case.

The ordinary duplex pump is especially inefficient in point of economy, but when the foregoing conditions occur, or where the pumps are permitted to run with a short stroke, their efficiency is still further decreased.

W. E. Rector.

Brooklyn, N. Y.

Remodeling a Pump

Several years ago on taking charge of a plant, I found a bilge or sewerage pump which was supplying a piping system carrying a pressure of about 40 pounds per square inch. The closing of the valves under this pressure was so sudden that it caused a hammer or pound which made all the piping shake, and several joints leak. I overcame this trouble by remodeling the pump as shown in the accompanying sketches.

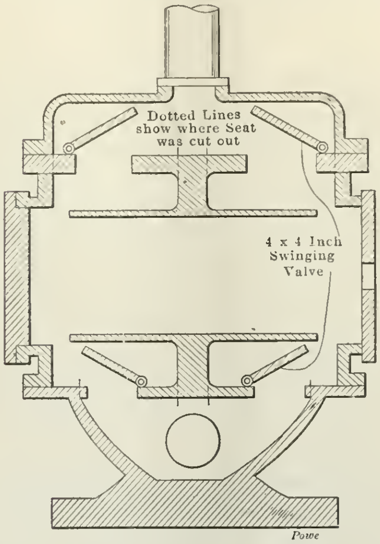

Dotted Lines show where Seat was cut out
4 x 4 Inch Swinging Valve
Power

Fig. 1. Pump Before Remodeling

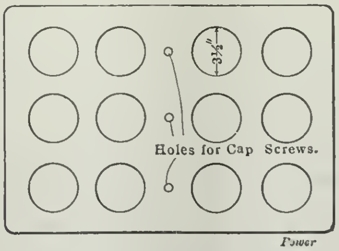


Fig. 2. Valve Plate

Fig. 1 shows how the inside of the pump was cut out to afford a less obstructed passage to the valve plates. I had two ½-inch plates, see Fig. 2, cast at a local foundry, and then planed and bored for twelve 3½-inch valves; also, twenty-four 3½-inch valve seats cast of brass and twenty-four studs for them.

The seats were finished in our own shop, being made a driving fit in the plates, and the studs were finished to ⅝ inch. Putting in the plates made it necessary to use two gaskets on each plate, also to procure longer cap screws for bolting the pump together again.

Fig. 3 shows the pump completed with the plates and valves in place. We had to raise the steam end of the pump off of the foundation a distance equal to the thickness of the valve plate in order to aline it with the water end. The total cost of remodeling this pump did not exceed $30. And the pump has since been giving entire satisfaction.

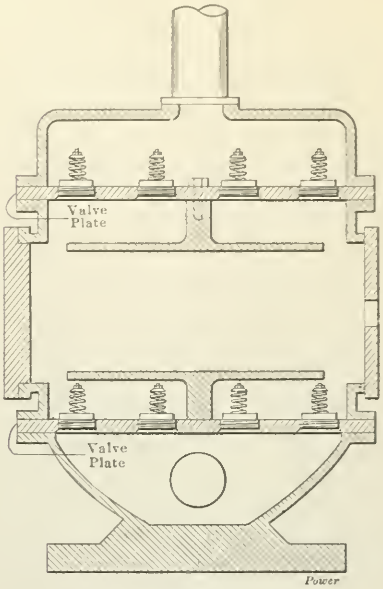


Fig. 3. Remodeled Pump

W. H. Beisley.

Natchez, Miss.

Shortage in Coal Weights

Some of us are very careful to weigh our coal before it goes into the furnace, but how many consider whether the cargo held out as per bill of lading? I have had my eyes opened during the past few months. For instance, I received a new cargo after all the old supply was cleared up, so I did not have to guess how much coal there was already on hand. This cargo weighed up correct in short tons, but it should have been in long tons.

On receiving another cargo of coal, it was dumped into cars of the same dimensions and loaded about the same; one load was carefully weighed, taking the total number of car loads and averaging the weight, the cargo came out 55 tons short, weighing in short tons. On the first cargo, which was 100 tons short, I got no redress as the coal company made an affidavit that its weight was correct.

I believe it would be well to put in a derrick scale and weigh each bucket as it comes from the vessel and then, if the weights are not as per bill of lading, apply to the Interstate Commerce Commission.

Thomas Hawkin.

Rockland, Me.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

The Economy of Vigilance in Small Matters

There have been several good hints on the first page of POWER during the last few weeks. The one in the issue of October 25 strikes me as being especially timely. I am inclined to think that when an isolated plant is shut down and service bought from the central station, it is, in a majority of cases, the fault of the engineer and not of the owner. Too often the isolated plant is not operated nearly as economically as it might be if the engineer did his best.

I was once called to take charge of a plant in which the operating expenses seemed to be running too high to suit the management. The man who hired me had a talk with me in regard to the coal bills and other expenses, but, as the plant was new and had only been running three years, he did not seem to know where the trouble lay.

The plant consisted of three fire-tube boilers, 72 inches by 16 feet in size, two high-speed engines and such other equipment as is usually found in a plant of this kind.

After getting fairly "squared away" I began to look for the cause of the high operating expense. I learned that the man who had formerly had charge of the plant had been accustomed to having the flues cleaned only once a week. I had them cleaned every day, and noticed quite a change in the coal bill. I also learned that they had been in the habit of carrying the water level in the boiler at a hight in the gage glasses of 3 or 10 inches. The bottoms of the glasses were about 4 inches above the top tubes, so that they had been carrying about one foot of water over the top of the flues, thus leaving a very small steam space and getting very wet steam.

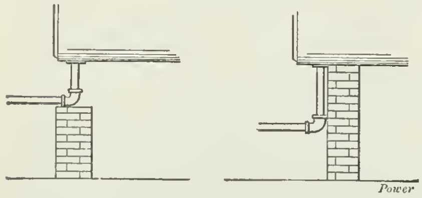


FIG. 1. FIG. 2.

ORIGINAL AND IMPROVED ARRANGEMENT OF BLOWOFF

The coal pile was some 25 or 30 feet from the boiler and it was the habit of the fireman, when stoking in a fire, to carry each shovelful to the boiler and leave the furnace door open while doing so. I noticed that there was a constant cloud of vapor rising from the hotwell, and on investigation found that the blowoff valves on the boilers were leaking.

The main exhaust pipe from the engines ran under the engine-room floor, out into the boiler room and then up overhead to the heating system. Under the boiler-room floor, I found a ¾-inch valve in the exhaust pipe, which had been put in to bleed the water out of the exhaust pipe. A long valve stem had been attached to this valve and run up through the concrete floor, for the purpose of opening and closing the valve. This valve stem had been broken and the valve left open to the sewer, thus creating a considerable waste when back pressure was used for heating.

The condensation from the heating system came back into a return tank in the boiler room, was pumped from the tank into a feed-water heater with one pump, and from the heater into the boilers with another. There were no automatic float-operated valves on the pumps, so that considerable hot water was lost by overflowing from the heater, it being difficult to regulate both pumps so that this would not occur. On taking the temperature of the returns, I found that they were coming back at a temperature of about 200 degrees Fahrenheit. I therefore arranged to pump the returns directly back to the boilers, using the heater only when taking fresh water.

On opening the door in the back end of one of the boiler settings, the ashes were found to be wet. The blowoff pipe had a brick pier built up under it, as shown in Fig. 1. I found that this pipe had been leaking badly and was burned clear through, nearly the whole way around, only about 1½ inches of metal remaining. One of the other blowoff pipes was found to be in much the same condition. Probably the only reason that the pipe stayed on the boiler was the fact that it was held there by the brick pier. These blowoff pipes were torn out, as were also the brick piers, and extra-heavy pipe put in as shown in Fig. 2.

One of the engines was of the inclosed, self-oiling type. The oil drips, which came out around the piston-rod packing of this engine, were led by a drain to the sewer. This was disconnected and the drips run through a filter and used again. Both engines were indicated and the valves properly set.

While any of the above mentioned items would not create much of a loss by itself, yet, when all were corrected, it resulted in a saving of 25 per cent. on the first year's operation.

It is the little things that count.

R. L. RAYBURN.

Kansas City, Mo.

Water Hammer

In the discussions in POWER on boiler explosions and water hammer, the general opinion is that water hammer due to cutting in a boiler is often the cause of an explosion.

From experiences I have had I came to the conclusion that water hammer does not cause a great many of the explosions credited to it, except in those cases where water has a chance to trap over the stop valves.

I am operating a battery of vertical water-tube boilers. After cleaning and filling, the usual method of allowing the air to escape through the water column while raising steam did not seem to work in this case. As soon as the valve was cracked to cut in the boiler, the snapping, hammering and vibrating would start. To be up 30 feet from the floor on a boiler that is trembling under your feet, and expect something to let go with each snap, is not very pleasant. Extra precautions were taken in raising steam and cutting in the boilers, still we had the same trouble every time a boiler was cut in.

The stop valves were of the angle pattern fitted to the steam nozzle on the boilers. The steam header is 6 feet below the stop valves and connected to them with long-sweep bends. Therefore, there was a perfect drain and I am positive that water was not the cause of the hammering and vibrations. We tapped the steam nozzles and put in ⅜-inch vents to allow the air to escape while raising steam, and that ended the trouble, not only in the boiler room but in the engine room as well, where the vacuum was frequently lost.

This case may have been an exceptionally bad one, but how many plants are there that have perfect vent arrangements for removing the air from the boilers while raising steam? In a great many, the stop valves are placed several feet from the boilers so as to avoid a water trap above the valve. With this arrangement a perfect air trap is formed below the valve and on cracking the valve vibration starts.

My experience has convinced me that

if every engineer would put a vent below and a drain above the stop valves and instruct his men to try both before cutting in a boiler, there would be less explosions credited to water hammer and unequal pressure.

H. J. MISTELE.

Milwaukee, Wis.

Isolated Plant Costs

I have been much interested in the discussion which has been going forward in the columns of POWER recently on the subject of "The Isolated Plant versus the Central Station." I therefore offer my experience in this regard. The subtended figures are authentic and were furnished by the audit department of a large building in the operation of the plant in which I am engaged.

Cost of generating plant in excess of that required for central-station service:

Switchboard and wiring..	$1600.00	
1—100 kw. generating unit, including foundation..................	3575.00	
1—75 kw. generating unit, including foundation..................	2775.00	
Exhaust pipe trenches and covers................	125.00	
Piping for engines, heater, pipe covering, etc......	1700.00	
1—200 H.P. boiler and necessary connections...	3300.00	
Total..............		$13,075.00

Expense of operating plant:

12% interest and depreciation against plant cost of $13 075.00............	$1569.00	
Coal....................	4987.50	
Water..................	250.00	
Wages of chief engineer...	1800.00	
Wages of ass't. engineer..	1144.00	
Wages of fireman........	832.00	
Oil, waste, supplies, repairs, lamp renewals, ashes................	975.00	
Rent for 2000 sq. ft. of space taken up by generating plant at 30 cents per foot...............	600.00	
		$12,157.50

Electricity, if purchased from the central station at Mr. Tweedy's price of 5 cents per kilowatt-hour, would cost:

356, 500 K.W. hours at 5 cents....	$17,825.50
Building plant....................	12,157.50
Saving Over Central Station....	$ 5668.50
To this must be added the cost of coal for heating and keeping steam the year around for manufacturing purposes which would be........	$ 1900.00
Labor to take care of plant.........	1600.00
Engineers' supplies, water, ashes, repairs, etc......................	500.00
Total saving over the cost of central station service....................	$ 8668.50

CHARLES W. PEACOCK.

Chicago, Ill.

I have been very much interested in the articles appearing from time to time in POWER, in regard to the relative cost of electricity as furnished by the central station and the isolated plant.

In the plant of which I have charge there is 400 horsepower in boilers, a greater part of which is used exclusively for heating purposes. One boiler of 80 horsepower rated capacity, carrying 60 pounds pressure, supplies steam for the laundry, cooking, etc.; the balance of the boilers are operated at from one to ten pounds pressure according to the weather conditions.

Electric current for lighting is furnished by the central station; the average monthly consumption is about 2700 kilowatt-hours at a cost of about 6.5 cents per kilowatt-hour, making a yearly cost of about $2115.

I have been contending for some time that our light was costing too much, but have not as yet been able to convince the management that we could produce the same current much cheaper in a plant of our own.

The heating boilers are operated from seven and a half to eight months each year while the 80-horsepower boiler is run the entire year.

As the plant has to be operated 24 hours per day, the only extra help required would be one assistant engineer. The extra coal consumed would be very little, especially in cold weather, as the exhaust steam would be used in the heating system; even in warm weather, the exhaust could be turned into the water heater which supplies the institution with hot water and in which live steam is now used.

I should like to see some statistics from plants of about the size of this one where current is generated.

H. A. MUNYON.

Syracuse, N. Y.

Installing Globe Valves

I have been interested in the different views expressed in recent numbers of POWER relative to the installation of globe valves.

Ordinarily, I believe that the better way is to put the valve in so that the flow of steam or water, as the case may be, will be against the under side of the disk.

In an instance I have in mind, however, the opposite arrangement would have been preferable. An 8-inch throttle valve of the outside-screw type having a brass spindle was connected to a Corliss engine. Owing to the failure of the trap attached to the drip pipe, this valve was often subjected to the effects of water hammer.

The engineer thought that the engine was not getting enough steam, although the throttle seemed to be wide open. When I entered the engine room I heard the throttle valve clicking away every time the engine took steam. This was conclusive proof that the valve was not fully open, for, had it been, the valve spindle would have drawn the disk tight against the cap and thereby prevented the clicking sound.

At the first opportunity we took the valve apart and found that the spindle was so enlarged, due to the bumping under the disk, that we could not open the valve more than ¾ inch, while it should have been opened 2 inches in order to utilize the full capacity of the pipe.

I put the spindle in a lathe and reduced the thread to work as designed and at the same time took dimensions and later on had a spindle made of mild steel which did not upset.

One morning before starting time the engineer informed me that the engine had "turned over during the night and as hot as if she had been running."

The trouble was that the valve had broken, probably from the effect of water hammer. Not having a spare part on hand, a shutdown of half a day was caused until we made a temporary valve by utilizing a blank flange which was faced off and riveted to a yoke to engage the spindle.

Another example of a similar nature was that of an underwriters' fire pump which had a throttle valve of the outside-screw type wherein the pressure was under the disk. The valve was so located that under certain conditions water hammer would occur. The valve got so that it would not open to more than one-half of its capacity because the spindle was upset from the effect of water hammer as in the first case.

I repaired this spindle by the same process as that applied to the engine valve.

In both instances, I suggested turning the valves, "end for end," and thereby bringing the pressure on the top of the disk which would prevent undue pressure against the spindle in the event of water hammer.

Relative to the throttle valve on the fire pump, in the case of fire, there would have been a chance for controversy if it could be proved that the pump could not get sufficient steam to actuate it to its full capacity. Experience teaches me that inspectors should note the condition of the throttle valves on fire pumps.

J. W. PARKER.

Clinton, Mass.

Metallic Packing

In the power house where I am employed there are three 500-horsepower double-acting gas engines and one 14x16-inch automatic high speed steam engine, all fitted with metallic packing. We have found that the packing cuts the rods to a taper which necessitates having them turned in the lathe in order to make the packing hold. The gas-engine rods run about one year but that of the steam engine, which was but recently installed, has run only about eight months and will have to be replaced by a new rod, it is cut so badly.

Why do the rods cut to a taper?

The gas-engine packing is lubricated by pumping a good grade of cylinder oil into the caging of the packing by an oil pump which delivers the oil into the packing on the inward stroke of the piston. Can any reader explain why the rods cut tapering? The tension of the garter springs should be equal at both ends of the stroke.

W. D. Marquest.

Alexandria, Ind.

One or Two Boilers

I have been waiting in the hope that some of the contributors to the "One or Two Boilers" question that has had considerable discussion of late, would reply to R. McLaren's statement published in the October 18 number.

I regret that Mr. McLaren did not give some data that would enable us to learn how a saving of over 25 per cent. was effected, for it is quite evident that a man who can cut his fuel expense down 25 per cent. without any outlay for fuel-saving devices on the part of his employer, is wasting time in remaining in a two-boiler plant. If Mr. McLaren had told us what type of boiler he has in use, the boilermaker's rating of it, the grate area and kind of draft employed, also whether conditions were the same in each case, as well as the amount of evaporation under each set of conditions, he might have thrown much light on a question that is of vital importance to every person who has to pay fuel bills as well as to engineers who design or operate the plants.

A question that has claimed the attention of engineers for many years is the advisability of driving steam boilers to their limit. This is a very broad question and any practical experience that will help to its satisfactory solution would doubtlessly be well received.

In many large power plants the boilers are forced to the limit; in fact, this is the almost universal practice in electric-light and power plants. As the engineers and managers of such plants are generally men of intelligence and experience, the natural inference is that the practice is commendable. But in what way is it commendable?

We know that most large power systems are owned by corporations in which the members of the board of directors are elected by the stockholders from time to time. It is the desire of each board of directors to make a good showing regardless of the wear and tear on the plant. A case that came to my notice recently may serve as an illustration. The chief engineer of a large corporation was shown something new in the fuel- and labor-saving line and it appealed to his judgment so that he wanted to try it in one of the plants where fuel was the equivalent of over $8 per ton. He entered a requisition for the necessary funds. He was called on at the first following board meeting and asked if it was necessary to have the device in order to keep the plant in operation. He replied that it was not but that it was evident that the device would pay for itself in reduced operating expenses within a year, and that he thought any investment that would pay at least 100 per cent. *per annum* and probably last as long as the boilers was worthy of consideration, etc. His request was turned down with a statement to the effect that the amount he proposed to spend for betterment could be used to better advantage as dividends. And there you are! It is a question of "frenzied finance" against the judgment of a competent engineer, therefore it does not seem safe to bank too heavily on the practice employed in large power plants, for the question of cost of maintenance is seldom given much attention.

W. H. Odell.

Yonkers, N. Y.

In the October 18 issue, there is a note by R. McLaren, entitled "One or Two Boilers," in which he does not venture an opinion, but gives the readers his own choice.

Mr. McLaren's experience would lead to two conclusions: First, there should be two boilers installed, for with only one boiler he would have had to shut down his plant when it needed repairs; second, if he can save so much fuel by running one boiler instead of two, he should always run only one boiler.

W. H. Keller.

Ansted, W. Va.

Thermometers versus Gages

Mr. Peabody's article under the above title in the October 25 issue is very interesting and profitable, but there is room for considerable argument before the steam gage is scrapped and exclusive use made of thermometers.

From the steam tables it is seen that for each equal increment of temperature above 32 degrees Fahrenheit the increment of pressure is a very small fraction of the temperature increment. At 150 degrees Fahrenheit the pressure increment begins to increase more rapidly until at 300 degrees it is equal to the temperature increment. Above this temperature they converge until at 400 degrees a change of one degree is accompanied by a change of 2.8 pounds in the pressure. From this it is clearly evident that with equal divisions on a thermometer and pressure gage, the thermometer would indicate with much greater accuracy the condition of the steam up to 300 degrees. Above 300 the pressure gage would indicate more accurately the condition of the steam.

Since, in actual practice thermometers and steam gages do not have equal divisions, it will be interesting to see what results would be given from the use of an actual steam gage and thermometer. If we assume a 12-inch steam gage and a 10-inch thermometer graduated to pounds and degrees respectively, we shall have about a 20-inch scale on the gage divided into 250 divisions, and a 10-inch scale on the thermometer divided into 370 divisions. This means that the length of a pound division on the gage will be very nearly three degrees on the thermometer. With this arrangement it is evident that the thermometer will be more accurate up to the temperature where one degree change in temperature is accompanied by a change of pressure of one-third of a pound. This is at 220 degrees. Above this temperature the gage would indicate the condition of the steam much more accurately. At 150 pounds, which is about the average of present-day boiler practice, a change of one pound on the gage would be attended by only one-sixth the same linear movement on the thermometer. From the foregoing it is safe to state that a thermometer will give more accurate results up to about atmospheric pressure, but that for boiler work it would hardly seem that a "little glass thermometer" is as convenient to use or as accurate in its results as a "big nickeled gage."

William Nottberg.

Kansas City, Mo.

Diagrams for Criticism

Referring to the diagrams for criticism in the November 1 issue, Mr. Fryant states that he is reasonably sure that the jagged expansion line was not caused by the indicator. My experience has been that such a line is caused by the indicator only. The peak on the expansion line indicates that the indicator piston was sticking, and the wavy condition of the expansion line indicates the inertia of the instrument.

T. J. Hammersley.

Milwaukee, Wis.

Quarter Turn Drive

In reply to Mr. Bitterlich's inquiries in the November 1 issue, the following is offered in the hope that it may be of help to him:

All drive-belt formulas are empirical, giving the millwright but a general idea of what is required, and on the subject of quarter-turn drives, in particular, even the latest textbooks are vague. For instance, it is stated that "the line representing the plumb line will be the center of the vertical belt," when, as a matter of fact, in a quarter-turn drive employing only two pulleys, neither belt is vertical.

In order to run a quarter-turn drive successfully, the following points must be observed:

The distance between the near faces of pulleys must not be less than four times the width of belt. and twice that distance is very desirable.

The pulleys must be so placed that a

plumb line, as *E F* in the accompanying sketch, tangent to, touching and dividing the face of one pulley is also tangent to, touching and dividing the face of the other pulley. The belt must lead off each of such faces. In the sketch, the line *E F* passes tangent to the faces *A* and *B*. The belt leads off of the driver at *A* and on to the driven at *C*; leads off the driven at *B* and on to the driver at *D*. Consequently, neither belt is vertical.

Both pulleys should be as small as is consistent with the amount of power to be transmitted.

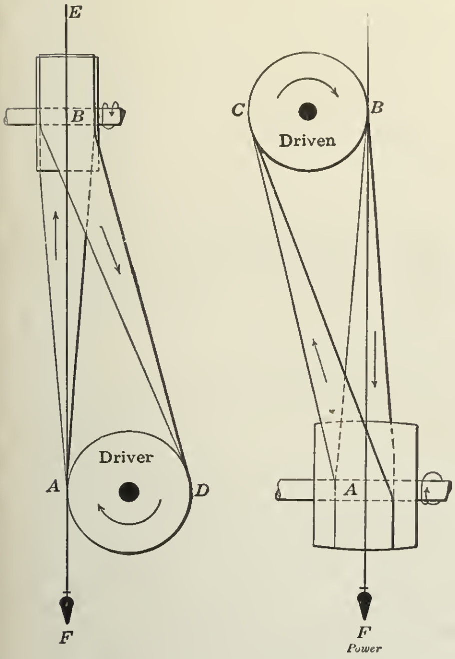


Diagram of Quarter-turn Drive

The face of the driving pulley should be crowned at least 30 per cent. more than the usual crowning given to a crowned pulley.

The ratio between the width of the driver and that of the driven should be as 5 is to 3.

About 25 per cent. of the contact is lost with a quarter-turn drive even when the pulleys are of the same size, owing to the imperfect elasticity of the belt. A narrow heavy belt is preferable to a wider and lighter belt, as the latter stretches rapidly on one edge and has a short life.

Leather-link belts give the greatest satisfaction. Being hinged down the middle, such a belt will accommodate itself to the high crowning of the driver, which increases the efficiency of the drive.

A double belt with a narrow upper layer on top and to one side, depending on whether the belt is right- or left-hand running, will also give good results, as the inequality of thickness forces this side against the pulley and makes a good contact.

A heavy-ply narrow rubber belt will give good service if laced smoothly and evenly, with a half turn in the belt, so as to bring each side of the belt alternately against the pulley faces, thus stretching each edge of the belt equally. With a short drive the belt is difficult to throw on, when laced thus, but when run at belt speeds of 1000 feet per minute and more, it will give great satisfaction, both in efficiency and durability.

R. Reddie.

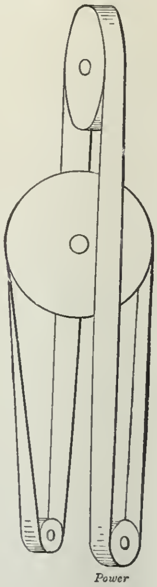
New Market, Tenn.

In the issue of November 1, W. J. Bitterlich asks for information relative to a quarter-turn belt installation.

The minimum distance between shaft centers considered to be good practice is about seven times the width of the belt. The single belt is best used as a basis for calculation. Thus, in Mr. Bitterlich's problem the distance between centers should be kept at least seven times the 6-inch width, even though a change to a 4-inch double belt were made. Designers are not inclined to use this extremely short distance extensively, however, and the following formula is the basis for a permanent and economical operating design:

$$L = C + 15\ W,$$

where L equals the distance between shaft centers in inches; C equals the circumference of the larger pulley in inches, and W equals the width of the belt in inches.



Arrangement to Secure Effective Distance Between Centers

With this formula the distance between centers in Mr. Bitterlich's problem should be about 12 feet. The effective shaft distance for vertical quarter-turn drives may be obtained by using idlers on the floor, as indicated in the accompanying figure.

The widest belt which I have seen in use on a quarter turn was a double belt 12 inches wide. The belt transmitted 60 horsepower.

As to the maximum belt speed consistent with good practice, the following is taken from Kent's "Mechanical Engineers' Pocketbook": "The belt speed for maximum economy should be from 4000 to 4500 feet per minute."

There are often a great many local conditions, however, which prevent the use of this ideal speed, and one must design the belt for the nearest approach to it which may be within the limits of possibility, using the values mentioned as a general guide.

J. A. Pratt.

Williamson School, Penn.

Thermal Storage

In a recent issue, under the caption "Thermal Storage," the results of a test were given, showing that the evaporation had been increased from 123.6 per cent. of the rated capacity of the boilers to 217.3 per cent., an increase of 93.7 per cent., secured by means of so called thermal storage tanks.

It would appear to me much better practice to install live-steam purifiers of generous capacity and thus secure the advantages of the softened water as well as those resulting from the thermal storage.

I desire to score a point in favor of nonautomatic boiler-feed control. Advantages are obtained by maintaining the water level as high as conditions will permit—two and one-half to three gages—the advantages of having enough water stored in the boiler so that the feed pump may be almost closed down for ten or fifteen minutes if necessary, when a peak load comes on, in this way almost doubling the evaporative capacity, thus making it possible to maintain satisfactory service until the firemen have a chance to get the fires up to the usual requirements.

I offer the results of a test of a small light and power plant as an example of what can be accomplished in a small plant during a peak load.

Combined rated capacity of 2 return-tube boilers, horsepower	276
Equivalent evaporation from and at 212 degrees, pounds	14,09[illegible]
Horsepower developed for one hour	408.6
Per cent. of rated capacity	148
Load in kilowats with two engines	270

The plant was operated with moderate natural draft, and West Virginia coal said to have a heat value of 14,070 B.t.u. per pound was used.

While this load of 270 kilowatts was on, an additional load of 110 kilowatts was added, making it necessary to put on an additional engine. This load of 380 kilowatts was carried for 25 minutes, during which time the feed pump was almost shut down, making it possible to carry the load. This was done for three days in succession, although on the other two days the excess load was slightly lighter. No evaporative test was made while this load was on.

Waldo Weaver.

Middletown, O.

Combustion and Boiler Efficiency*

By Edward A. Uehling

The importance of carbon dioxide as an index to combustion and, in connection with flue-gas temperatures, to boiler efficiency, is discussed. The curves show the relation between the carbon dioxide and the oxygen.

A steam engine correctly designed, properly constructed, accurately adjusted and adapted to the work to be performed will give maximum economy. The attendant can do little or nothing either to increase or to decrease its efficiency. This is also true of the steam turbine and the dynamo, but the efficiency of a steam generator depends on far more complex conditions, many of which are not only beyond control but are continually changing. Some of these are: first, *atmospheric conditions;* second, *the quality of the fuel,* which at best is never quite uniform, even when similar in kind, and often varies greatly in its content of moisture, fixed carbon, volatile combustible matter, oxygen, ash, etc.; third, *the condition of the boiler* as to setting, dirty heating surfaces, air infiltration, draft, etc; fourth, *the required output,* which varies in some plants between very wide limits.

The heat energy carried off by the flue gas in general boiler-room practice is rarely as low as 15 per cent. and not infrequently reaches 40 per cent. of the calorific value of the fuel burned. This loss is made up of two distinct factors, the sensible heat of the flue gas and the potential heat of the combustible constitutents. Other things being constant, the former varies directly as the stack temperature and inversely as the percentage of CO_2. The latter is independent of the stack temperature, but for any given percentage of combustible constitutents in the flue gas the loss also varies inversely as the percentage of CO_2. The only heat loss that is independent of the percentage of CO_2 in the flue gas is that due to the moisture contained in the fuel. but it is dependent upon the stack temperature.

For any given boiler and setting, driven at a given rate, the temperature of the flue gas depends upon the condition of the heating surfaces and on the amount of air infiltration. When the heating surfaces become dirty, the stack temperature will be correspondingly higher, other conditions being the same. Air filtration lowers the stack temperature in proportion to its ratio to the actual products of combustion. Because of dirty heating surfaces, it should be above normal, but air infiltration may actually bring it below normal, so that the stack temperature alone cannot be depended upon as an index to either economical or wasteful operation, especially since it gives no clue to the efficiency of the furnace. Provided the boiler is properly designed and set and is otherwise in proper condition, its economy depends entirely upon the regulation of the furnace.

Since an excess of oxygen is necessary for complete combustion, it is held by some combustion experts that the percentage of oxygen is a better index to the efficiency of combustion and boiler economy than CO_2. But when it is considered that CO_2 is more readily determined than O, and that there are a number of practicable instruments on the market that will autographically record the percentage of CO_2 in the flue gas at short intervals, there can be little room for doubt as to which of the constituents is the better adapted to serve as the index of economical combustion.

Analyses show conclusively that there is no relation between either O or CO_2 and CO. All that can be discerned with certainty is that there is a tendency toward higher CO as the percentage of CO_2 goes up. The sum of the percentages of CO_2, O and CO contained in flue gas varies with the excess of air, the completeness of combustion, the composition of the fuel and the temperature at which the gas is analyzed. When combustion is complete and the fuel consists of carbon only, the sum of CO_2 and O cannot exceed the percentage of oxygen in the air; that is, 21 per cent. It will always be less than this, because the gas is saturated with moisture, which in general gas analyses is reckoned with the nitrogen. This introduces a possible difference of 1.3 per cent. due to the differences in the moisture content of the gas between 62 and 92 degrees.

When carbon is burned to CO every molecule of O_2 combines with two atoms of carbon and forms two molecules of CO. The gas may, therefore, contain

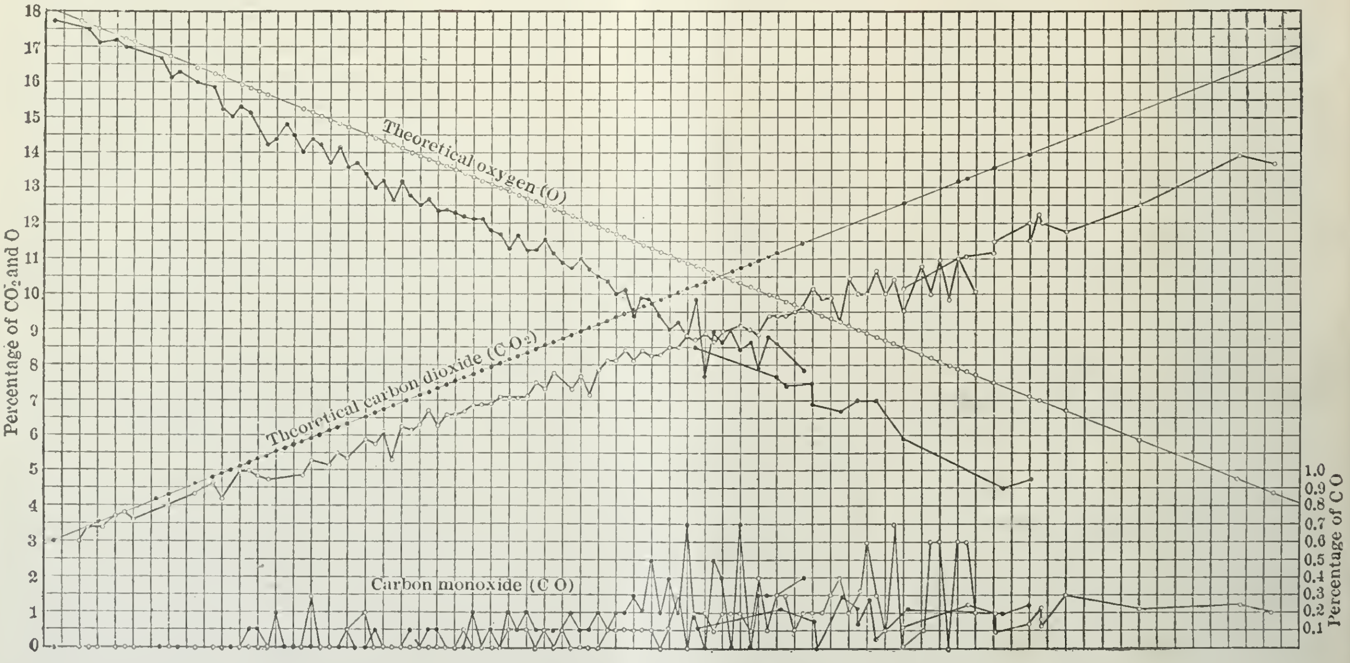


Fig. 1. Diagrammatic Representation of Government Analysis

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*Abstract of paper delivered before the American Society of Mechanical Engineers.

34.7 per cent. of CO as a maximum. If combustion is incomplete and CO is present, the sum of CO_2, O_2 and CO in the dry gas may be greater than 21 per cent. by an amount equal to

$$\frac{34.7 - 21}{34.7} \times CO = 0.4\, CO$$

so that in this case the sum of the oxygen components in saturated gas would be

$$21 + 0.4\, CO - \frac{H_2O \times 21}{100}$$

Since the percentage of CO in normal flue gas reaches 1 per cent. only in exceptional cases and the reduction due to moisture is not likely to be less than 1 per cent., flue-gas analyses in which the sum of the oxygen components exceeds 21 per cent. must be looked on with suspicion. In burning fuel containing a considerable quantity of hydrogen the sum of CO_2, O_2 and CO will always be less than 21 per cent., because all the oxygen which combines with the H_2 forms water vapor, which is condensed, while the nitrogen remains as a diluent.

The highest practicable furnace efficiency is obtained when the greatest possible part of the potential energy contained in the coal is converted into sensible heat by combustion. The highest boiler efficiency results when a minimum of this heat is carried off by the flue gas. The former is realized when combustion is complete, the latter when the weight and temperature of the flue gas are a minimum. Since the weight of flue gas decreases as the percentage of CO_2 increases, high boiler efficiency requires high CO_2. On the other hand, when the furnace is run so as to produce a maximum percentage of CO_2 there is danger of reducing its efficiency because of incomplete combustion due to a minimum of free oxygen. It would seem, therefore, that the highest combined efficiency of boiler and furnace would be obtained when the gain due to reduced weight of flue gas, carried by an increased percentage of CO_2, is balanced by the loss due to incomplete combustion.

Next to careless or unskilful firing, air infiltration causes the greatest loss in the flue gas. Unless especially provided against, it is rarely less than 15 to 20 per cent. of the volume of the flue gas and not infrequently reaches 50 per cent. and over when cracks in the settings are allowed to exist and warped and badly fitting cleaning doors are tolerated.

The curves shown in the accompanying figures were plotted from the results of a number of Government tests. The straight lines represent the theoretical amounts of CO_2 and O which the combustion products would contain if pure carbon were burned with pure air. The analyses were plotted with regard to the percentage of CO_2 and O respectively. The CO_2 was placed in its proper position on the theoretical CO_2 line for the percentages of CO_2 and then the accompanying percentages of O and CO were plotted on the same ordinate in the position called for by their value. In the same manner the O was placed on the theoretical oxygen line and the CO_2 and CO were plotted according to their values on the same ordinate. To show the values of CO more clearly, they are multiplied by five.

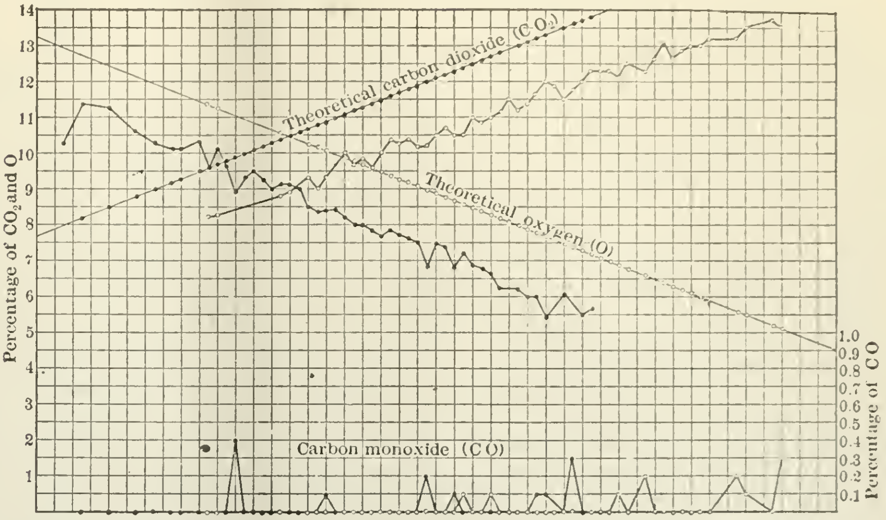

FIG. 2. VARIATIONS OF O, CO AND CO_2 AS SHOWN BY ANOTHER SET OF TESTS

Fig. 1 shows that the percentage of O diminishes as the CO_2 increases, and *vice versa*. It will be noted that both the O and CO_2 lines converge toward the theoretical line as the CO_2 diminishes, which is as it should be if the analyses are correct, because when the CO_2 is zero, the per cent. of O must be that of the air, 21 per cent.

Considering the great variety of fuels represented by the Government analyses it is remarkable with what uniformity the CO_2 and O vary in accordance with what should be expected from theoretical considerations. It is justifiable to conclude that the relation of CO_2 to O in Fig. 1 holds true for all ordinary conditions of simple combustion. It therefore follows that if the percentage of CO_2 contained in the products of combustion is known, the accompanying percentage of O may be inferred from it with a practical degree of approximation. That no such reliable relation exists between the CO and either of the former has already been stated. That this should be so becomes evident when it is considered that the relation between CO_2 and O depends on a natural law, whereas the percentage of CO in a large degree depends on caprice.

The CO line in Fig. 1 illustrates the irregularity of this component. It was stated at the outset that the tendency toward a higher percentage of CO in flue gas becomes greater as the percentage of CO_2 goes up; but the curves show a maximum of over 0.4 per cent. of CO with CO_2 between 10 and 11 per cent., whereas with CO_2 between 11 and 14 per cent. the CO does not average above 0.2 per cent. This apparent anomaly is explained by the fact that the Government analyses includes from 20 to 30 per cent. of infiltrated air and that the real combustion products contained about 25 per cent. more CO_2 and 25 per cent. less O than the analyses show. In reality, therefore, the high CO readings occurred with CO_2 from 12.5 to 13.75 per cent. This would place the high CO in its proper relation to CO_2.

Fig. 2 shows the same characteristics as Fig. 1; that is, the O line follows a trend which should meet the theoretical O line at 21 per cent. when CO_2 reaches zero, and the CO_2 line meets the zero point of the theoretical CO_2 line when O reaches 21 per cent. The CO is irregular and has a moderate tendency toward higher values as the CO_2 reaches the higher values. With the exception of the few analyses previously discussed, this diagram shows very good combustion.

CONCLUSIONS

The conclusions are that the percentage of CO_2 and the stack temperature are the two controlling factors in the production of furnace and boiler efficiency. In view of the fact that both of these components are so easily determined and that apparatus is in existence which will continuously indicate and record them, it is lamentable that more general use is not made of these indexes. It is not generally feasible, nor is it at all necessary, in order to obtain the most economical results, that calculations be based on the ultimate analysis of the coal, but a full analysis of the flue gas should form part of the daily routine. Inasmuch, however, as the most economical percentage of CO_2 varies considerably with the kind of fuel used, and also to some extent on the method of firing, hand or mechanical, the kind of stoker and rate of driving, it becomes very desirable, if not absolutely necessary, to make a series of tests in which all the essential elements are determined. Having established the most economical value of CO_2, this component of the flue gas will serve as a correct guide to the fireman. An automatic record, preferably a continuous one, for the engineer is of the greatest value. If, in addition, the stack temperature is indicated for the fireman and recorded for the engineer in charge there should be little difficulty in getting maximum efficiency for any particular installation and for any grade of fuel.

Heat Transfer in Surface Condenser*

By George A. Orrok

A series of tests to determine the variation in heat transmission through different condenser materials with variations in the temperature difference, the velocity of the water and the velocity of the steam, and the effect of air mixed with the steam.

Many attempts have been made to ascertain the laws of heat transmission for condenser practice but the results of the various investigators have varied widely. These differences, however, did not trouble designers seriously as long as small condensers and vacuums of over two pounds absolute were the practice, but with the development of the steam turbine the need arose for condensers capable of condensing more than 200,000 pounds of steam per hour at an absolute pressure of less than one pound.

To meet the demand for more definite information upon the subject the writer conducted a series of experiments which confirm and complete much of the work of the earlier investigators. The object in making the tests was to determine the heat transfer through various kinds of condenser tubes and the laws governing its variation under different conditions of steam temperature, pressure and velocity, velocity of circulating water and mean temperature difference.

In order to obtain these data, a small surface condenser was constructed with a relatively small cooling surface and was operated under conditions as near as possible to actual condenser conditions. The steam-inlet, dry-vacuum and circulating-water lines were so arranged and controlled that any desired vacuum, any desired velocity of circulating water and any desired mean temperature difference could be maintained and that any one of these conditions could be varied through a considerable range quite independent of the other two. A hotwell was constructed and so connected to the condenser that the temperature of the condensed steam and its amount could be determined. Thermometers, pressure gages and a water meter were installed and a record was made of all data.

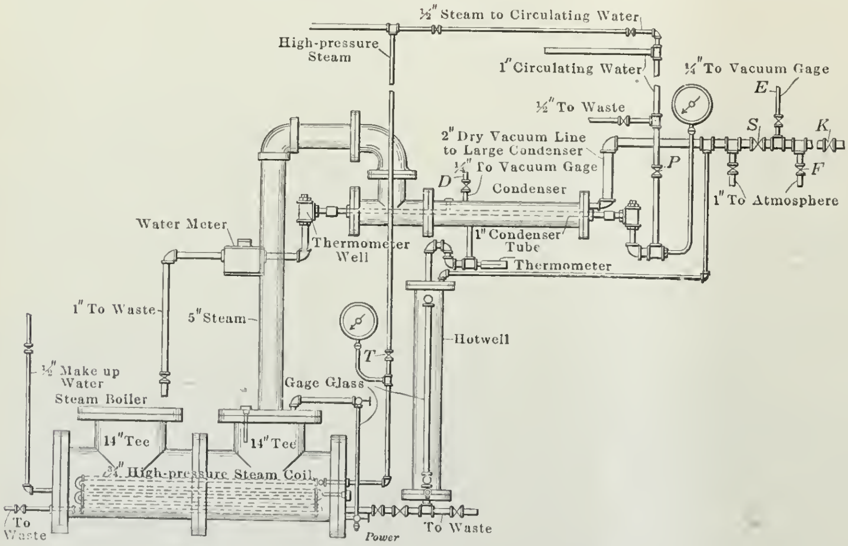

Fig. 1. Apparatus as Actually Used

At first steam was taken from a 12-inch exhaust main, but the quality was found to vary so much that it was found necessary to alter the apparatus somewhat and build a small independent boiler for generating steam as it was needed by means of a high-pressure steam coil. The boiler was constructed of two 14-inch flanged tees bolted together, with blank flanges bolted on the ends, and a steam coil, consisting of approximately 22 feet of ¾-inch wrought-steel pipe, was placed in the boiler in such a way that it was entirely below the water level and still left a sufficient disengaging surface for the steam generated in the boiler. Steam from a high-pressure main was supplied to this coil through a 1-inch pipe. Makeup water for the boiler was taken from a city-water line through a ½-inch pipe. A 5-inch pipe connected the boiler with the condenser, which was made of extra-heavy 5-inch wrought-steel pipe with cast-iron screwed flanges on each end; to these were bolted cast-iron blank flanges. A 1-inch condenser tube passed through the center of the condenser and projected through each end, carrying thermometer wells. The bottom of the hotwell was also connected to the boiler by a ¾-inch line containing a gate and a check valve so that the condensed steam could be returned directly by gravity to the boiler; this obviated the necessity of breaking the vacuum and emptying the hotwell after each test. The whole apparatus was as shown in Fig. 1.

With this arrangement, for any desired opening of the valve *S*, a constant velocity of steam through the condenser could be maintained by regulating the air valve *F* and the valve *K*, so that the steam passing through the valve *S* had a constant drop in pressure. This drop, due to velocity of the steam, was considered to be the difference in pressure as read at *E* and *D*.

The subject of air leakage was investigated by closing all valves except *S* and *K* in the vacuum line, when it was found possible to hold a vacuum in the small condenser practically equal to that in the large condenser. The valve *S* was then closed and observations were taken as to the rate of fall of the vacuum in the apparatus. It was found that the vacuum would drop only 28 to 27 inches in 15 minutes. These observations were taken before every series of tests.

In order that the condition of the tubes as to cleanliness should not affect the heat temperature, all tubes used for testing were thoroughly cleaned every ten or twelve tests. A series of tests under constant conditions were found to have a maximum variation of not over 10 per cent. and an average variation of 3 to 4 per cent. In these tests, the vacuum was varied by regulating the amount of steam admitted to the steam coil through the valve *T*.

Tests were run at vacuums of 7, 15, 20, 25 and 27 inches of mercury, with the circulating water at a temperature of about 40 degrees Fahrenheit and a velocity of 8.6 feet per second on the following kinds of tubes:

Admiralty; admiralty, oxidized; admiralty, vulcanized on inside only; admiralty, vulcanized on outside only; admiralty, vulcanized on both sides; admiralty, lead lined; admiralty, old tube from condenser; monel; copper; copper, aluminum lined; aluminum bronze; zinc; tin; cupro-nickel; shelby steel; glass.

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*Abstract of paper delivered before the American Society of Mechanical Engineers.

VARIATION OF HEAT TRANSFER WITH TEMPERATURE DIFFERENCE

After the apparatus had been working some time and had been giving consistent results, a set of approximately one hundred tests was run to determine the possible variation of the heat transmission with the mean temperature difference. The quantities of cooling water were so large that the temperature rise was always small and it was considered that the arithmetical mean of the water temperatures and the steam temperature might be used without appreciable error in the calculation of the heat transfer. These tests were run at four different velocities of approximately 2, 4, 6 and 8 feet per second, the results were plotted on logarithmic paper and a smooth curve was drawn through each set of plotted points. These curves were practically straight lines, as shown in Fig. 2.

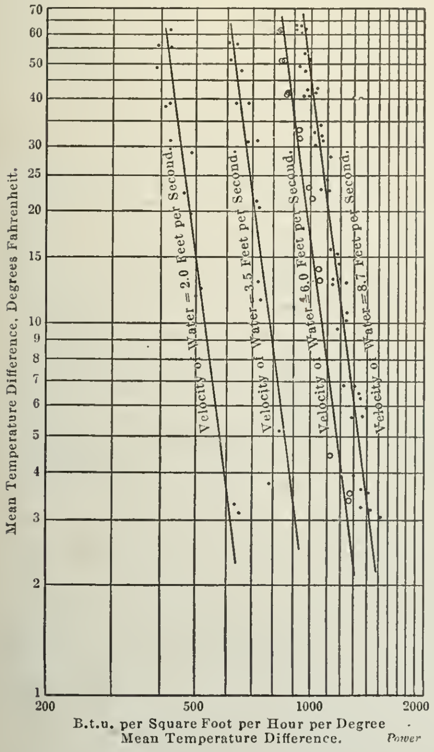

FIG. 2. RELATION OF HEAT TRANSFER TO TEMPERATURE DIFFERENCE

VARIATION OF HEAT TRANSFER WITH VELOCITY OF WATER

A series of one hundred tests were run to determine the variation of the heat transfer to the velocity of the circulating water. These were run at various velocities of from 1 to 11 feet per second and at 15- and 27-inch vacuums, the results of which are plotted in Fig. 3.

VARIATION OF HEAT TRANSFER WITH VELOCITY OF STEAM

That the velocity of the steam across the tube surface has an effect on the transmission has been asserted by a number of authorities. In the present experiments, however, no effect on the heat transmission which might be charged to the velocity of the steam approaching the tube surface was detected.

Consider a condenser consisting of a single tube, through which the cooling water is flowing, surrounded by an atmosphere of steam. Condensation will take place at the outer surface of the tube and the steam will rush in to the tube from all sides. Meanwhile, the space surrounding the tube will attain a constant pressure due to the regular influx of steam into the condenser and its condensation on the tube. This constant pressure will determine the temperature of the steam and if there is no air leakage the temperature will be that due to the saturated vapor of water at the constant pressure. It has been determined experimentally that the temperature and pressure remain sensibly constant throughout the condenser, except for an exceedingly minute space in the neighborhood of the tube surface. There is then a constant flow normal to the surface of the tube, besides such residual velocity as may be in the steam after its passage through the exhaust pipe. This residual velocity in practice is usually from 200 to 600 feet per second and in turbine installations may be as high as 800 feet per second. In direction, it follows the axis of the eduction nozzle until deflected by the tubes, sides of the condenser or baffle plates and constantly loses velocity until, at the further end of the steam travel, this velocity is entirely lost. The component of this residual velocity longitudinally to the tube is of no value in bringing a particle of steam to the condensing surface; neither is the component of this velocity normal to the tube surface, for its action is balanced by carrying away the steam on the opposite side of the tube. Its only action then is to distribute the steam among the tubes.

When a particle of steam comes in contact with the surface of the tube, it is immediately condensed and contracts enormously in volume. At atmospheric pressure the volume of water is approximately 1/1660 of the steam volume. At 28 inches of vacuum, the contraction is about 1/21,500, and at 29.5 inches of vacuum, or ¼ pound absolute, 77,000 cubic feet of steam condenses to 1 cubic foot of water. The vacuum thus created induces a flow of steam normal to the steam surface, the velocity of which, neglecting friction, can easily be calculated for various conditions by means of the formula

$$V = \sqrt{2\,gh}$$

Transforming this formula,

$$V = 96.25\sqrt{(P_t - P_a)\,\sigma}$$

where

P_t = Vacuum pressure (pounds absolute);

P_a = Partial pressure due to the air in the condenser;

σ = Specific volume of steam at pressure $(P_t - P_a)$.

The velocity with which the steam particles strike the tube surface is apparently independent of the distance they travel before striking and is reasonably independent of the vacuum. The velocity varies from about 1750 feet per second at 1 pound absolute to about 1900 feet per second at 14.7 pounds absolute. These values of V are shown in Fig. 4 and the effect of a small quantity of air in the condenser on the velocity of flow of the steam is shown to be very small.

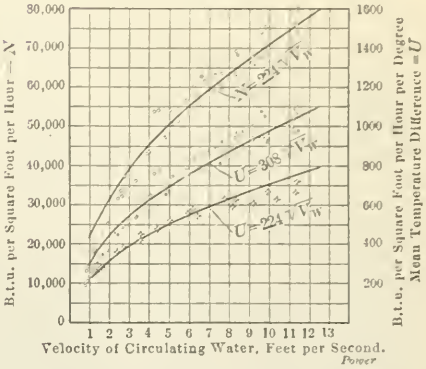

FIG. 3. VARIATION OF HEAT TRANSMISSION WITH WATER VELOCITY

THE EFFECT OF AIR UPON HEAT TRANSMISSION

During the early part of these tests it was impossible even to approximate air-free steam, so that consistent transfer results could not be obtained. With the changing of the apparatus, however, it was possible to keep the air down to a rather small quantity and the results obtained were much more consistent. It is probable in these cases that the partial air pressure never exceeded ½ inch and in most cases was below this. By close

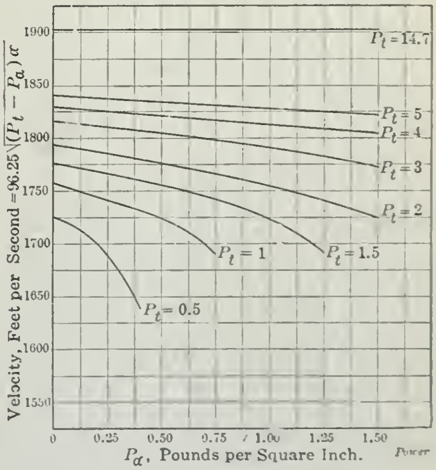

FIG. 4. VARIATION OF HEAT TRANSFER WITH STEAM VELOCITY

attention to all possible sources of air leaks in the final tests, it is probable that the partial air pressure never exceeded ¼ inch and was sometimes much less than this.

CONCLUSIONS

The heat transferred from steam surrounding a metallic tube to cold water flowing through the tube is proportional to the seven-eighths power of the mean temperature difference of the water and steam temperatures. This is equivalent to the statement that the coefficient of heat transfer is inversely proportional to the eighth root of the mean temperature difference.

The coefficient of heat transmission is approximately proportional to the square root of the velocity of the cooling water.

This coefficient is independent of the vacuum and of the velocity of the steam among the tubes or in the condenser passages. It may be proportional to the square root of the velocity normal to the tubes, but in all common cases this velocity does not vary more than a tenth part.

The effect of air upon the amount of heat transferred is very marked, particularly at high vacuums, and most of this air is due to leakage through the walls and joints of the apparatus.

Taking the heat transfer of the copper tubing as 1, under similar conditions the transfer for other materials is approximately as follows: admiralty, 0.98; aluminum lined, 0.97; admiralty oxidized (black), 0.92; aluminum bronze, 0.87; cupro-nickel, 0.80; tin, 0.79; admiralty lead-lined, 0.79; zinc, 0.75; monel metal, 0.74; shelby steel, 0.63; old admiralty (badly corroded), 0.55; admiralty vulcanized inside, 0.47; glass, 0.25; admiralty vulcanized on both sides, 0.17. This coefficient (due to the material of the tube) will be designated by μ. Corrosion, oxidation, vulcanizing, pitting, etc., have also a marked effect in reducing the transfer. This reduction, best shown by the admiralty tube which gave $\mu = 0.55$, may reduce the transfer at least 50 per cent. The foregoing conclusions may be expressed mathematically as follows:

$$U = K\frac{C\rho^5\mu\sqrt{V_w}}{\theta^{\frac{1}{8}}}$$

where

U = Coefficient of heat transmission;
C = Cleanliness coefficient varying from 1 to 0.5;
μ = Material coefficient varying from 1 to 0.17;
ρ = Ratio of the partial pressure due to the steam, to the total steam and air pressure, which varies from 1 to 0;
V_w = Water velocity in feet per second;
θ = Mean temperature difference;
K = A constant, probably about 630.

The effect of the length of tube, or rather, the length of water travel, has not been considered and the design of the condenser must be such that there is a free steam passage to every tube.

This expression for U is cumbersome to use and for modern turbine condenser work certain conditions may be taken as well settled. The guaranteed vacuum is usually 28 inches; the entering circulating water is usually 70 degrees and a 20-degree temperature rise is considered economical. Under these conditions θ = 18.3 and $\theta^{\frac{1}{8}} = 1.44$, θ being calculated on the geometrical curve as 18.2. For these cases it will be nearly as accurate and much simpler to calculate θ by the logarithmic method, neglecting θ in the denominator and using 435 or $\frac{630}{1.44}$ for K^1. The expression will then be

$$U = K^1 C P^5 \mu \sqrt{V_w}$$

This equation agrees with the results of a number of tests on full-sized condensers under varying conditions. There were no attempts to determine the amount of air handled by the air pump in these cases, but the amounts of air indicated by the formula are such as agree with the pressures and temperatures taken.

Test of a 9000 Kilowatt Turbine*

By F. H. Varney

Late in December, 1908, the Pacific Gas and Electric Company installed a 9000-kilowatt vertical turbo-generator set to operate in connection with its other stations in Oakland, consisting of one gas-engine, three steam-engine and eleven water-power plants. Beside the demand for power which the set was expected to meet, it was also thought that the regulation would be improved, which expectation was realized.

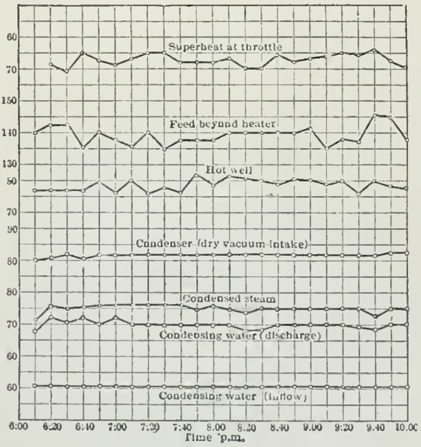

FIG. 1.

Owing to the urgent need for the plant, the efficiency test was postponed until the station had been in operation for 14 months. The conditions of the test were as follows: The turbo-generator was run on a closely regulated commercial load, which was maintained as nearly uniform as possible by the load despatcher. Two runs of four hours each were made at loads approximating 7000 kilowatts and 9000 kilowatts.

As the tests, two in number, were conducted for the purpose of determining only the efficiency of the turbine, provision was made for the observation and measurement of the water from the turbine, the total make-up and auxiliary water, the power delivered by the generator at the switchboard, the exciter output and the pressures and temperatures required to complete and interpret the observations.

The various data were plotted and several charts constructed of which Figs. 1 and 2 are charts plotted from the data obtained in the second test.

The contract specifications provided for a steam pressure at the throttle of 175 pounds gage, a superheat of 125 degrees Fahrenheit and a vacuum of 28 inches.

The guarantees were as follows:

Generator Output, Kilowatts.	Pounds Water per Kilowatt-hour.
5000	16.75
7000	16.40
9000	16.40

Variations from the specified conditions were to be subject to the following corrections:

For each 10 pounds pressure.....1 per cent.
For each 12½ pounds deg. superheat.1 per cent.
For each inch vacuum....1.08 lb. per kw.-hr.

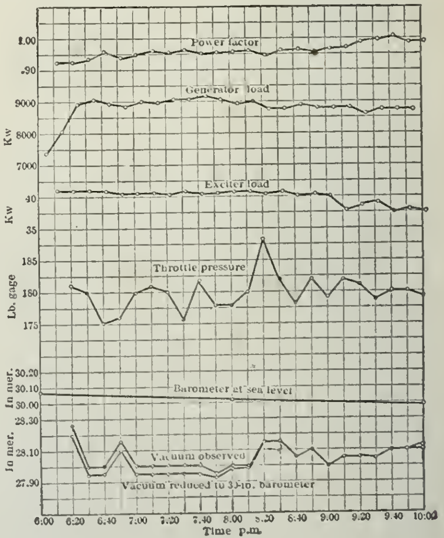

FIG. 2.

In the second test, lasting from 6 p.m. to 10 p.m., the final corrected values gave a water rate of 15.42 pounds per kilowatt-hour.

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*An abstract of a paper read before the American Society of Mechanical Engineers.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for November, 1910, *was* 166,00.

December 6............ 34,000
December 13........... 34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

The Work of the Institute

In the prospectus of the Institute of Operating Engineers attention is called to the lack of unity of purpose which results in an unsettled condition and weakness in a vocation followed by more than a quarter of a million of men in the United States. It is to assist in placing the vocation of steam engineering on the high plane of honored and respected callings which, though not wholly professional, require professional attainments of a high order if mastership is desired.

While palliative measures are discussed and ameliorative conditions sought for the various classes of employees, the operating engineer, the nature of whose work is largely solitary, is being forgotten. There is no systematic method of advancement where engineers are employed in numbers nor is there any accepted standard of proof of merit for the man who has mastered his vocation step by step and is able and ready to demonstrate his fitness to fill the position of highest responsibility in any power plant. There is, on the other hand, a constantly increasing difficulty in obtaining really skilled operating men; men with manual training, mental equipment, demeanor, tact and personal appearance fit for the positions to be filled. This difficulty is in part, if not wholly, caused by the lack of any effort to secure coöperation between the employer, the operating engineer and the ambitious subordinate worker. Sooner or later the economic relation between these factors, the bread and butter question, must be recognized, met and settled. In the methods proposed by the Institute of Operating Engineers an effort is initiated which it is expected will meet this question in a broad, clearly defined and systematic manner.

The institute itself is to be a nationwide coöperative college; and as the time has passed when a college was constituted by merely a group of buildings, a campus and a corps of instructors, the students can pursue their studies at home, at the evening school or where they will. Their correspondence with the home office and their examination papers will give their standing and mark their attainments.

Already many employers of skilled labor in many lines have recognized the importance of this new movement and have signified their appreciation by offers of coöperation with, and financial support, of the attempt to create a supply of trained men so associated that determination to be proficient in a chosen calling is the mainspring of action.

The final results achieved by the institute will be a unity of purpose to place the vocation of steam-engine operation on the high plane of skilled pursuits to which it is entitled by its importance.

Boiler Inspections and License Laws

The boiler inspector's duties are very exacting and he does not receive the hearty support and coöperation from a great many of those with whom he is brought in daily contact. When an inspector sends out a notice that he will be at a certain plant to make an internal inspection of a boiler, say, at 8:30 on a Sunday morning, and requests that the boiler be gotten ready for the inspection, the chances are that the work will not be half done. The inspector has to put up with these conditions in every State in the Union, except Massachusetts, in which State he can decline to make the inspection if the boiler is not properly prepared and will not issue a certificate until an efficient inspection has been made.

In many cases the engineer is not to blame, as frequently the notice is not sent to him personally, but to his employers and they do not notify him until it is too late to get the boiler in proper condition. In fact, sometimes they do not notify the engineer at all, and when the inspector arrives at the plant, he finds it closed, or in charge of a watchman, but no boilers ready. Sunday being a busy day for the inspector, he leaves his card with the watchman to give to the engineer or tacks it on the engine-room door, and visits the next plant on his list, where he is very likely to meet with the same reception.

Again, the inspector often finds it necessary to have repairs made, but the engineer, in many cases, has not the authority to order them and has to refer the matter to his employers, in which case the employers often consider the repairs unnecessary and will delay having them made until an accident occurs. In cases like this an inspector's life is not a happy one, for he is between two fires; he knows the repairs are actually necessary but has not the power or authority to compel the shutting down of the boiler until they are made, and if he suspends the insurance until such repairs are made, his company will probably lose all the insurance that the owners of the plant carry with them. However, the inspector should not take any risks, for by suspending the insurance he is not only protecting his company but is protecting the owners of the plant and their employees, besides his own reputation.

Every engineer should know how and should in every way assist the inspector to make a thorough examination of the boilers, and, if repairs are recommended, should have them made at once. It is an old proverb and a good one that says, "A stitch in time saves nine." In Massachusetts the engineer's and the inspector's duties are made plain by the boiler-inspection law, and it would be advisable for every man in charge of a steam plant to get a copy of that law and study it carefully.

The coöperation of all engineering organizations, and the support of all engineers who are not members of such organizations are needed to use their influence to put into effect proper license laws. If the National Association of Stationary Engineers would hold biennial sessions instead of annual sessions and devote a sum of money annually toward helping to procure license laws, it would not be many years before every State would enact laws that would prove a benefit to the owners of power plants and their operatives. The question of wages would take care of itself, for the reason that the engineer would then educate himself so as to entitle him to the highest grade of license, and not merely be the "stopper and starter" so often referred to. By such actions the national association would be living up to the full text of its preamble, which reads: "Its meetings shall be devoted to the business of the association and at all times preference shall be given to the education of engineers, and to securing the enactment of engineers' license laws in order to prevent the destruction of life and property in the generation and transmission of steam as a motive power." Nothing can be plainer than this, and the subordinate associations should demand aid and assistance from the national organization in procuring license laws.

The engineer is the most important employee in a power plant, and he should thoroughly understand his plant from A to Z. Many men now operating steam plants know nothing about the construction of the steam boilers they are operating, the thickness of the shell, thickness of the heads, the strength of the seams and the number and location of the braces; and, not knowing these few points, they cannot tell whether or not they are carrying too much pressure on the boilers, but simply carry the pressure because it is necessary for the operation of their plant. If New York and other States had laws similar to those of Massachusetts and these were properly enforced, life and property would be better protected and the engineers who are now opposed to license laws would in a short time be their strongest advocates. State Senator George L. Meade in his speech at the opening of the national convention in Rochester last September stated that a competent licensed engineer is a far better protection against boiler explosions than any system of inspections that could be devised, for the reason that he makes it a point to thoroughly examine his own boilers; also, that the granting of licenses in all the professions and trades had elevated the standard of those licensed. When we have such encouraging words from a State senator, the engineers of New York State should make an effort to have the incoming legislature enact the proper laws.

A Shaftless Pump

Like all new contrivances of simple construction, the Humphrey pump incites wonder as to why "somebody" hadn't thought of it before. This apparatus is an internal-combustion pump in which the water itself serves as a piston and its oscillations, due to kinetic energy delivered to it by the explosion of the charge, are utilized to expel the burned gases, draw in the fresh charge and compress it for the next explosion. It has no shaft or other mechanical parts, excepting a few simple valves. In this issue appears a part of a paper recently read by the inventor of this ingenious apparatus in which some important modifications are described. It is rather surprising to learn that the pump can be built to lift water from a level below the intake valves and deliver it against pretty high pressures. The modification of the simple pump by means of which this is feasible is not included in this week's section of Mr. Humphrey's paper because of space limitations; it will be described next week.

An even more surprising possibility of the pump is the alteration of the compression pressure without changing the hight to which the water is raised. In the simple pump, the hight to which the water is driven by the explosion of the charge determines the pressure to which it will compress the next charge on its second return "stroke." By merely adding a bell, projecting into the combustion chamber, the compression space, outside the bell, may be proportioned to give high compression without high cushion pressure. This arrangement also effects complete scavenging of the space into which the fresh charge is drawn.

The "combinations" of which the underlying principles of the pump are susceptible without entailing complicated construction appear to be extraordinarily numerous.

Inches or Pounds

At this time when everything relating to the generation of power and its transmission is moving or has only recently emerged from a state of transition from one base or form of nomenclature to another, it would seem opportune to establish a correct method of using some few engineering terms. If a navigator should use miles in designating distances eastward and degrees of longitude when referring to a westward direction, he would present no more incongruous spectacle than the engineer who measures pressures up to that of the atmosphere in inches of mercury reckoned backward and thence in pounds, calling that pressure one pound which is really 15.7 pounds per square inch.

Taking Unfair Advantage

When an engine meets with an accident unscrupulous builders of competing machines lose no time and spend no pains to exploit the misfortune to the discredit of the engine, and to prejudice intending purchasers against it, and this whether the engine was at fault or not. An instance of this kind was the breaking of a receiver on a McIntosh & Seymour engine at the Harvard station of the Boston Elevated Railway Company. The accident was a plain case of overpressure, as was plainly stated in our article describing it. The subjection of this receiver to a pressure several times that which it was designed to bear, was no fault of the engine or its builders. Of this we have the positive assurance of the company's engineers. Any receiver, unless of abnormal proportions, would have failed under the conditions to which this one was subjected and nothing that the engine builders could have foreseen or provided would have prevented it. Any citation of the accident to the prejudice of the engine is unfair and uncalled for.

Smoke Up and Deduct the Fine from the Coal Bill

Such an absurd ruling was recently made by Judge Hines in the Municipal Court of Chicago. It appears that the Boston Fuel Company sued the Cooke Brewing Company for $999. The company last named had been fined $100 for violating the smoke ordinance, and deducted that amount of money from its coal bill. The Boston Fuel Company, of course, sued to recover and the Jury determined that if coal produced objectionable black smoke and if the user of that coal were fined as a consequence, the amount of the fine could be deducted from the coal bill. The suit will be appealed and it is to be hoped that some of the smoke in question does not befog the minds of those who will render the verdict.

Inquiries of General Interest

Proportions of Hose Nozzles

What are the best proportions for a hose nozzle to throw a stream the farthest?

P. H. N.

In common practice, nozzles are made 6 to 10 times the hose diameter in length with the hole at the end one-third the hose diameter.

Water Leg of a Boiler

What is meant by the term "water leg" and what part of a boiler is it?

W. L. B.

The term applies to the narrow water space between the inner and outer sheets of the firebox of vertical and locomotive types of boilers.

Steam for Preventing Clinkers

I have been told that a jet of steam under the grates will prevent the formation of clinkers. Is this true, and is it an economical practice?

S. P. C.

Clinkers are caused by the melting and running together of the incombustible in the coal by the heat of the fire. If steam enough is passed through the fire to keep the temperature below the melting point of the ash, clinkers will not form. It is certain that there is no economy in using steam to reduce the temperature of the fire under the boiler which makes the steam.

Horsepower and Boilers

What is the correct number of square feet of heating surface to be reckoned for a horsepower in different types of boilers?

H. P. B.

Ten square feet of heating surface per horsepower has always been the standard used by the makers of water-tube boilers, and from 12 to 15 the standard of the makers of other types. But recently the opinion has prevailed that the heating surface in other types is as efficient as that in the water tube and 10 square feet of heating surface per horsepower is now the rating for all types of boilers.

Two Hundred-Horsepower Horizontal Boiler

Are horizontal return-tubular boilers of 200 or more horsepower in successful use anywhere, and what would be the dimensions of one of 200 horsepower?

H. H. B.

They are made and used up to 300 horsepower. One of 200 horsepower would be 7 feet in diameter, containing one hundred and twenty-four 3½-inch tubes, 18 feet long, and for 125 pounds pressure be made of ⅝-inch plates with butt and strap joints quadruple riveted.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Diameter of Air Compressor Valves

I find a difficulty in determining the proper diameters of air-compressor valves of the poppet type and should be obliged if you would give me what is considered good practice. Some say the velocity of air through the valves should not exceed 2000 to 3000 feet per minute; others say that 5000 to 6000 can be taken with safety.

A. C. V.

Modern practice for piston speeds of 500 to 700 feet per minute dictates that the opening through the discharge valve shall be from 15 to 25 per cent. of the piston area. Inlet-valve passages should be as great if possible. While the volume of air after compression is less than at admission the time of discharge is much less than that of intake, causing an increase in speed of discharge. It is not necessary that the peripheral area under the lip of the open poppet valve should be as great as the port area through the seat, as the restriction at valve lip seems to be so momentary that full lift is not required.

External Corrosion of Boilers

What are the most common causes of external corrosion of boilers?

E. C. B.

Leaking seams, rivets and gaskets, also wet ashes and soot if permitted to remain in contact with the plate.

Underground Pipe Covering

I have two lines of steam pipe each 400 feet long laid underground in a wooden box packed with shavings. This method is unsatisfactory for the shavings gather moisture and the pipes corrode on the outside. How would it do to lay them in tile?

W. P. C.

The same care should be exercised in laying underground steam pipes as where run overhead. They must be well insulated and kept dry. A perfectly drained trench with the pipes run above the possibility of contact with water only will give satisfactory results.

Steam per Horsepower Hour

What is the average number of pounds of steam required per horsepower per hour by simple automatic cutoff engines of 100 horsepower and by compound-condensing engines of 500 horsepower?

S. H. H.

For the simple engine 25 to 30 pounds, and for the compound-condensing about one-half as much.

Temperature of Red Hot Iron

What is the temperature of iron at a dark red heat?

T. H. I.

Iron shows red at a temperature of 700 degrees in the dark; in daylight at 1000 to 1300 degrees, according to the light.

Steam Pipes and Mains

I have two 17x22-inch engines running 175 revolutions per minute. What should be the size of the steam pipes and the main which supplies them?

S. P. M.

A formula for finding the diameter of the steam pipe for an engine reads:

$$Diameter = \sqrt{\frac{HP}{6}}$$

A 17-inch engine should develop 175 horsepower and substituting this value in the equation, it reads:

$$\sqrt{\frac{175}{6}} = 5.4 \textit{ inches}$$

the diameter of a pipe necessary to supply one engine. To supply two engines the pipe must have a carrying capacity of two pipes, each 5.4 inches in diameter. To find the diameter of one pipe which with the same head will have a carrying capacity equal to two smaller ones of the same size, multiply the diameter of the smaller by 1.3;

$$5.4 \times 1.3 = 7.02 \textit{ inches}$$

the diameter of steam pipe required to supply two 17-inch engines. In selecting the pipes the nearest commercial size should be taken.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Impulse Reaction Turbine

Messrs. Willans & Robinson, Ltd., and E. G. Izod, of Victoria Works, Rugby, England, have recently evolved and patented a combined impulse and reaction turbine which can be operated with either low-pressure steam, high-pressure steam or a combination of both low- and high-pressure steam.

The following information was obtained from a description in a recent number of *The Mechanical Engineer*, London:

The impulse wheel is made of larger diameter than the end of the drum or reaction portion of the turbine which is adjacent to it, and the portion of the outer casing within which it is inclosed is similarly made of larger diameter than the portion of the casing which surrounds this end of the drum. The high-pressure nozzles for delivering steam to the impulse blades pass through one side of the enlarged portion of the casing which surrounds the outer portion of this wheel, while an inlet for low-pressure steam is made through the opposite side of this portion of the casing. The steam passage leading to this low-pressure steam inlet is led back alongside of the high-pressure steam passage, and a stop valve is so arranged in each that both may be operated by the movement of one lever extending from the main governor of the turbine. Stop valves in the high-pressure and low-pressure steam passages are also provided to allow of the turbine being operated either by high-pressure steam alone, or by low-pressure steam alone, or by a combination of high- and low-pressure steam.

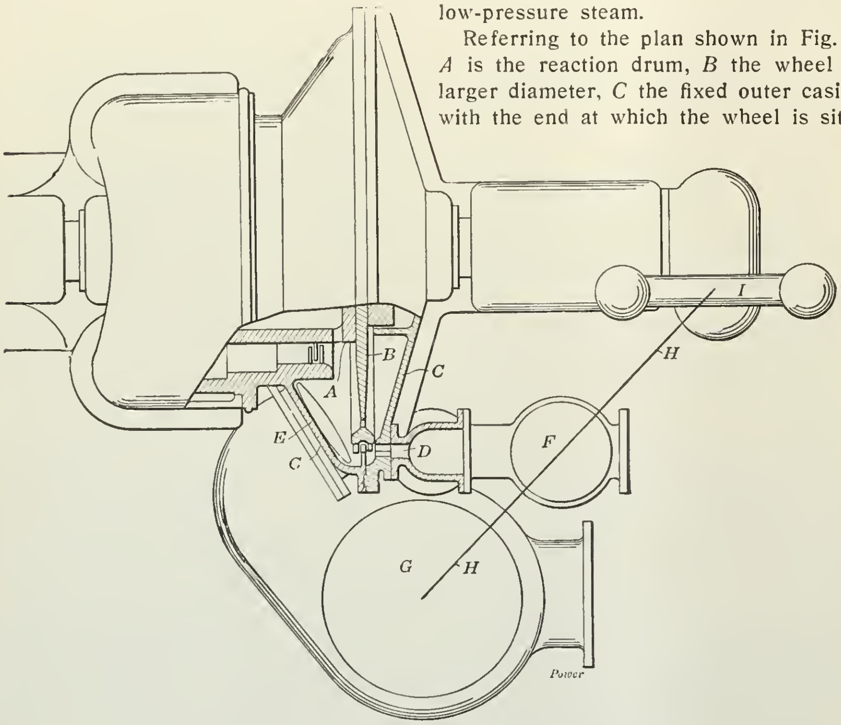


Fig. 1. Combined Impulse and Reaction Turbine

Referring to the plan shown in Fig. 1, *A* is the reaction drum, *B* the wheel of larger diameter, *C* the fixed outer casing with the end at which the wheel is situated correspondingly enlarged in diameter, *D* a nozzle passed through one side of the enlarged portion of the casing for directing high-pressure steam against the blades of the impulse wheel, *E* a port formed through the opposite side of this enlarged part of the casing for admitting steam of low pressure to the space between the impulse wheel and reaction drum, *F* a valve on the high-pressure steam-supply pipe, *G* a valve on the low-pressure steam-supply pipe and *H* a lever operated by a governor at *I*, by which movement may be given simultaneously to both valves.

The sectional elevation, Fig. 2, shows the means employed for operating the two valves from the governor lever *H*. This lever is mounted on a fulcrum *J*, and from one end of it the low-pressure valve *G* is acted on as shown, while the high-pressure valve *F* coupled to a rod *K* is connected to lever *H* by means of a regulating lever *L* mounted on fulcrum *M*, the motion from the main lever *H* being imparted by means of stops *N* and *O* which are fixed to lever *H*, the top stop *O* being adjustable.

The spring *P* bearing against a collar on rod *K* and pressing against spring case *Q* always tends to close the valve *F*. The spring *R* being a compensating spring allows the low-pressure valve *G* to have a slight lead, and enables both valves to be tightly closed on their respective seats.

By regulating the adjustment *S* on the low-pressure gear and the stop *O* on the high-pressure gear, it is possible to run the turbine under the following conditions: As a high- or low-pressure turbine; or as a mixed pressure; and the high- or low-pressure steam may be supplied or cut off at any given predetermined load by means of these adjustments.

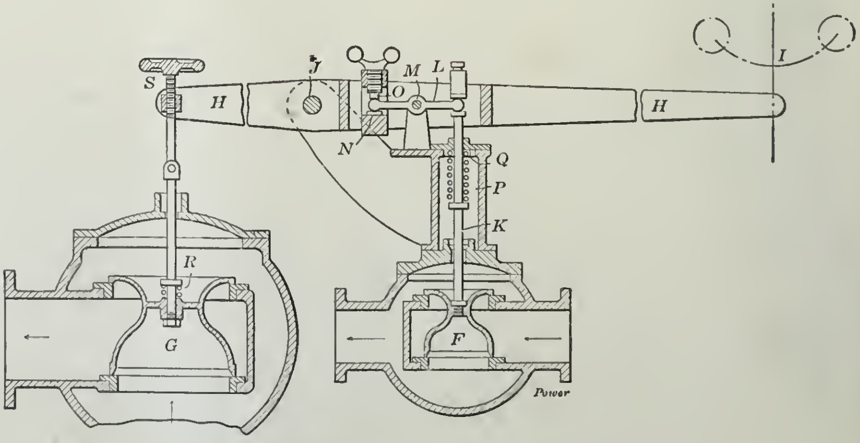


Fig. 2. Valve Arrangement of the Combined Turbine

The Heath Valvograph

W. F. Stanley & Co., Ltd., of Great Turnstile, London, have recently placed on the market a simple and accurate instrument by means of which it is possible to read from, or construct, valve diagrams, and solve with the least possible construction and without calculation the various problems which arise in designing a new valve or when making alterations to an existing one. It is adaptable to all valves where the Zeuner diagram can be used, and when once the principle of the instrument is understood, anyone may become proficient in its use with very little practice. In the hands of an experienced engineer it will also be found most useful and time saving, while to the student it will afford great assistance, enabling him to study more closely than he could by any other means the effects produced by modification of the steam distribution.

The instrument, according to *The Mechanical Engineer*, of London, is made in transparent celluloid, and is marked as shown in Fig. 1 with the following: An advance line *A B* giving position of the eccentric, and also readings for lap and port opening; a cutoff and admission line *C D;* and lead lines *E*. These are four in number and give lead by sixteenths up to ¼ inch; the lines *E'* and *F* are for use when determining the valve travel. Divisions *K* form an ordinary protractor for use when measuring the angle of advance; and a special protractor *J*, divided to give cutoffs in fractions of the stroke, is also given.

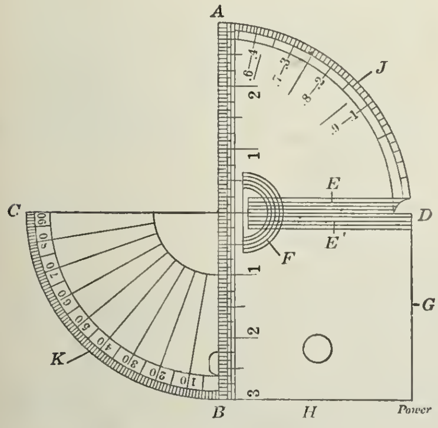


Fig. 1. Heath's Valvograph

To use this when the cutoff falls on the right-hand side of the diagram, the instrument is placed in a vertical position with its axes *A B* and *C D* coinciding with those of the diagram, when a line drawn from the center through the required cutoff on the protractor will intersect the valve-travel circle at the correct point. If, however, the cutoff should fall on the left-hand side of the diagram, the instrument is moved round so that the required cutoff on the protractor coincides with the horizontal axis of the diagram; then a line drawn from the center to *C* will give the cutoff as before. To aid rapid setting, gage lines are marked on this protractor giving a cutoff advancing by eighths in place of hundredths, and two square edges *G* and *H* allow the instrument to be moved to any part of the diagram without alteration of the angle at which it has been set.

Referring to Fig. 1, it will be seen that the instrument is divided to read lap, lead and port opening to 1/16 inch, point of cutoff to one-hundredth of the stroke and angle of advance to one degree; but, when required, finer readings than these may be made by estimation.

By placing the instrument over any valve diagram, as indicated in Fig. 2, it will be noticed that the advance line *A B* coincides with the angle of advance, the cutoff line *C D* intersects the valve-travel circle at points of cutoff and admission, and is tangent to the lap circle, the lap and port opening being read on the scales marked on the advance line, while the lead (if any) is indicated by the particular lead line *E*, which cuts the valve-travel circle at the point where it is intersected by its horizontal axis. From this it is apparent that in whatever position the instrument may be placed over a circle which represents the valve travel, provided *A B* passes through the center of that circle, lap, port opening, angle of advance, points of cutoff and admission and lead will always remain in correct relation to one another. Thus, in setting down a diagram, the rectangular axes *L M* and *N O* are first drawn, and then with *X* as center a circle having a radius equal to half the valve travel is described. The instrument is now placed over this figure so that it registers any two known dimensions, when it will be set for all other readings. While thus set, with a finely pointed pencil, lines *A B* and *C D* are drawn, when the diagram will be complete for the steam side, and all readings may be taken from it by means of the scales and protractors provided. When the cutoff is known, this should first be marked off as already described, and used as one of the points for setting.

To construct the exhaust side of the diagram, it is only necessary to place the instrument so that *A B* falls on the angle of advance and the zero point of the two scales marked on the advance line falls below the center *X* to the extent of the exhaust or inside lap, if any; then *C D* will cut the valve-travel circle at the points of release and compression. Line *C D* is then drawn in as before.

In cases where the valve travel is unknown, but the cutoff lead and port opening are given, the travel may be determined in the following manner: The axes *L M* and *N O* are drawn, and with *X* as center a circle is described (lightly), having a radius equal to the port opening; also, by aid of the cutoff protractor, the cutoff line *X P* is set up. The instrument is now placed in such a manner that the smallest arc of the series *F* touches the line *X P* and the arc which is a distance from the smallest one equal to the desired amount of lead touches the horizontal axis *L M*, the line *C D* being within the angle *P X L*, and the center *X* being cut by the line *E'*, which equals half the lead; that is, if the lead happened to be ⅛ inch, the arc which incloses two spaces of ⅛ inch would be used in conjunction with the line which is one space or ¹⁄₁₆ below *C D*. A line is now marked along the exposed portion of *C D*, and the instrument is then moved by aid of the square edge *H* till *A B* is tangent to the port-opening circle. With the exposed portion of *A B* this tangent is drawn to cut the line *X P*, and the angle so formed is bisected by letting any one of the arcs *F* touch these two lines while *C D* passes through the point where they intersect. A line is now drawn in as before, with the exposed portion of *C D*, and the point where it cuts the first line will be a distance from the center *X* equal to half the valve travel. The valve-travel circle is then drawn with this radius, and the diagram completed as already described.

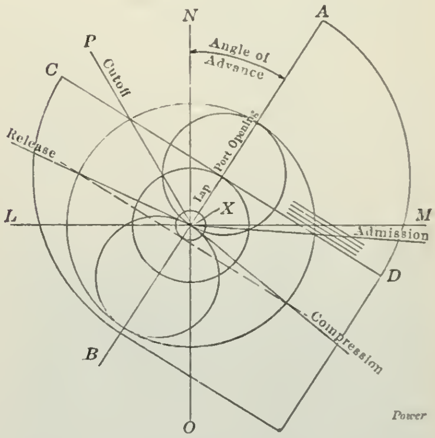


Fig. 2. Instrument Placed over Valve Diagram

When the valve travel exceeds 6 inches, the diagram should be drawn half size, and all readings, except those of the two protractors, doubled. Should the diagram be of the opposite hand to that shown in Fig. 2 the instrument is inverted, and used as already described, only using lines *E'* as lead lines and line *E* for determination of valve travel. It is always a good plan to set down the diagram on some corner of the valve drawing, as it forms a handy record which may be consulted at any future time, and when desired the lap and valve circles may be added, and the diagram made quite complete.

Automatic Water Jet Eductor

The Koerting automatic water-jet eductor is used for automatically pumping water. A sectional view of the eductor proper is shown in Fig. 1. It operates as follows:

A jet of pressure water passing from a narrow into a wider nozzle thereby creates a vacuum. This causes the water to be transported to rise in the supply pipe and to pass mixed with the pressure water through the wider nozzle to the discharge.

The installation of this eductor simply consists in making a few pipe connections, and in starting the motive power, i.e., the pressure water. The pressure in pounds per square inch at the eductor should be 2½ times the elevation in feet. So, for instance, a pressure of 25 pounds is desirable for an elevation of 10 feet. The eductor takes water of any temperature up to the boiling point, and is built for an hourly capacity of from 125 to 20,000 gallons.

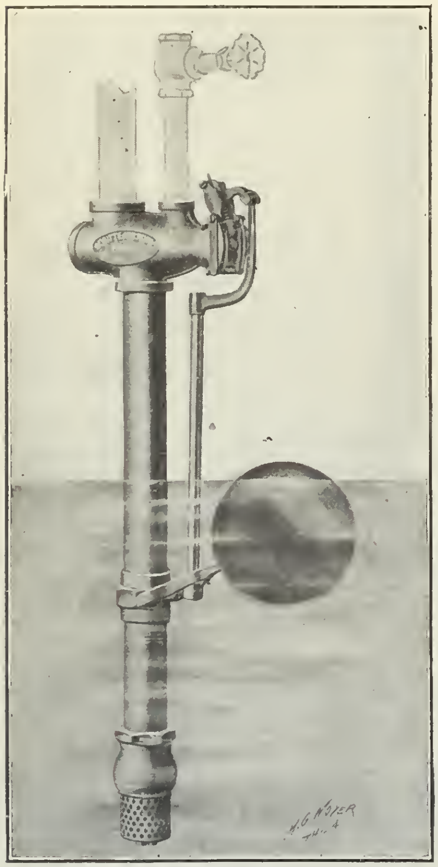


Fig. 1. Automatic Jet Eductor

The water-jet eductor is used for removing water from pits and quarries, for emptying foundation pits and gas-holder tanks, for quenching coke, for discharging ashes from stationary or marine boiler rooms to a convenient location, for washing the filter sand in water works and to sink mining shafts.

The special feature of the eductor, Fig. 2, is the float which starts the operation automatically whenever there is water in the pit to be emptied, turning on the pressure water full, and stopping as soon as all the water is removed. The eductor is installed in a wooden or cement barrel. It is made by the Schutte & Koerting Company, Thompson and Twelfth streets, Philadelphia, Penn.

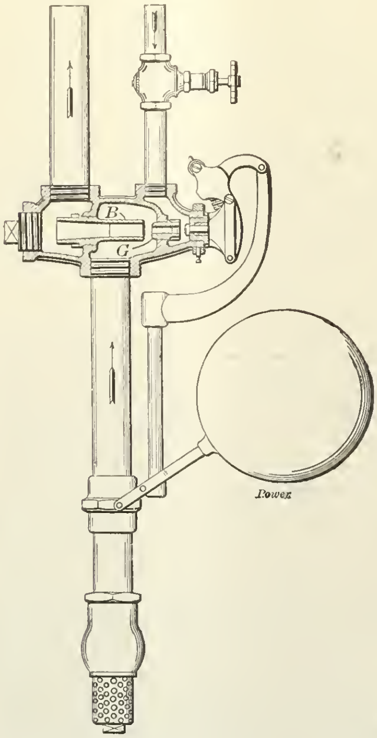
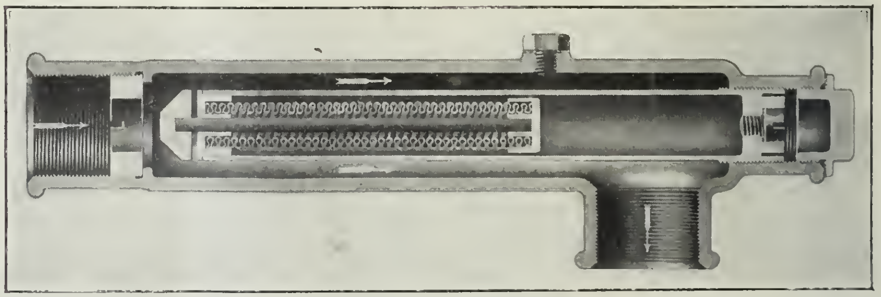


Fig. 2. Sectional View of Operating Mechanism

Sarco Steam Trap

The apparatus illustrated herewith consists of a valve with a conical top, which derives its motion from the expansion of a fluid that is extremely sensitive to fluctuations in temperature.



Sectional View of Sarco Steam Trap

The fluid is contained in a hermetically sealed tube and the whole fitted to a casing suitable for direct connection into the steam line. While this is essentially a low-pressure trap, it can be used for high-pressure work. The steam, however, must flow in the direction of the arrow. For low pressures it makes no difference in what direction the steam passes through the trap, although it is better to have the flow in the same direction as for high-pressure work.

The trap can be installed in any position desired. It is easy to clean without disconnecting by simply blowing steam throught it. There can be no freezing of the steam pipes as the trap opens automatically with the cooling of condensation and stays open until steam enters the trap, when the valve closes. All condensation is ejected as fast as it forms so that the line is kept clear. The inner cartridge can be replaced at small cost, should it deteriorate, and without taking the trap off the line.

To place the trap in operation it is necessary to connect it with a pipe and then unscrew the cartridge or inner tube about ¾ inch and allow steam to flow through freely for about one minute. The expansion of the chemical fluid in the cartridge will cause the piston to advance and rest against the valve seat. After this the cartridge is screwed up against the seat until steam no longer escapes. The trap is then ready for use and operates as previously described.

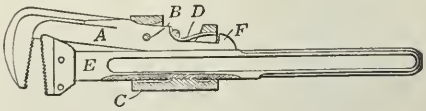
This trap is manufactured by the Sarco Fuel Saving and Engineering Company, West Street building, New York City.

Wright Pipe Wrench

The line cut shows a new pipe wrench which is being put on the market by J. F. Wright, Canton, O.

The jaw *A* pivots on the pin *B* in the yoke *C*, which is held against serrations on the under side of the stationary jaw *E* by the spring *D*. While the pin *B* permits the rocking of the jaw *A* in yoke, it does not take any of the strain. This is taken by the lug *F* which is formed on the end of *A* and engages with the yoke.

It is a "one-hand" tool and easily operated.



The Wright Pipe Wrench

Morehead Condenser Trap

The new Morehead condenser trap has been especially designed and is employed for use on the return line of vacuum heating systems. when a difference in pressure is maintained between ends of the system, the aim being to relieve any steam-heating apparatus of air and water of condension contained therein.

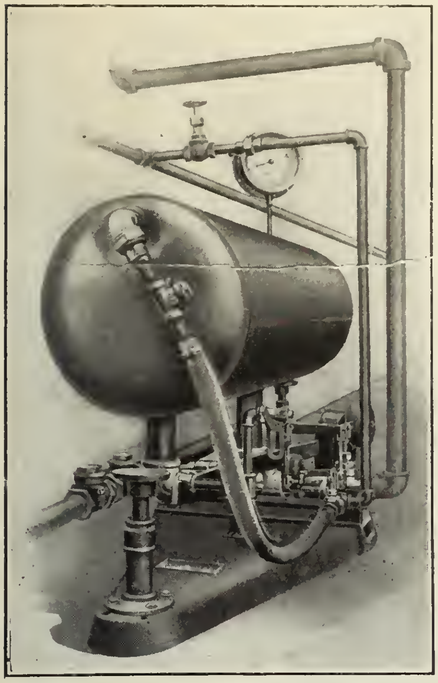


Fig. 1. Morehead Condenser Trap

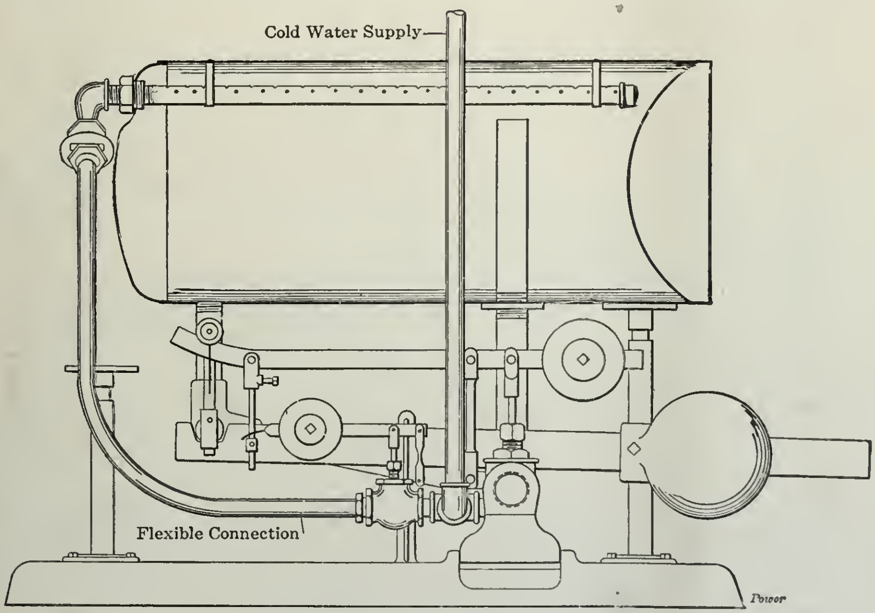


Fig. 2. Section through Trap

The principle of operation of this condenser trap is extremely simple and similar to that of the return-steam trap of the same make.

A cold-water connection is made through a flexible tube into the tank of the trap, as shown in the illustration. The flexible connection terminates in a perforated pipe inside, and running the entire length of the tank or condensing chamber. An automatically operated valve regulates the intermittent supply of water to the condensing chamber, the cold water striking the steam from the heating system is condensed and thus creates a vacuum. A surprisingly small amount of water is required in the condensing process.

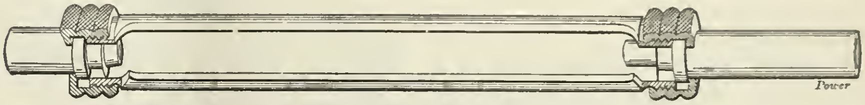
This trap is manufactured by the Morehead Manufacturing Company, Detroit, Mich.

The Leonhardt Safety Water Glass Shield

The accompanying illustration shows the method of application of this shield to a water glass. The shield consists of a metallic sleeve, slotted so that the water level may be seen, and capped at each end with two nurled nuts which form the connection to the water column by means of a rubber gasket and a short section of metallic tubing. The shield prevents glass from flying about the boiler room and injuring the employees in case the gage glass breaks. It also acts as a mechanical protection to the glass and prevents breakage when the glass is being placed in position on the water column.

The sleeve at each end extends into the ends of the glass through the gasket and assists in protecting the glass from the condensed steam which drips from the end of the sleeve instead of running down the inside of the glass tube. The gasket also protects the ends of the glass from being eaten by the action of the water in the glass.

This appliance is made by the Leonhardt Safety Water and Lubricator Glass Shield Company, 167 Dearborn street, Chicago, Ill.



Safety Water Glass Shield

Federal Boiler Inspection Bill

There is a bill before the United States Senate which provides for Federal inspection of the boilers on the locomotives of all common carriers engaged in interstate transportation.

The bill provides for the appointment by the President of an inspector-general and two assistant inspectors-general of locomotive boilers. The area comprised by the various States and Territories is to be divided into 100 inspection districts, each to be covered by an inspector who shall be appointed after competitive examination under the Civil Service rules.

If the bill is passed, it will become effective June 1, 1911.

Demonstration by Lantern Slide

Siemens Brothers Dynamo Works, Ltd., of London, have adopted a practice which might, with advantage, be followed by American manufacturers. They have a collection of one hundred and seventy different lantern slides illustrative of their product and its application which are loaned free of charge for lectures, demonstrations, etc. Many societies have lanterns in their meeting rooms and there is no doubt that if such collections of slides were available they would be very much in demand and used to the mutual advantage of the users and the manufacturers of the apparatus illustrated.

SOCIETY NOTES

The United Steam Engineers of Detroit, which organization is composed of National Association of Stationary Engineers' associations Nos. 1 and 7, International Union of Steam Engineers Nos. 5 and 324, Universal Craftsman's Council of Engineers No. 2, Marine Engineers Beneficial Association No. 4, with about 1400 members in all, have taken up with the University of Michigan the possibility of having a resident instructor at Detroit to devote his time to the organizations, giving lectures and engineering instruction at the association headquarters. On October 23 a meeting of the educational committee of the United Steam Engineers was held, at which the entire faculty of the University of Michigan, engineering department, were present, and an important step was taken toward promoting the extension work of the university along the lines suggested above.

PERSONAL

Frank Salomon, formerly chief engineer of Alfred H. Schütte, with headquarters at Cologne, Germany, has accepted an important position with the Otto Gas Engine Works, Philadelphia, Penn.

H. B. Prout, formerly with the Westinghouse Machine Company, in the turbine department, has recently allied himself with the Turbine Equipment Company, 30 Church street, New York City.

NEW PUBLICATIONS

The National Electric Light Association is now issuing to its members the annual *Proceedings*. These two volumes include the report of the thirty-third convention held at St. Louis last May, and mark the twenty-fifth anniversary of the association. The volumes include 2070 pages and 483 illustrations. Among the latter are a fine portrait of President Frueauff and several elaborate colored engravings in the paper on "Street Arcs," by Mr. Ryan. The *Proceedings* embrace some seventy papers and committee reports, all of which are carefully indexed and an abstract is given of each paper or report for ready references. A year ago the edition was 4500 sets. This year, owing to the rapid increase in membership, the edition is not less than 7000 sets, and it is interesting to know that these represent about twenty-eight tons of printed matter.

"The Institute of Operating Engineers" is the title of a 6x9-inch pamphlet setting forth the reasons for the organization of the institute, giving a list of the temporary officers and committees and a draft of the proposed constitution.

In the text there is a great deal of interesting information relating to membership, initiation fees, annual review, central and district organization, executive committee and council, proposed methods of instruction and training of junior members and apprentices, also extracts from the editorials of the leading engineers' periodicals. A copy should be in the hands of every operating engineer in the country. It is furnished on request by M. W. Rice, temporary secretary, 29 West Thirty-ninth street, New York City.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. William Johnson, Montagne, Mich., assignor of one-half to George H. Nelson, Whitehall, Mich. 976,180.

MOTOR. Carl W. Weiss, New York, N. Y. 976,236.

TURBINE. Charles E. Hicks, Mechanicsville, N. Y. 976,260.

ROTARY ENGINE. Lee O. Gilliland, Chicago, Ill. assignor of one-half to Reinhold Reichardt, Chicago, Ill. 976,355.

ROTARY ENGINE. Charles Andrew Bender, Harrisburg, Penn. 976,539.

POWER GENERATOR. Leander J. Cavanaugh and Edward T. Young, Boston, Mass. 976,547.

GAS ENGINE. Edgar F. Prall, New York, N. Y. 976,691.

MOTOR. Charles L. Wilkins, Columbus, Ohio. 976,763.

INTERNAL COMBUSTION ENGINE. Peter M. Allmon, Parker, Ind. 976,770.

ENGINE. Peter W. Kane, Detroit, Mich. 976,808.

BOILERS, FURNACES AND GAS PRODUCERS

LIQUID-FUEL BURNER. William Scrimgeour, Portsmouth, Va., assignor to Mires Fuel-Oil Equipment Company, Norfolk, Va. 976,221.

LIQUID-FUEL BURNER. Benton Moore, Cherryvale, Kan. 976,592.

LIQUID-FUEL BURNER. William R. McHill, Moran, Kan., assignor of one-third to James H. Cramer, Moran, Kan. 976,820.

POWER-PLANT AUXILIARIES AND APPLIANCES

VALVE. Benjamin F. Silliman, Cleveland, Ohio, assignor to the Cleveland Steel Tool Company, Cleveland, Ohio, a Corporation of Ohio. 976,222.

CONDENSER SYSTEM. Louis R. Alberger, Greenwich, Conn., assignor to Alberger Condenser Company, New York, N. Y., a Corporation of New York. 976,245.

CENTRIFUGAL PUMP. David Wesley Blair, New York, N. Y. 976,340.

PACKING FOR STEAM TURBINES. Gustav Huhn, Berlin, Germany. 976,369.

CENTRIFUGAL PUMP. Rudolf Salzer, Trenton, N. J., assignor to the De Laval Steam Turbine Company. 976,400.

VALVE. William S. Fairhurst, New York, N. Y. 976,449.

VALVE. Johnston Nolan, Philadelphia, Penn. 976,489.

PISTON PACKING. Robert A. Fowden, Philadelphia, Penn. 976,563.

LUBRICATOR. John Andrew Coppers and Eldredge Early Booth, Denison, Texas. 976,637.

VALVE. Guido Schuster, Beszlereze, Austria-Hungary. 976,702.

VALVE REGULATOR. Thomas M. Wilkins, East Randolph, N. Y., assignor to Ziermore Regulator Company, Johnsonburg, Penn., a Corporation of Pennsylvania. 976,713.

AUTOMATIC CUTOFF. William C. Coleman, Wichita, Kan. 976,724.

ENGINE VALVE. Rudolf Wintzer, Milwaukee, Wis., assignor to Power and Mining Machinery Company, New York, N. Y., a Corporation of New Jersey. 976,766.

SPARKING PLUG FOR INTERNAL COMBUSTION ENGINE. Robert Frederick Hall, Moseley, Birmingham, England. 976,797.

VALVE. James H. Hulings, Parsons, W. Va., assignor of one-half to Wade H. Miller, Parsons, W. Va. 976,804.

CONTAINER FOR LUBRICANTS. Adolf Nemecek, Josefstadt, Austria-Hungary. 976,825.

ELECTRICAL INVENTIONS AND APPLICATIONS

DYNAMO ELECTRIC MACHINE. Robert B. Williamson, Milwaukee, Wis., assignor to Allis-Chalmers Company, a Corporation of New Jersey. 976,420.

ELECTRIC SWITCH. John Hastings Wooll, San Francisco, Cal. 976,422.

SWITCH. Herbery W. Cheney, Norwood, Ohio, assignor to Allis-Chalmers Company, a Corporation of New Jersey, and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 976,549.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Power
and The Engineer

Vol. 32 NEW YORK, DECEMBER 20, 1910 No. 51

IMAGINE a case like this: A young man is given a good education which includes a thorough training in all the branches of mathematics. After acquiring this, his activities are such that he has absolutely no occasion to make any use whatever of his mathematical equipment for, say, fifteen years. Then suddenly he is called upon to apply some of the more advanced portions of the training which he had received.

How well do you suppose he would succeed? Remember, he has not used his knowledge for fifteen years!

Would it be reasonable to expect that he would do other than fall down hard?

The odds are a thousand to one that he would fail.

Again, take the case of an athlete—a runner, for instance. After several months of training, say he is able to get himself into condition to negotiate a mile in four and a half minutes. Then, he goes out of training and does not do any running for two or three months.

Put him in a "real" race, and he would make a fool of himself.

In steam engineering—in fact, in any line of endeavor the same causes produce the same effects.

Inaction is death!

The minute that you begin to stand still, your power and value begin to go back.

* * * * *

What the world wants is not so much to be informed as to be reminded. Therefore, this time, be reminded of some of the lessons taught by that venerable allegory concerning the hare and the tortoise.

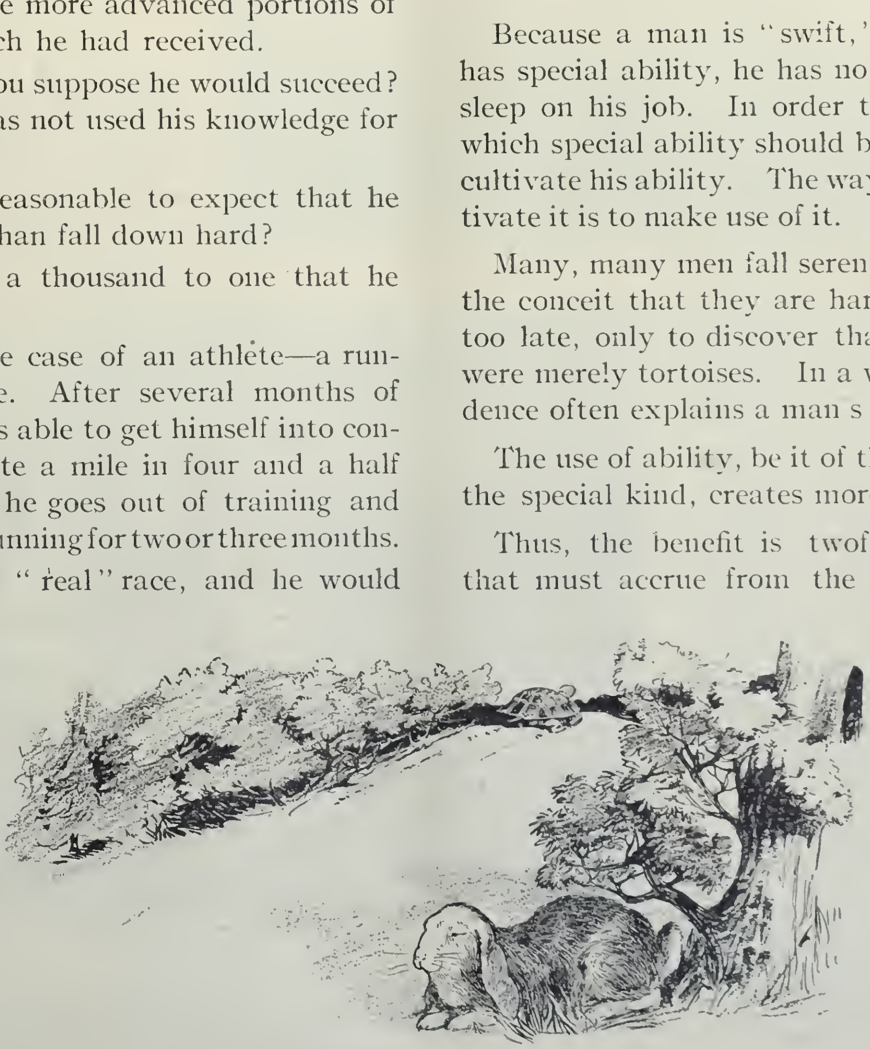
Because a man is "swift," that is to say, has special ability, he has no license to go to sleep on his job. In order to reap the fruit which special ability should bear, a man must cultivate his ability. The way in which to cultivate it is to make use of it.

Many, many men fall serenely to sleep with the conceit that they are hares and wake up too late, only to discover that after all they were merely tortoises. In a word, over confidence often explains a man s lack of success.

The use of ability, be it of the ordinary or of the special kind, creates more ability.

Thus, the benefit is twofold: the profits that must accrue from the employment of the ability already possessed, and the acquisition of more ability.

To open the way for more opportunities, make the right use of those which you now possess.



Western Union Plant in Chicago

By Osborn Monnett

A plant furnishing power for a system of independent electric drive in connection with all the machines of a large printing establishment. One of the most interesting features of the installation is the coal-handling apparatus.

The Western Newspaper Union, of Chicago, which does a general printing business on a large scale, has lately moved into its new building at the corner of Clinton and Adams streets. This building is of modern fireproof construction and contains its own power plant which furnishes power for the work done by the company, motor drive being installed on all presses and other machines. The general layout of the plant is shown in Fig. 5. Referring to this, it will be noticed that there is an unusually large coal-storage capacity, this space being an areaway under the alley at the rear of the building. Provision for the storage of approximately 350 tons of coal is thus made without taking up any ground space of the building itself.

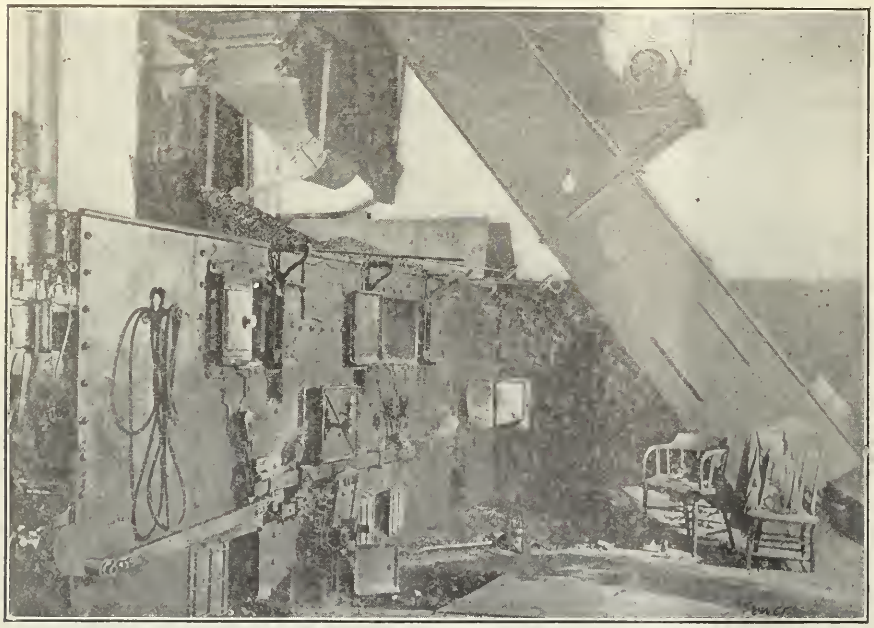


Fig. 1. Front of Battery, Showing Conveyer

Boiler-room Equipment

The boiler room contains three boilers of the Atlas type, two of which are installed in a battery and one in a single setting. Each contains 1740 square feet of heating surface and is fitted with a Model stoker, having a grate 5 feet 6 inches by 6 feet, giving a ratio of grate to heating surface of 1 to 53. Although somewhat unusual in a plant of this size, a complete coal-handling apparatus has been installed. The conveyer, as shown in Fig. 1, is of the Jeffrey type, consisting of a short covered motor-driven bucket elevator, into which the coal is dumped on a level with the boiler-room floor and discharged into a traveling weighing hopper, distributing the coal to any one of the three stokers. It is capable of delivering coal at the rate of 15 tons per hour, each hopper holding one ton. Of course, ordinarily the coal will not be used at anywhere near the maximum rating of the bucket elevator, but this rate becomes of value in case coal is needed quickly at any one point; therefore, the rating of the machine bears no special relation to the amount of coal that is burned per hour.

The ashes are dumped from the ash-pits into a telescopic ash hoist by means of which they are deposited into wagons in the alley overhead.

In connection with the stokers, which are driven by small steam engines, there has been installed a Spencer damper regulator, arranged not only to operate the dampers but also to control the speed of the stoker drive and thus regulate the feeding of the coal according to the load requirements. This permits the operation of the plant under practically smokeless conditions regardless of the load. The gases discharge into a breeching back of the boilers through which they pass to the base of the stack, which is 5 feet in diameter and 150 feet high. Vulcan soot blowers make it possible to clean the soot from the tubes by opening a steam valve on each boiler. On a recent test a boiler and furnace efficiency of 70.29 per cent. was obtained when using a coal having a heat value of 11,150 B.t.u.

Pumps

In the boiler room there are two duplex pumps and one motor-driven triplex pump, the latter being used to handle the feed water during the summer. Either of the duplex pumps may also be used for this purpose, and, in fact, the three pumps, as shown in Fig. 2, are interconnected so that any one may serve as a feed pump or house pump. From the pumps an extra-heavy pipe is run to a manifold at the side of one of the boilers, from which point the feed is controlled.

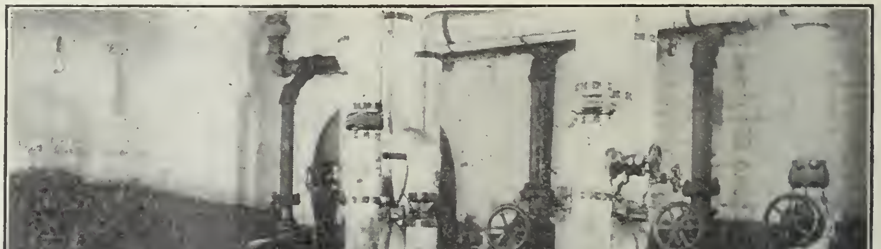


Fig. 2. Feed Pumps

Connecting with this manifold is also a pipe from the city-water supply so that any boiler may be filled conveniently after being washed out. Two permanent feed-water weighing tanks make possible at any time a determination of the evaporation of any or all of the boilers.

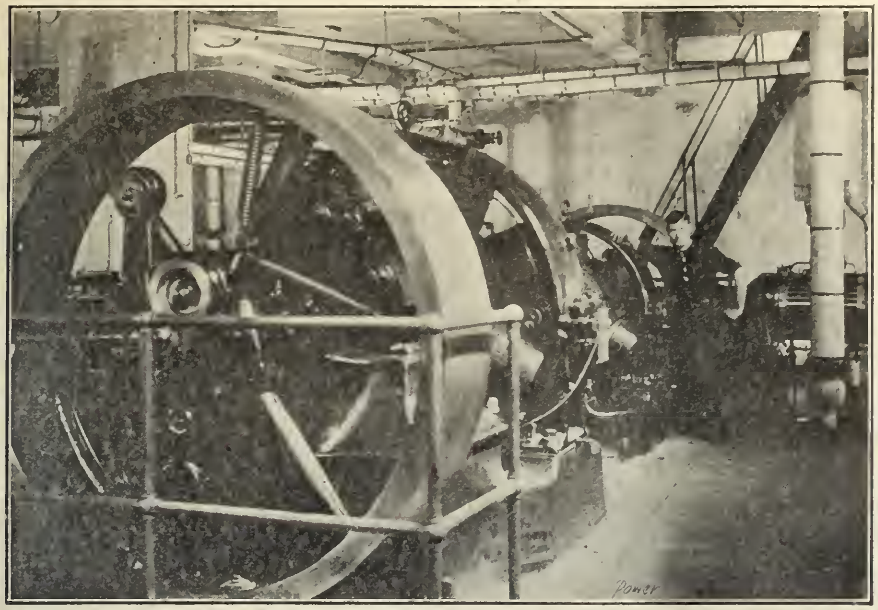


Fig. 3. View of Small Units

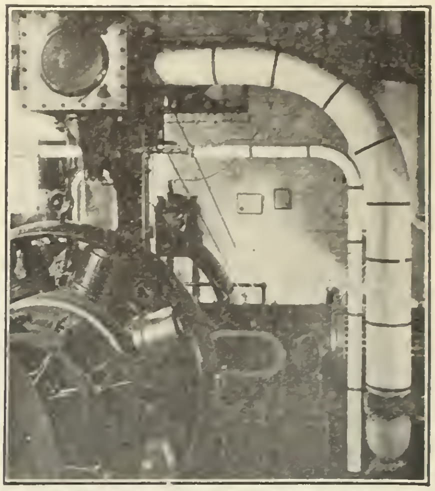


Fig. 4. Angle-compound Unit

Piping

From the boiler nozzles the steam passes into 6-inch Lagonda automatic stop and check valves, thence into the steam header through long-radius bends and globe valves. The header extends from a point over the boilers straight into the engine room, where it terminates in a 10-inch steam main, out of the top of which the several branches to the engines are taken, the steam first passing in each case through a separator located directly above the engine throttle. The exhaust lines rise from the engines to a 10-inch exhaust header located overhead and after passing through an oil separator the steam enters an 8-inch heating main which connects with the radiators throughout the building, having a Webster vacuum return. There is also a 12-inch main exhaust line passing to the roof in which is placed an atmospheric relief valve. Tapping the main exhaust directly after leaving the oil separator is an 8-inch connection leading to the feed-water heater. Also at this point there is a 5-inch connection to a heater, through which the building receives its supply of hot water. The piping is all extra heavy with screwed flanges. Both the live- and exhaust-steam lines are well drained at every point where condensation is liable to occur, the several branches leading into a trap, in the case of the high-pressure lines, from which point the condensation is discharged to the drip tank to be used over again. The exhaust lines are also drained by traps discharging to the drip tank or sewer.

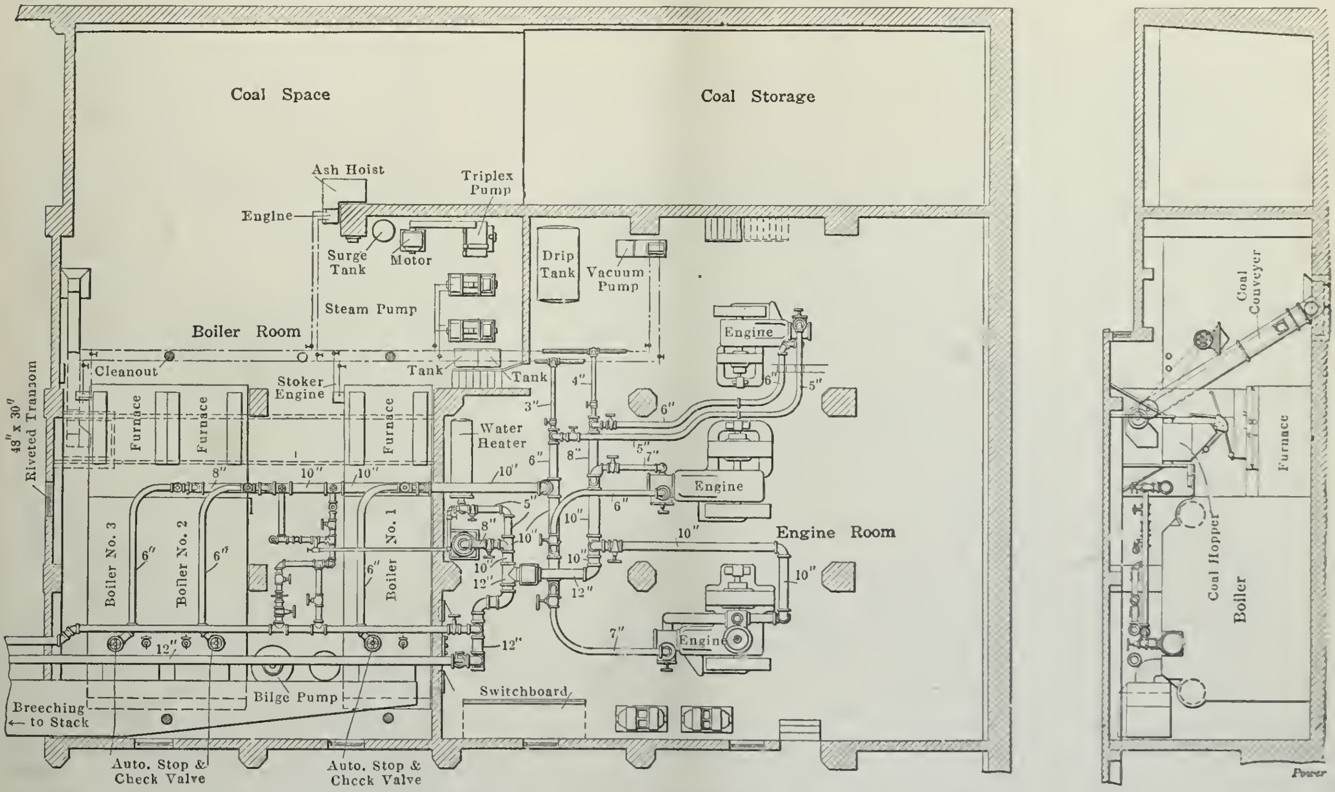


Fig. 5. General Plan and Elevation through Boiler Room

ENGINE ROOM

Three American-Ball generating units are installed having rated capacities of 200, 120 and 75 kilowatts. The engine of the largest unit, shown in Fig. 4, is of the angle-compound type, 17 and 28 by 14 inches and runs at 260 revolutions per minute. The engines driving the other two generators, see Fig. 3, are of the simple horizontal type, one 16x16 inches and the other 14x12 inches.

Direct current is delivered on the two-wire system at 230 volts, and duplicate compensating sets, each of 15 kilowatts capacity, are used to supply lighting current at 115 volts. These compensating sets are arranged to be run in parallel when necessary.

About one hundred and sixty motors are now in use, ranging from ⅛ to 12 horsepower. Adjacent to the switchboard there is a gage board for the three boilers, enabling the operator to have command of the entire plant from this point.

The lighting and power circuits throughout the entire building are in exposed conduit. This is unusual in a concrete building, where the ordinary method is to conceal the conduits in the concrete as the construction progresses. This system makes it possible to add to or subtract from the circuits without destroying the general plan.

The entire plant was designed and its construction supervised by Charles G. Atkins, consulting engineer.

The Steam Turbine in Germany

By F. E. Junge and E. Heinrich

A comparison of the Rateau and Curtis principles of turbine construction. The authors also present a simple method of estimating steam velocities in the blades, force exerted in the direction of blade travel, and "indicated" efficiency of energy conversion, by means of triangles.

STEAM VELOCITIES AND CRITICAL PRESSURE RATIO

If the ratio of pressure before the stage to pressure behind the stage exceeds 1.83 with superheated steam or 1.73 with saturated steam, then conically diverging nozzles must be employed. This obtains as a rule for Curtis turbines working with steam velocities between 2000 and 3000 feet per second. If the pressure ratio remains below the critical value stated, as is the case with the Rateau-Zoelly type, then orifices with parallel walls are used for the guiding apparatus. In contradistinction to Curtis, Zoelly and Rateau employ steam velocities of from 1000 to 1600 feet per second.

THE CURTIS AND RATEAU PRINCIPLES

The main arguments for and against the two types, as generally propounded by the representatives of the respective systems, are as follows: In favor of the Curtis principle is argued extensive expansion of the steam in the first stage, therefore low pressure and low temperature in the casing, whereby the turbine becomes less sensitive to varying temperatures. Further, comparatively little pressure upon the high-pressure stuffing box, hence small leakage losses. Moreover, reduction of axial length of turbine and therefore low cost per horsepower. Finally, the possibility of power regulation by increasing or decreasing the number of nozzles which are active in the first stage.

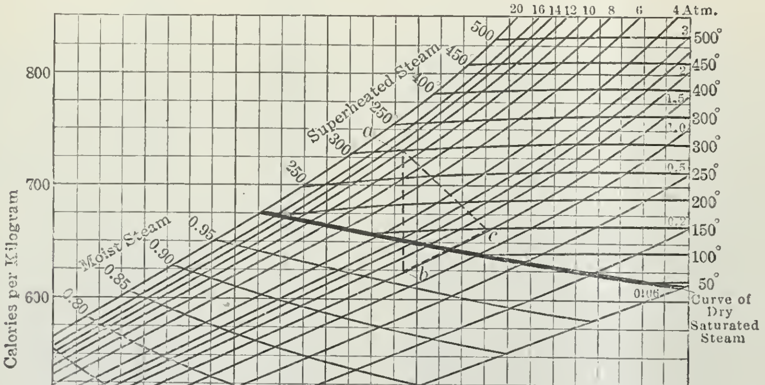


FIG. 6. MOLLIER DIAGRAM; ENERGY THEORETICALLY AVAILABLE IN STEAM

To the high-pressure part built according to the Rateau principle the objection is offered that the full initial temperature of the admission steam enters the casing, that the stuffing box has to stand a comparatively high pressure and that therefore the leakage losses through the high-pressure stuffing box are accordingly high. Further that the losses due to leakage between the various stages, especially in the high-pressure part, are considerable and have an unfavorable effect on steam consumption.

On the other hand, the Rateau advocates offer the objection to the Curtis principle, notwithstanding affirmations to the contrary, that the high steam velocity with which the wheel has to work has in the course of time a destructive effect upon the blades and therefore also an unfavorable effect on the steam consumption. This is said to be especially the case when the turbine works with low superheat and if the Curtis wheel is employed also in the low-pressure part of the turbine; for it is known that the destructive effect of the steam jet increases with the moisture content of the steam. It is further stated, on the basis of results obtained in practical operation, that in impulse turbines the higher temperature of the steam entering into the casing can cause no appreciable difficulties if suitable construction be employed. Also, that reduction of axial length has been attained by recent improvements of design, providing a smaller number of stages; that the high-pressure stuffing box can be kept perfectly tight by employing a modern arrangement of carbon-packing rings, consisting each of two or three segments, and, finally, that regulation by nozzles, or devices for varying the number of active nozzles in the first stage, can be employed in the Zoelly turbine.

THE MOLLIER DIAGRAM

Before entering into the special discussion it is advisable to get clear about the fundamental ideas and methods of investigation. Without taking recourse to theoretical explications we will set forth the practical application. Fig. 6 is the Mollier diagram for determining the available energy in steam under different conditions. The present form of the diagram covers the pressure, temperature, heat content and specific weight of the steam. In newer forms of the Mollier diagram the curves of specific volume are added to the above, but they are omitted here in order not to complicate the presentation. If only two values, for example the temperature and pressure of the steam, are given, then all the others can be read from the diagram. All values are, of course, based on unit weight of steam. The ordinates of the diagram represent the total heat content of a kilogram of steam and the abscissas the entropy of the steam, which, of course, remains constant with adiabatic expansion. As the entropy values are represented by the abscissas an adiabatic change of condition is represented by a vertical line in the diagram. Having to utilize in the turbine the drop of heat energy between a certain initial and a certain final pressure, in order to obtain the energy theoretically available in an ideal engine, it is only necessary to trace along a vertical line from the point of initial pressure to the point where the line intersects the curve of the final pressure, and then read the difference between the initial and final heat content from the diagram.

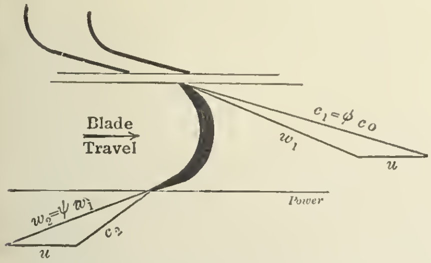


FIG. 7. VELOCITY TRIANGLES

For instance, if steam of 12 kilograms per square centimeter (171 pounds per square inch) at 300 degrees Centigrade is to expand down to 1.5 kilograms per square centimeter (21.4 pounds per square inch) absolute pressure, what is the theoretically available energy per kilogram of steam? The initial point *a* of the process is located by the intersection of the 12-kilogram curve and the 300-degree curve. The vertical line drawn from the point *a* meets the pressure curve 1.5 at the point *b*. Then, from the left-hand scale, the heat content per kilogram at the point *a* is 730 calories and at the point *b*, 625 calories. Hence, the available energy is 105 calories per kilogram, or 189 B.t.u. per pound of steam.

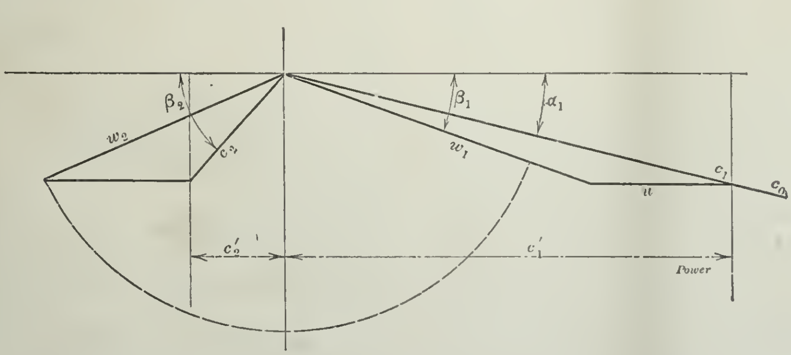


FIG. 8. DIAGRAM FOR IMPULSE WHEEL WITH ONE VELOCITY STAGE

LOSSES

In actual steam-turbine practice losses occur by friction, shock and eddy currents of the steam, which are transposed into heat. This heat cannot disappear and so is added to the original heat in the steam. Therefore during the whole course of the process heat is added to the working fluid. Hence, the expansion will not end at the point *b* but at a higher point, as at *c*, of the pressure curve, the vertical distance between the two points representing 37 calories equals 66.6 B.t.u. per pound, which represents the total amount of these losses. The heat balance comes out as follows: available energy, 189 B.t.u.; losses, 66.6 B.t.u.; utilized energy,

$$189 - 66.6 = 122.4 \text{ B.t.u.}$$

Hence the efficiency of the process is

$$\frac{122.4}{189} = 0.647$$

or practically 65 per cent.

As this figure covers only those losses which are imparted to the steam in form of heat and does not include the exterior heat and mechanical losses, it is usually called the "internal efficiency." But it affords a fair measure of comparison between two turbines.

THEORETICAL HEAT DROP AND THEORETICAL VELOCITY OF ISSUE

To the theoretically available energy, which may be called h_0 per pound, corresponds the theoretical velocity of issue, c_0. In order to establish a relation between the two the energy values must be expressed in the same units. As the mechanical equivalent of a B.t.u. is 778 foot-pounds, h_0 B.t.u. represent 778 *h* foot-pounds. To the velocity c_0 corresponds an energy value of $\frac{c_0^2}{2g}$ foot-pounds per pound of steam. In this expression $g = 32.16$ feet per second, the acceleration by gravity. Hence we have the equations of energy

$$\frac{c_0^2}{2g} = 778\, h_0$$

and

$$c_0 = 223.7 \sqrt{h_0}$$

VELOCITY TRIANGLES

Owing to the losses occurring in the guiding apparatus the steam issuing from that apparatus has not the ideal velocity c_0, but a lower velocity, $c_1 = \phi c_0$, ϕ being the coefficient of loss in the guiding apparatus. Fig. 7 shows the velocity triangles for the entrance into and exit from the moving blade, by the steam. Now, in determining what is the velocity of flow of the steam in the blade channels, one must bear in mind that the blades are not at rest, but travel with the velocity *u* per second, which is the circumferential velocity of the wheel. But since, on the drawing board, we must represent the blades as stationary, it is necessary in order to get at the entrance velocity into the blade—the so-called relative entrance velocity—to go back in the opposite direction from the end of the velocity line c_1 by the amount *u*. This locates the end of the line w_1, which represents the velocity of the steam relative to the blade at the entrance. When the steam passes through the blades, losses occur through friction, shock and eddy currents, so that the relative exit velocity $w_2 = \psi w_1$, wherein ψ is the coefficient of loss in the rotating blades.

In order to get from the relative outlet velocity w_2, the absolute outlet velocity c_2, it is necessary again to consider the movements of the blades and this time to advance from the end of the vector w_2 in the direction of the blade travel by the amount *u*. This locates the end of the line representing the absolute velocity of issue c_2. In Fig. 8 we have combined the two velocity triangles as it is generally done in practice.

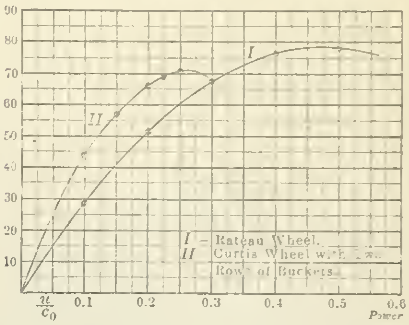


FIG. 9. CURVES OF INDICATED EFFICIENCY

DETERMINATION OF INDICATED EFFICIENCY

Indicated efficiency ηi is the ratio of the work transmitted to the blades to the theoretical work. The losses which are accounted for in this expression are only those in nozzles and blades. By changing the direction of flow and the magnitude of absolute velocity when traveling along the curve of the blade, the steam exerts a force which acts in the direction of the movement of the blades. Now the question presents itself, how large is the impelling force of the deviated steam? Before answering this question we must refer to the elementary law of mechanics which establishes the relation between force, mass and change of velocity within the unit of time (acceleration and retardation). According to this law, force = mass × acceleration. If the acceleration is positive, that is to say, if the velocity increases, then the force is imparted to the mass. If the acceleration is negative, the movement of the body being retarded, as is the case with the steam in our example, then the mass supplies the force.

In order to determine the change of velocity per second, the components of the absolute entrance velocity vector c_1 and the absolute exit velocity vector c_2 in the direction of blade movement are plotted, as shown in Fig. 8. The sum of the two projections, $c' + c_2'$, obviously represents the change of velocity per second. The mass of one pound of steam is $\frac{1}{g}$. Hence one pound of steam has imparted to the blades the force

$$P = \frac{1}{g}(c_1' + c_2')$$

and, since u = travel per second, the work done by unit weight of steam is

$$w = P\,u = \frac{u}{g}(c_1' + c_2')$$

As was mentioned above, the available energy contained in one pound of steam working in an ideal engine is $\frac{c_0^2}{2g}$. Therefore, the indicated efficiency is

$$\eta_i = \frac{\frac{u}{g}(c_1' + c_2')}{\frac{c_0^2}{2g}} = \frac{2u(c_1' + c_2')}{c_0^2} \qquad (1)$$

If h_0 represents the available energy per unit of weight in heat units, $\eta_i h_0$ is the energy transmitted to the blades, and

$$R = (1 - \eta_i)h_0 \qquad (2)$$

is the loss in nozzles and blades. If η_i is to be determined for various velocity conditions, the velocity diagrams are drawn on the board, c_1' and c_2' are obtained from the diagram and η_i is calculated from formula 1. When doing so, it is advisable to vary the ratio $\frac{u}{c_0}$ (blade velocity to theoretical velocity of issue), which determines the value η_i. If c_0 is taken equal to the unit of length, then the denominator of the fraction in equation (1) becomes 1 and the efficiency appears simply as twice the product of the two factors u and $(c_1' + c_2')$. In Fig. 9 we have drawn the η_i curve for various values of $\frac{u}{c_0}$. It is evident that the efficiency η_i increases with the ratio $\frac{u}{c_0}$ up to a certain maximum value, which for Rateau wheels lies in the neighborhood of 0.5.

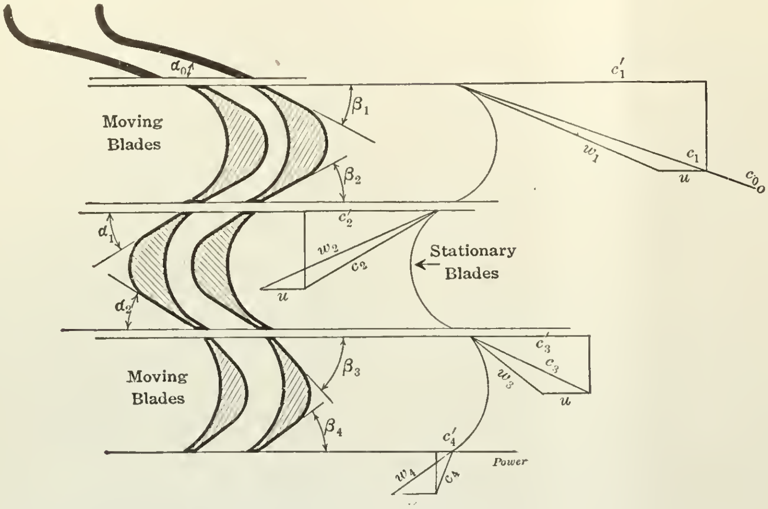


FIG. 10. VELOCITY DIAGRAM FOR CURTIS BLADING

APPLICATION TO TWO ROWS OF BLADES

If the velocity c_2 at which the steam leaves the first row of moving blades is still high enough, it is utilized in a second row of blades, which are fastened on the same wheel disk. For this purpose the direction of the flow of the steam must be so controlled by an intermediary row of stationary blades that it impinges upon the blades of the second row in a manner yielding the highest efficiency. Fig. 10 shows such a combination of blades. The indicated efficiency is determined in the manner just described for one row of blades. We must therefore add in the numerator of the fraction in equation (1) to the indicated work transmitted to the first row of moving blades the work transmitted to the second row. Thus, if c_3 and c_4 are the entrance velocities into the second row of moving blades, and c_3' and c_4 the components of these velocities in the direction of blade movement, the following simple expression is obtained for the Curtis wheel:

$$\eta_i = \frac{2u(c_1' + c_2' + c_3' + c_4')}{c_0^2} \qquad (3)$$

wherein c_0 is again the ideal velocity of issue corresponding to the entire drop of energy h_0. Fig. 9 contains, besides the efficiency curve for one velocity stage, the curve for two velocity stages, for special conditions.

SPECIAL FEATURES OF THE CURTIS PRINCIPLE

In accordance with what was said of the Curtis principle in our first article (POWER for November 8) we repeat that the efficiency of the Curtis blading may be increased by not expanding down to back pressure in the nozzle but converting a certain fraction of the energy drop in the first row of rotating blades. With this practice it is, however, not possible to avoid a certain leakage loss, because within the first blade channel the pressure is higher than without. Assuming that h_2 heat units are transferred into velocity energy in the first row of moving blades and that the losses occurring therein are ξ per cent. of the energy $\frac{w_2^2}{2g}$, which is represented by the relative velocity of issue, this energy must be equal to the energy which corresponds to the relative entrance velocity plus the energy 778 h_2 converted in the blade channel, minus the losses in the blade channel $\frac{\xi}{100}\frac{w_2^2}{2g}$. Therefore,

$$\frac{w_2^2}{2g} = \frac{w_1^2}{2g} + 778\,h_2 - \frac{\xi}{100}\frac{w_2^2}{2g}$$

and

$$w_2 = \sqrt{\frac{w_1^2 + 2\,g\,778\,h_2}{1 + \frac{\xi}{100}}} \qquad (4)$$

The total number of newspapers published in the world at present is estimated at about 60,000, distributed as follows: United States and Canada, 23,461; Germany, 8049; Great Britain, 9500; France, 6681; Japan, 1000; Italy, 2757; Austria-Hungary, 2958; Asia, exclusive of Japan, 1000; Spain, 1000; Russia, 1000; Australia, 1000; Greece, 130; Switzerland, 1005; Holland, 980; Belgium, 956; all others, 1000. Of these more than half are printed in the English language.

The White Star liner "Olympic" will be fitted with two sets of reciprocating engines for driving the outer screws, while a third screw, placed centrally, will be actuated by a low-pressure turbine, served by the exhaust steam from the rotary sets. She and her sister vessel, the "Titanic," are considerably larger than the "Lusitania" and the "Mauretania," but not so speedy. Their length overall is something like 100 feet greater than the Cunarders, and their horsepower 46,000, as compared with the others' 70,000.

Test of a 10,000-Kilowatt Turbine[*]

By Samuel S. Naphtaly

> *This unit, consisting of a double-flow, combined impulse and reaction turbine connected to a three-phase, sixty-cycle generator, installed in the plant of the City Electric Company of San Francisco, showed a water rate of approximately 13.5 pounds per kilowatt-hour at full load.*

The object of the test was to determine the steam consumption of the turbine under normal operating conditions and further to determine whether the builder's guarantees of economy, generator temperature, etc., had been complied with. The specifications called for a turbine of 10,000 kilowatts normal capacity with an additional overload capacity of 50 per cent., operating at 175 pounds gage pressure, and 100 degrees superheat with a 28-inch vacuum, measured in the exhaust. Under these conditions, when running at 1800 revolutions per minute, the guaranteed performance was as follows:

Full load, 10,000 kilowatts, 14.3 pounds of steam per kilowatt-hour.

One-half load, 5000 kilowatts, 16.2 pounds of steam per kilowatt-hour.

Fifty per cent. overload, 15,000 kilowatts, 15.3 pounds of steam per kilowatt-hour.

The turbine was of the double-flow type, containing both impulse and reaction elements, a section of which is shown in Fig. 1. The governing apparatus presented no particularly novel features, there being two governor-controlled inlet valves, each admitting steam to a group of nozzles, with full load carried on the primary valve alone. Overloads were maintained by a group of nozzles operated by the secondary valve. These valves were hydraulically operated, oil under pressure being supplied by the oil pump driven by the turbine. The operation of the hydraulic gear was accomplished by the familiar relay and floating lever. Hitherto, it has been customary to construct the relay for a mechanism of this character without lap; or if it does have lap, either negative or positive, the amount of this lap must be traveled before the mechanism can respond to the governor. A departure was made in this machine from previous practice by providing the relay plunger with both inside and outside laps, the relay being kept oscillating a small amount by means of a cam connected to the governor linkage, sufficiently to uncover slightly the ports at each oscillation, and causing a correspondingly slight motion of the main operating piston. The effect of the governor itself on this relay, is, of course, to change its plane of motion, insuring a response to the least change in position of the governor weights.

The generator was a three-phase, sixty-cycle machine having a normal rating of 525 amperes at 11,000 volts. The field was of nickel steel and made up of two halves joined end to end by shrink links. Each half was integral with its shaft, rendering any hole for the shaft unnecessary and consequently considerably increasing the strength of the revolving field.

All important readings were taken at five-minute intervals except in cases of energy output and water measurement, for which fifteen-minute intervals proved adequate. The electrical output was obtained with a polyphase wattmeter, and the steam-consumption measurements were made by actually weighing the water discharged from the surface condenser.

Tests were made for leakage of the cooling water (salt water) into the condenser by adding silver nitrate to samples of the condensed steam but no indications of leakage were found. The shaft glands were sealed by water taken from the condenser discharge before weighing and the leakage from the outside of the glands was caught and measured. In this way no special allowance was necessary for gland-sealing water. The stuffing boxes of the centrifugal hotwell pumps were sealed with water from the discharge. A small leakage occurred here which was caught and added to the leakage from the main unit gland. All pipes connected with the condenser or piping system, by which water might escape from the discharge or leak into the discharge water, were disconnected, blanked or arranged with two valves with an opening between for detecting leakage. Observations were made of the pressure and temperature of the steam at the throttle, pressures within the turbine at the inlet, intermediate and low-pressure points. The vacuum was indicated by a mercury column and the atmospheric pressure by a mercury barometer.

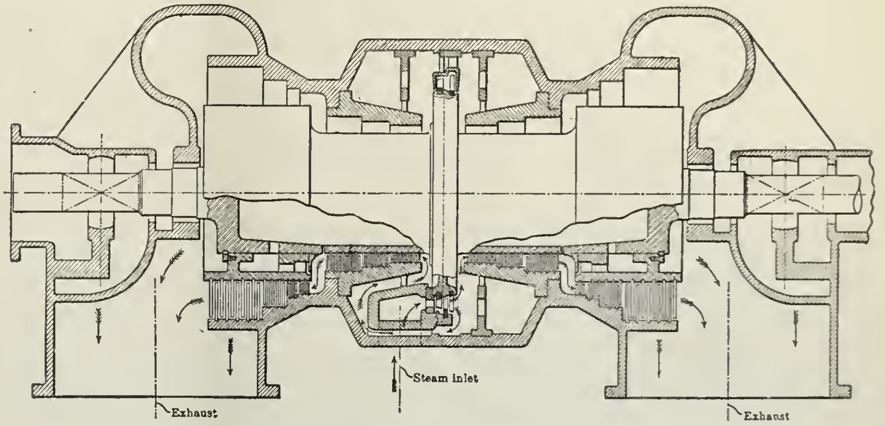


Fig. 1. Sectional View of Turbine

SUMMARY OF TEST.

Run.	Duration, Hours.	Actual Load, Kilowatts.	Power Factor, per Cent.	Steam Pressure, Pounds Gage.	Superheat, Degrees Fahrenheit.	Vacuum, 30 Inches.	Water per Hour, Total, Pounds.	Water per Kilowatt-hour as Tested.	Rankine cycle Efficiency as Tested.	Thermal Efficiency, per Cent.	Pound per Kilowatt-hour, Corrected 175 Pounds, 100 Degrees Superheat, 28 Inches Vacuum.
A	1.5	7,972	90	171	58	28.28	116,238	14.581	66.9	22.20	14.11
B	2.0	8,563	88	168	59	28.18	123,542	14.427	68.0	22.65	13.88
C	2.0	8,198	91	169	60	28.10	119,655	14.596	67.6	22.10	14.04
D	2.0	9,173	91	167	59	27.90	133,672	14.572	69.0	22.23	12.88
E	3.0	5,333	88	173	54	28.34	83,478	15.655	61.8	20.70	15.21
F	4.0	8,148	88	167	60	26.16	129,181	15.855	69.5	20.35	14.07
G	3.0	5,401	88	174	56	26.16	95,120	17.611	62.2	18.35	15.24

A summary of the test is given in Table 1. During the high-load tests, this was the only unit operating in the station, so that observations had to be made under the load variations incidental to station operation. In all, seven runs were made, two of which, *E* and *G*, were at low vacuums for the purpose of establishing the effect of variations of vacuum on this machine. Both the thermal efficiency and that of the Rankine cycle were computed. The thermal efficiency considers only the heat in the steam from the boiler, and is of use only for comparing results with other forms of prime movers, such as internal-combustion engines. The Rankine-cycle efficiency, or efficiency ratio as it is sometimes called, is the more equitable measure for comparison of steam engines, as it deals with the work done between the limits of the steam leaving the boiler and the exhaust entering the condenser.

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[*] Abstract of paper read before the American Society of Mechanical Engineers, December 6-9.

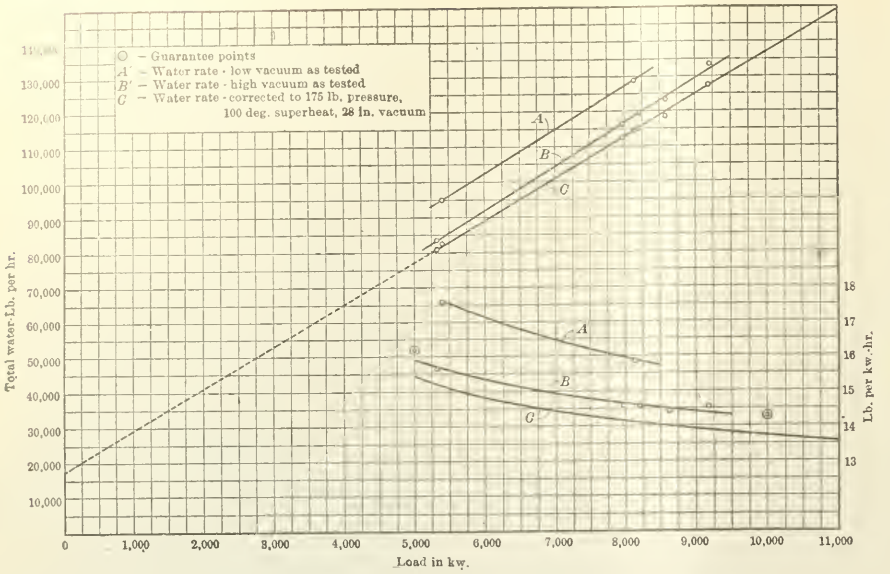


Fig. 2. Actual Water-rate Curves Obtained from Tests

The water rates obtained during the test are plotted in Fig. 2. From these it is reasonable to expect that if a greater load had been available, the steam consumption would have further improved until the secondary valve opened. Allowing for a 10-pound drop through the inlet valves the load would have been approximately 11,000 kilowatts when a steam consumption of 13.7 pounds would have been obtained with an efficiency of 7^1 per cent.

Corrections

As the operating conditions specified by the contract, namely, 175 pounds gage pressure, 100 degrees superheat and 28 inches vacuum, could not be maintained throughout the test, correction factors had to be determined upon, and an agreement was reached previous to the tests as follows:

Superheat—One per cent. change in steam consumption for ten degrees in superheat, which is the correction factor commonly employed for turbines.

Steam Pressure—One-half per cent. for each 10-pound change in pressure. However, as there was no reason for the full pressure not being maintained these were insignificant.

Vacuum—As the vacuum correction is affected by the design of the turbine, the amount of the correction could not be determined in advance of the tests; therefore runs *E* and *G* were made. From these and from the other tests it was found that 1 inch of vacuum affected the steam consumption 3 per cent. and 6 per cent. at full load and half load respectively.

Boiler Tube Soot Blowers

By Warren O. Rogers

Blowers fixed permanently in the boiler setting do the work quickly, and as effectually as by hand. They are always in place and ready for use. Several types of this class of tube cleaner, including those used for both return-tubular and water-tube boilers, are described.

Among the latest idea in soot blowers is that of arranging the device permanently in the brick setting in such a manner that the blower head is protected from the heat of the furnace when the tubes are not being blown. Quite a number of these devices have been designed.

The "Diamond" steam-flue blower belongs to the type that is permanently installed in the rear wall of the boiler setting, as shown in Fig. 1. It blows the soot in the natural direction of the draft to the front of the boiler and up the chimney. When not in use the nozzle is withdrawn into a sheath by the coiled spring in the handle to prevent injury from the heat. By rotating the handle the numerous jets of steam cover a circle 4½ feet in diameter. The work of cleaning can be carried on while the boiler is in operation. This device is made by the Diamond Power Specialty Company.

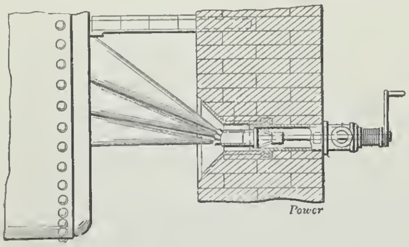


Fig. 1. "Diamond"

What is known as the "Henry" steam-flue cleaner is shown in Fig. 2. It is installed at the rear of the boiler and cleans with the draft, the soot being blown forward into the smoke-box extension and up the stack. The nozzle pipe, which covers one row of tubes at a time, is raised or lowered by means of the crank shown at the top. When not in use the head is drawn up close under the brickwork out of the direct path of the hot gases. Close contact is produced with each tube, and the tubes are cleaned in a few minutes. It is made by W. M. Sage.

Fig. 3 illustrates the "American" flue blower, made by Eugene J. Feiner. It is placed in the rear wall of the boiler setting, and may be operated from the rear or side walls. It blows with the natural draft of the boiler, and is arranged to blow a solid sheet of steam across the entire width of the boiler and directly into the tubes at the least possible angle, driving the soot forward to the smoke-stack.

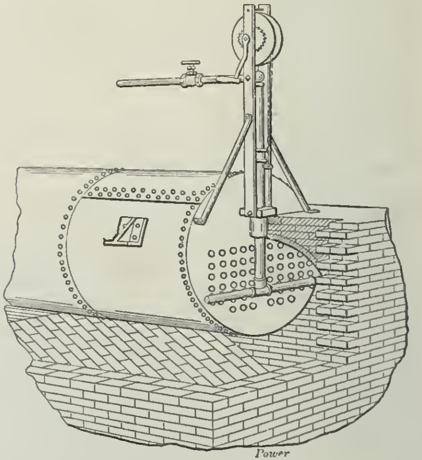


Fig. 2. "Henry"

The "U. S." tube blower, made by the U. S. Specialty Manufacturing Company, is shown in Fig. 4. It is a device for blowing soot out of boiler tubes

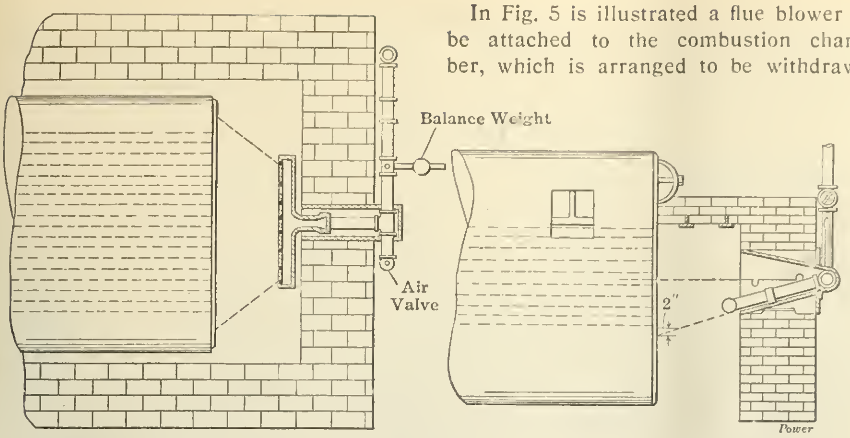


Fig. 3. "American"

with the draft. A casing passes through the rear wall of the boiler setting, and a T-shaped cast chamber outside the wall receives a 1½-inch supply pipe. Directly under this pipe in the chamber is a drain cock to dispose of the condensation, and on the end of the chamber is a stuffing box through which the handle rod passes to an inner tube which just fills the casing.

On the end of this tube is a hollow cast-iron arm, one-half the diameter of the boiler in length, having a 1/16-inch slot its entire length on the side toward the boiler. By pushing the rod endwise this arm is brought close to the rear end of the tubes into which the steam is delivered in a thin sheet. By means of a handle secured to a rod the arm is revolved so that the steam jets reach all of the tubes.

The blower can be used while the boiler is in operation, and when not in use the arm is pulled back and placed in a horizontal position on a shelf where it is not exposed to the heat.

In Fig. 5 is illustrated a flue blower to be attached to the combustion chamber, which is arranged to be withdrawn from the influence of the hot gases when not in use. The device is made by the Zenith Manufacturing Company.

It consists of a hollow casting, with a packing nut at each end, through which moves a hollow rod having a handle attached. The inner member of the rod, which has a length equal to the radius of the tube sheet, is perforated with holes corresponding to the tubes of the boiler, and swings on a steam-tight joint. The steam enters the ports in the hollow rod and, by turning the handle, all the tubes are cleaned by a jet of steam, and the soot is blown out in the same direction as the draft of the boiler. When not in use all parts of the apparatus are removed from the influence of the heat and gases, and a disk on the end of the blower closes the aperture through which the swinging joint is inserted.

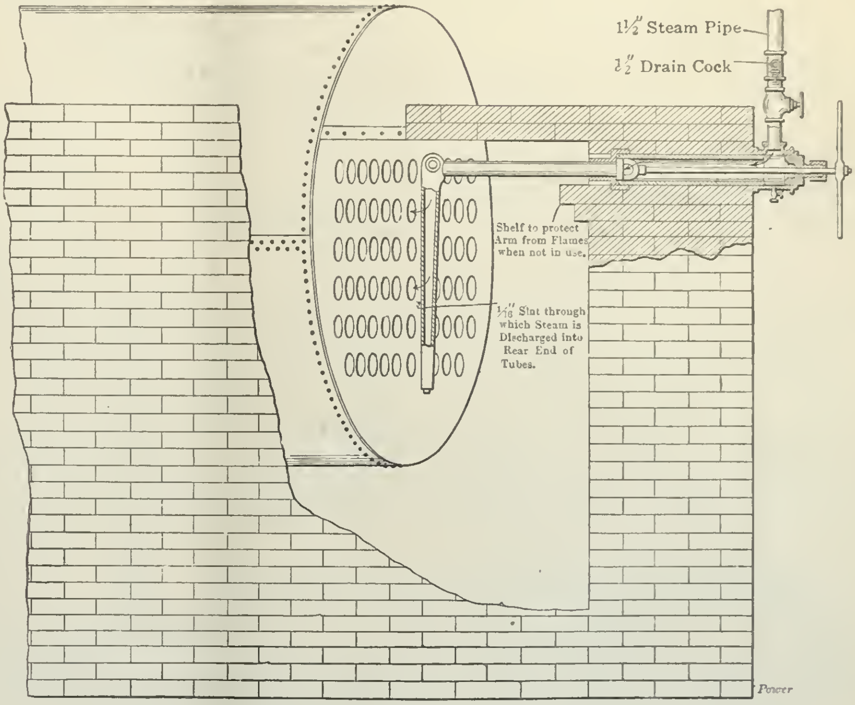

1½" Steam Pipe
½" Drain Cock
Shelf to protect Arm from Flames when not in use.
1/16" Slot through which Steam is Discharged into Rear End of Tubes.
Power

Fig. 4. "U. S." Tube Blower

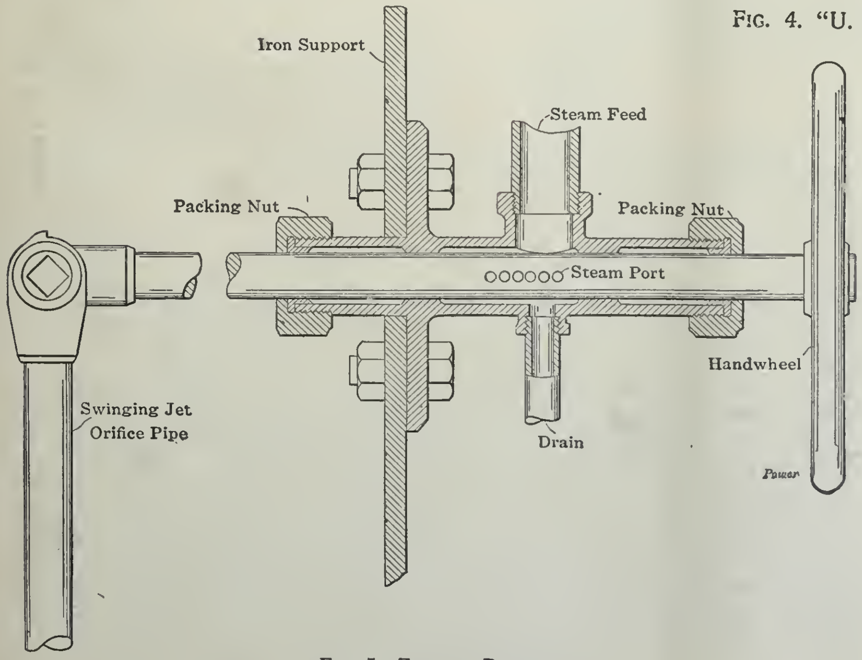


Fig. 5. Zenith Blower

The "Marion" flue blower, made by the Marion Machine, Foundry and Supply Company, is shown in Fig. 6. The blower is a permanent fixture in the rear wall of the boiler setting and, as usual, blows the soot in the direction of the draft out through the chimney. The feature of the device is the rotating nozzle which has three, sometimes four, openings, according to the size of the boiler, these openings all pointing to a different section of the tube sheet. On the base of the nozzle casting is a flat valve seat on which a disk with one opening is held by the steam pressure. The disk may be rotated by a valve stem attached to the indicator on the handwheel, and thus each nozzle opening may be blown in turn. As the nozzle is rotated while each opening is being blown, all of the boiler tubes are cleaned.

Fig. 7 shows the heavy cast-iron cap which protects the nozzle from the fire. The blower is located opposite the center of the tube space, but not in the center of the boiler, and each one is constructed especially to fit the existing conditions of combustion space, firewall, etc., in different boiler settings, a nozzle being furnished which will insure the steam reaching all the tubes.

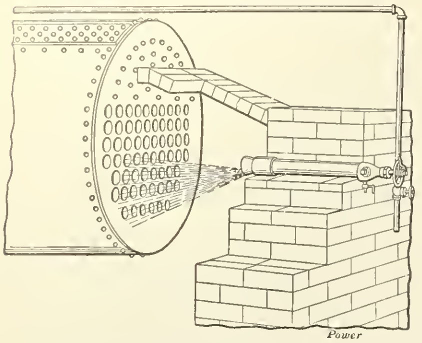


FIG. 6. "MARION"

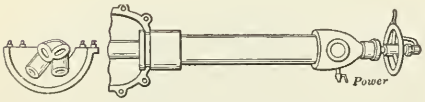


FIG. 7. MARION NOZZLE

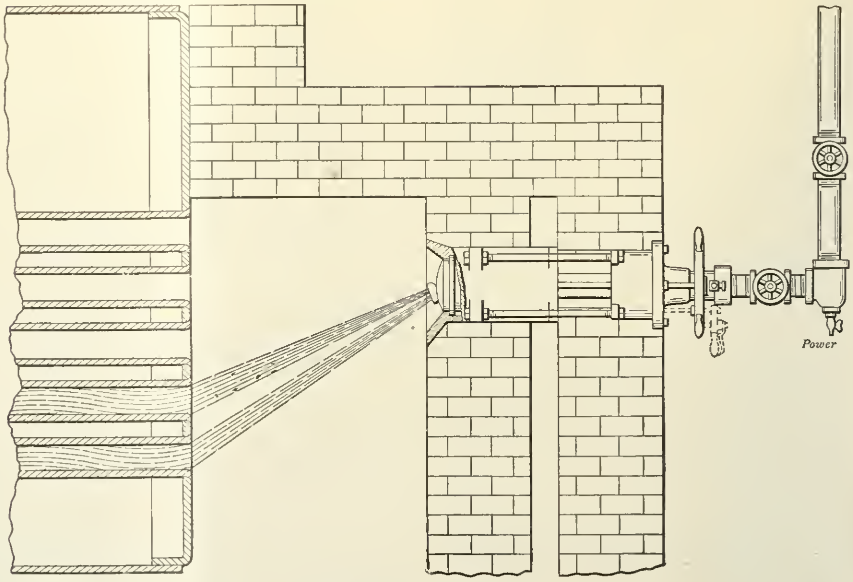


FIG. 8. "MONARCH"

The "Monarch" boiler-tube blower is manufactured by the Monarch Steam Blower Company. It is shown in side elevation in the rear wall of a boiler setting in Fig. 8. The apparatus consists of an inlet pipe, a rear plug and a sleeve connecting with the plug and provided with a slot at its inner end and a valve seat. It also has a hollow valve stem with a cupped rear end forming a steam-tight connection with the valve seat in the sleeve when in operation; at its inner end the stem is provided with outlet apertures arranged at an angle to its axis.

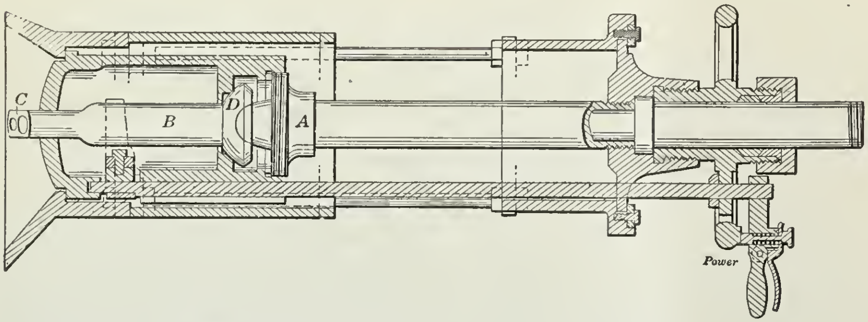


FIG. 9. SECTION THROUGH MONARCH BLOWER

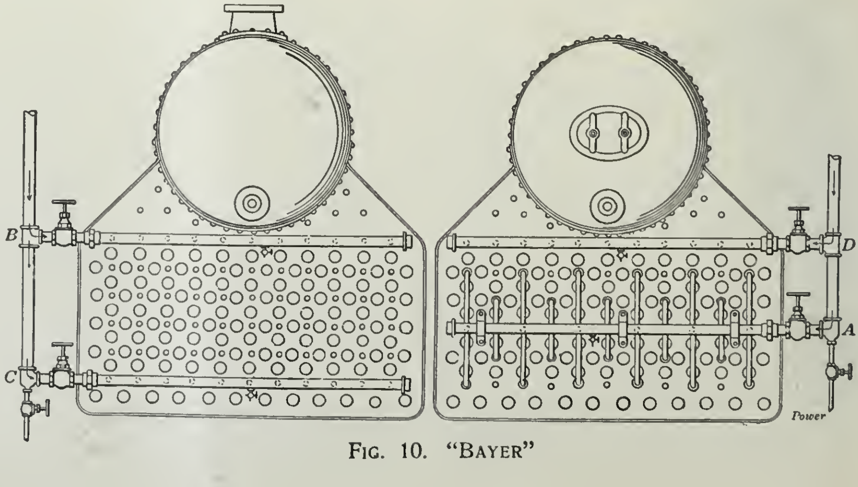


FIG. 10. "BAYER"

Means are adopted to swing the inner end of the valve stem radially in the slotted end of the sleeve, also for rotating the valve stem. The shifting device consists of a fork and block operated by a shaft turned by a handle which carries a spring pin engaging at set points with counterborings on the rim of the handwheel, which indicate the particular row of tubes being cleaned.

In operation steam is admitted through the globe valve in the steam pipe, passes on to the rear plug *A*, Fig. 9, directly into the valve stem *B* and out of the aperture *C* into the boiler tubes, as shown in Fig. 8. The impact of the steam when admitted forces the seat *D* of the valve stem against the seat of the cylindrical sleeve, making a steam-tight connection and concentrating the entire boiler pressure through the aperture *C*, and then through the length of the boiler tube. After the operator has set the controller in one of the counterbore holes, the blower is rotated by means of a handwheel, after which the controller is shifted to the other counterbored holes in their order. Thus the blower revolves at each shift and blows steam through the entire number of boiler tubes.

The "Vulcan" soot cleaner, made by the Vulcan Soot Cleaner Company, is illustrated in Fig. 11. It is designed for cleaning the accumulation of soot from the heating surfaces of water-tube boilers. It is made up of an arrangement of stationary piping connected to a steam main and is installed on the side of the boiler setting.

It has twelve or more vertical branches, with an operating valve for each, and perforated pipe extensions reaching between the tubes the entire width of the boiler. These extensions are so located and perforated as to direct steam jets at every part of the heating surface with effective force. Short three-way nozzles embedded in the brickwork complete the system, reaching the points of extreme temperature.

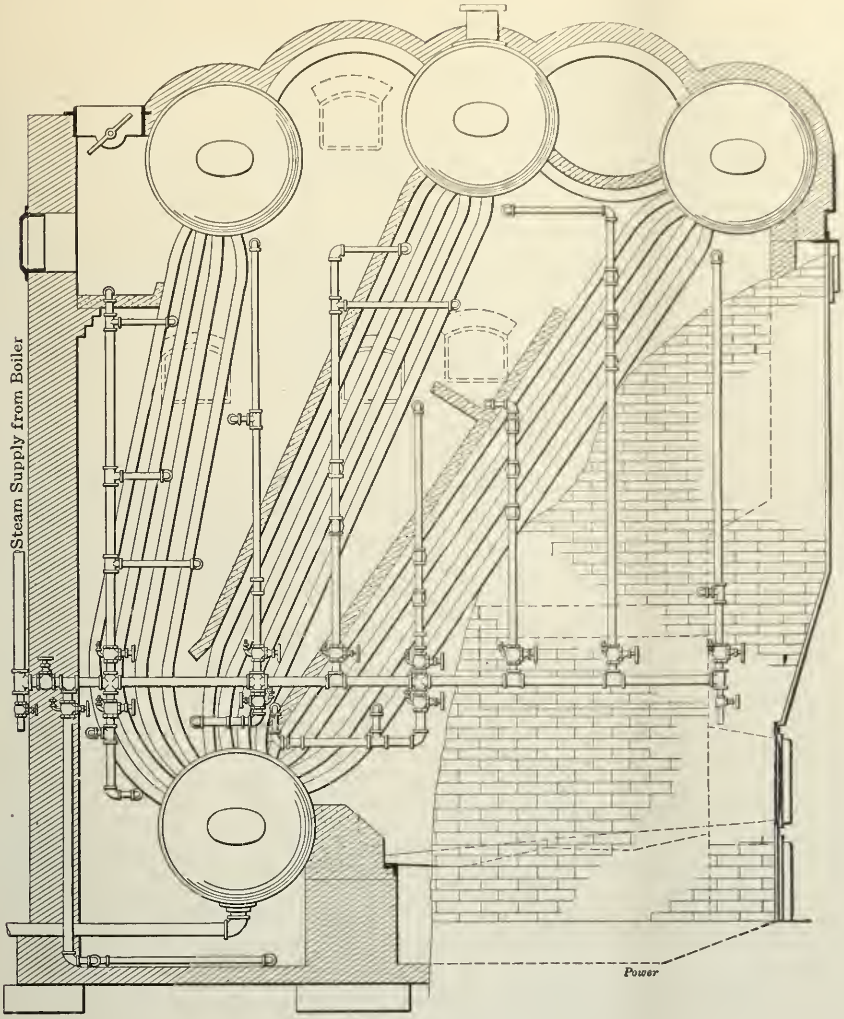


FIG. 11. "VULCAN"

The system is shown as applied to a Stirling boiler. It is operated by opening and closing the valves in succession and blowing with the draft, so that the operation interferes in no way with the regular working of the boiler.

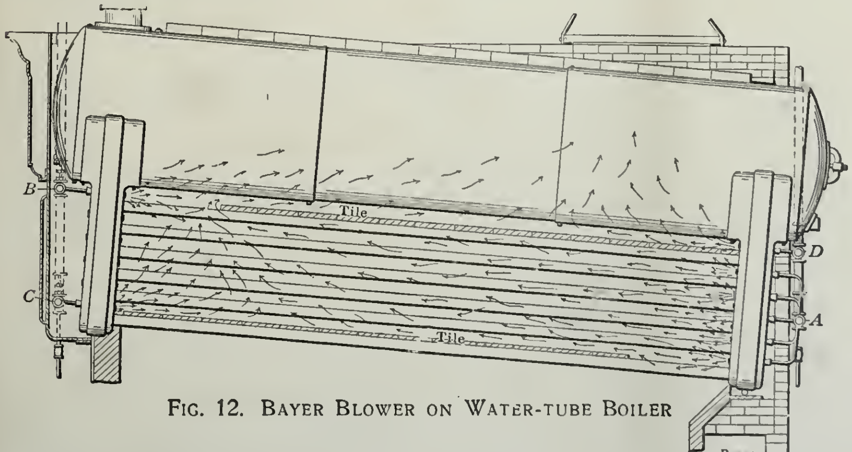


FIG. 12. BAYER BLOWER ON WATER-TUBE BOILER

The "Bayer" soot blower is a device for cleaning the tubes of water-tube boilers. In Fig. 10 is shown the elevation of the rear water leg of a water-tube boiler with the system in place. The lower header carries the nozzle for blowing the tubes between the upper and lower baffles. The upper header is used to agitate the dead spaces known to exist at the point where the jets come through the water legs. Fig. 13 shows a side elevation of the system as applied to a water-tube boiler.

When operating the system, it is necessary to open the controlling valves *A* and *B*, Fig. 12, about three minutes, letting them operate in conjunction. At the same time valves *C* and *D* are opened frequently during the operation in order to stir up the accumulations which have been blown into the space between the headers.

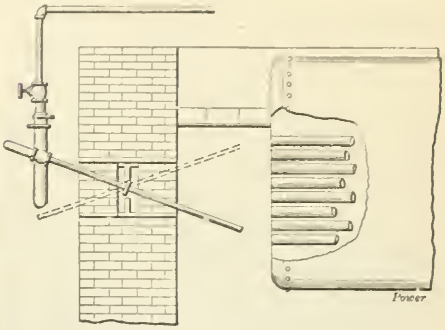


FIG. 13. "ARROW"

The system is easily installed. All that is necessary is to make a steam connection on the header or some other convenient source of steam supply and attach the blower by means of an ordinary union. The jet extends into the hollow stays flush with the inside of the water leg, and is thereby protected from the heat. This system is manufactured by the Bayer Soot Blower System.

The "Arrow" tube blower, shown in Fig. 13, is manufactured by the Arrow Specialty Company. It operates from the rear of the boiler, and the draft through the tubes assists the steam jet. The steam entering the tubes is sent through them at high velocity, much of the soot and fine ashes escaping up the chimney.

The steam pressure is not materially reduced as the blower tends to increase the draft, instead of retard it, so the rate of combustion and steam making is not interfered with.

The device consists of a single jet that can be used at any distance from the ends of the tubes, or it may be inserted into the tubes if desired. The motion of the jet is both horizontal and vertical, and the extreme positions are predetermined so that no steam can be wasted; all the steam issuing from the jet must enter the tubes.

[Since the foregoing article was written, descriptions of the improvements made in the design of both the Bayer and Monarch soot cleaner have been published in the December 6, 1910, number, pages 2169 and 2170.—EDITOR.]

Next fall, Pittsburg is to celebrate the centennial of steam navigation on the Ohio and Mississippi rivers. The "New Orleans" was the first steamboat to make the trip from Pittsburg to New Orleans, and it is claimed that the boilers were built by the great uncle of Col. T. R. Roosevelt.

Uncle Pegleg's Philosophy

The old man explains in a simple manner the operation of pulleys, or blocks as they are most always called, with one to a number of sheaves. He shows how a small pull can lift a big weight and brings out the difference the direction of pull makes.

"How much pull is there on the hook of that top pulley when they are hoisting a bag?"

They were putting a gravel roof on a neighboring mill and had a rig like Fig. 1 to hoist the bags of gravel.

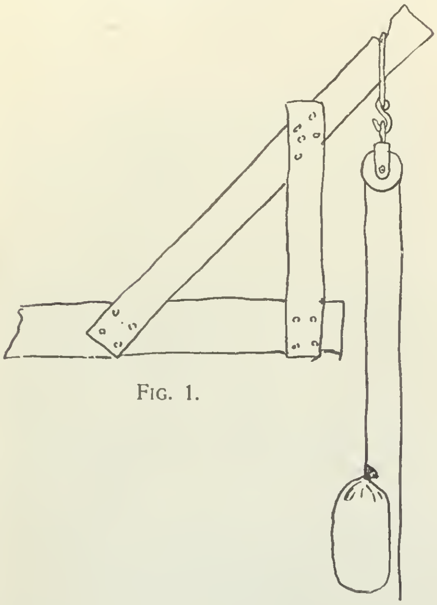


FIG. 1.

"If the bag weighs 100 pounds and it's hanging on the pulley and the pulley is hanging on the hook, I suppose the pull on the hook is the weight of the bag and the pulley and the rope."

The old man simply looked at me in a manner which suggested that I keep on thinking.

"Plus the friction?" I suggested.

"We can forget the friction," he said.

"Plus the extra pull to get it started," I guessed again, "like the big wrench on the elevator?"

"Well, that would come in," he said, "while the speed of the bag was being increased, but imagine it going up steady or standing still. You are thinking too far away and too complicated. Take the simple proportion and pick it apart."

I picked for some minutes but didn't get anywhere. Finally, the old man handed me a sketch like Fig. 2.

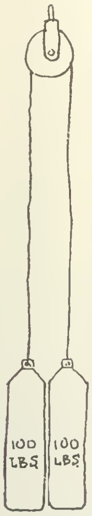


FIG. 2.

"How much pull is there on it now?" he asked.

"Two hundred pounds—for the gravel," I answered.

"Good, never mind the friction and the rest. Due to the gravel there's 200 pounds pull on the hook. Now, if I hold the rope as in Fig. 3 and let you take away the second bag of gravel, how hard have I got to pull to balance the original bag?"

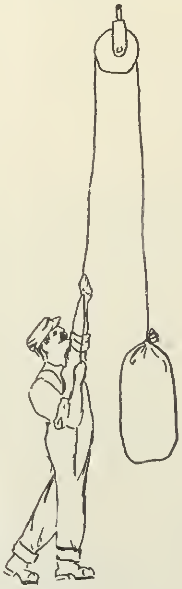


FIG. 3.

"One hundred pounds."

"Well, how does the hook know whether it is the bag or I that's pulling on it? The two ropes are pulling down with a force of 100 pounds each, and it's just the same when the horse does the pulling; so you see a 100-pound bag of gravel can pull 200 pounds."

"No, the horse or the man pulls one hundred and the bag pulls the other hundred," I remonstrated.

He sketched Fig. 4 and said:

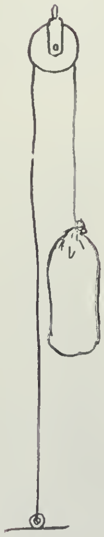


FIG. 4.

"Hang on the bag and you've got a pull of 200 pounds on the hook. Take off the bag and you've got no pull on the hook. How do you get away from it that a 100-pound bag pulls 200 pounds?"

"Ah, ha! I see," said I, "that's the way they gain power with a fall. The more ropes you use the more pull you get."

"No. The more ropes you put on the *less* pull you get, and you don't gain power at all."

"You don't gain power by using a fall?"

"Not a bit. To raise a 100-pound bag of gravel 55 feet in 10 seconds would take a horsepower whether you raised it with a four-rope fall or pulled it up with a dead pull."

"What is a fall good for then?" I asked.

"To let you do with a small force what it would otherwise take a big one to do. But force isn't power."

"What's the difference?"

"You know what force is, don't you?"

"Yes," I replied confidently.

"Well, what is it?"

After thinking it over, I concluded that I could exemplify better than I could define, and said, "When I push with my foot against the bench, that's a force."

"Yes, muscular force," approved the old man.

"And the bag of gravel pulls on the rope with a force?"

"The force of gravity; yes."

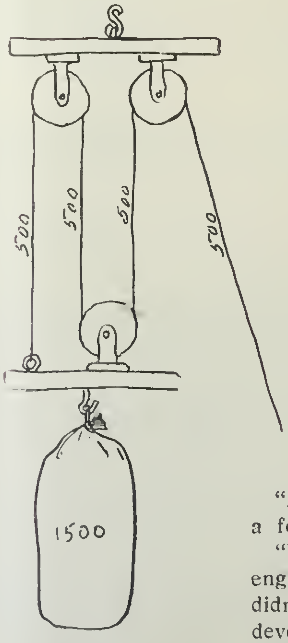


FIG. 5.

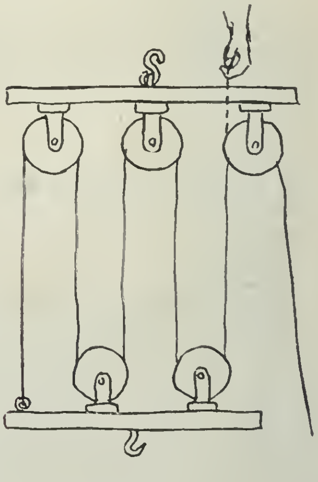


FIG. 6.

"And the steam pushes the piston with a force?"

"Yes," said the old man, "and, if the engine was on the center and the piston didn't move, there would be no energy developed and no power, would there?"

"No."

"You know what energy is?"

"Yes, force times space."

"Good. If the 100-pound bag of gravel is lifted 55 feet, how much energy will it take?"

"It will take

$$100 \times 55 = 5500 \textit{ foot-pounds,"}$$

I said.

"Correct. Now, you see, don't you, that it takes a more powerful machine to lift that weight in a second than it does in a minute? To lift 10.000 pounds of water 100 feet will take 1,000,000 foot-pounds of *energy*, whether you lift it in a second or a year. A mighty small pump will put it up there if you give it time enough, but it takes a powerful pump to put it up quick, and the quicker the more powerful the pump. *Power* is *energy* divided by *time;* foot-pounds per minute or per second. A horsepower is 550 foot-pounds per second; so if the bag was lifted in 10 seconds the power would be

$$\frac{\text{force} \times \text{space}}{\text{time}} = \frac{100 \times 55}{10} = 550 \text{ ft. lbs. per sec.}$$

or one horsepower."

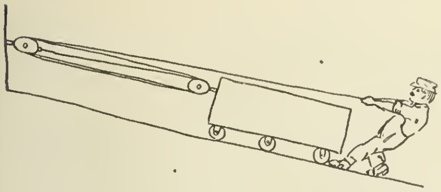


FIG. 7.

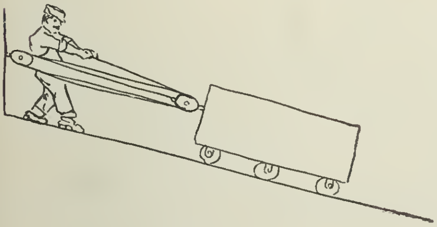


FIG. 8.

"I thought a horsepower was 33,000 foot-pounds," I said.

"So it is,

$$550 \text{ ft. lbs. per second}$$
$$550 \times 60 = 33000 \text{ ft. lbs. per min.}$$
$$33000 \times 60 = 1,980,000 \text{ ft. lbs. per hour}$$

"Now," Uncle Pegleg went on, "suppos'n that bag weighed 1500 pounds instead of 100, and the horse could not lift it. We could hang it like this," and he drew Fig. 5. "How many ropes is the bag hanging on?"

"Three," I answered.

"Then each rope has got to hold up only

$$1500 \div 3 = 500 \textit{ pounds,}$$

hasn't it?"

That was evident and I agreed to it.

"The pull on a free rope will run the same from end to end of it, won't it?" he pointed out. "You can't have 1000 pounds pull on one part of a rope and 500 on another without hitching it in between. Well, this rope is free from the end that is hitched to the lower block to the end that is hitched to the horse and the pull is the same in it all over. Then the pull in the part that runs to the horse is only 500 pounds and, if he can pull 500 pounds, he can lift the weight. By putting in more sheaves and ropes you can reduce the pull still more.

"In Fig. 6 you have five ropes, and the pull on each would be only

$$1500 \div 5 = 300 \textit{ pounds.}$$

The last pulley on top is only a leading pulley. You wouldn't have to pull any harder if you could throw it away and pull straight up as shown, but you'd have to stand up in the air to do it. Fellows who don't think, though, often use a tackle backward and pull against the load when they might as well be pulling with it."

I didn't look as though I understood, so Uncle Pegleg sketched Fig. 7.

"Here is a load that it takes, say, 900 pounds pull to drag up the incline. Supposing they are using a three-rope fall (Fig. 5), the man would have to pull 300 pounds to move the load, but if they went up beside the building, Fig. 8, and pulled, they would have to pull only

$$900 \div 4 = 225 \textit{ pounds,}$$

because all four ropes would be pulling on the load instead of four on the hitch and three on the load, as in Figs. 5 and 7. Use wit and save work."

"How are you going to save work?" I asked. "It takes so many foot-pounds to raise the load, don't it, whether you use your wits or not?"

"Yes," he answered. "You're right. You can't save work but you can save effort. You can pull 225 pounds instead of 300."

"I've only got to pull out so many feet of rope to pull the blocks together or to make the load come up so many feet, haven't I?"

"Sure," answered the old man.

"Well, if I pull out, say, 10 feet of rope with a pull of 225 pounds on it I do 2250 foot-pounds of work; and if I pull out 10 feet of rope with 300 pounds in it I do 3000 foot-pounds."

"Right," assented Uncle Pegleg.

"Then," said I, "it takes 2250 foot-pounds to lift the weight if you stand facing it and 3000 foot-pounds to lift it the same distance if you face the way its going?"

"No, it takes the same number of foot-pounds to lift it the same distance whichever way you pull or stand."

"Now, say," I remonstrated, "there's only so much rope on that fall, is there? I've only got to pull out so many feet of it to bring the blocks together, or to move the load up so much. If the pull is different and the number of feet of rope the same, then the work I do is different to lift the same load the same distance. How do you make it out?," I asked.

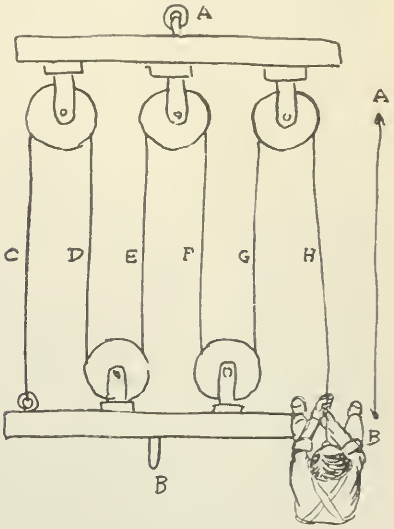


FIG. 9.

"You're wrong somewhere," said the old man, scratching his bald head. "It must be that the load doesn't come up so fast when you're pulling against it. Let's see," said he, sketching Fig. 9. "Suppose the man stands in the position shown and he pulls in 10 feet of rope. Now, if the anchor is at *A*, the distance *A B* from his hand to the leading pulley won't be shortened and the whole 10 feet will be taken up on the five strands *C*, *D*, *E*, *F* and *G*. They will be

$$10 \div 5 = 2 \textit{ feet}$$

shorter, so that a load hitched to *B* would move up 2 feet. If he had been pulling with a force of 100 pounds, he would have done

$$100 \times 10 = 1000 \textit{ foot-pounds}$$

and the load to oppose me would have been

$$1000 \div 2 = 500 \textit{ pounds.}$$

"Now suppose the anchor to have been at *B* and the load at *A*. The block with the three pulleys would have moved toward me. The distance *A B* would have been lessened. The 10 feet which he pulled would have been taken up by six ropes instead of five. Each rope would take up

$$10 \div 6 = 1\tfrac{2}{3} \textit{ feet}$$

instead of 2 feet, and the load attached to *A* would move up only $1\frac{2}{3}$ feet instead of 2 feet as it would if attached to *B*. If he still pulled 100 pounds and did 1000 foot-pounds of work, it would handle a load of

$$1000 \div 1\tfrac{2}{3} = 600 \text{ pounds}$$

instead of 500; and if there had been a load of only 500 pounds to handle, a pull of

$$500 \times \tfrac{2}{3} - 10 = 83\tfrac{1}{3} \text{ lbs}$$

would do it, because 500 pounds moved through 1⅔ feet is the work done on the load, and if the force that he uses is moved through 10 feet, it must be 83⅓ pounds to make the balance

$$500 \times 1\tfrac{2}{3} = 833\tfrac{1}{3} \therefore 10 \times 83\tfrac{1}{3} = 833\tfrac{1}{3}$$

I guess you'll find that these figures are in the same proportion as the others we had. What were they?"

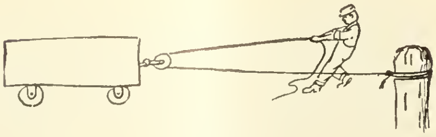


Fig. 10.

"It took 300 pounds one way and 225 pounds the other."

"Well," said he, setting the figures on his slide rule and poking it at me, "you see that

$$\frac{225}{300} = \frac{3}{4}$$

but

$$\frac{83\tfrac{1}{3}}{100} = \frac{5}{6}$$

We were dealing with a three-rope tackle in the first case and a five-rope in the second. The more ropes you use the less you gain proportionally by making one of them work instead of lead. I could have shown you better with a single pulley. In the first case (Figs. 1, 2, 3, 4), the pull on the hoisting rope has got to be equal to the load, because the pulley only leads and the load travels as fast as the rope does, but in this case, Fig. 10, the load only comes ahead 1 foot for every 2 feet of rope that the man pulls in so that his pull has got to be only one-half as much."

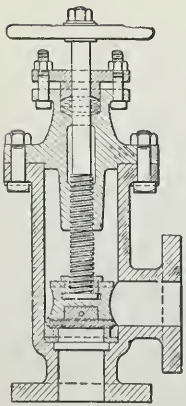
Blowoff Valves of the Angle Type

By Charles J. Simeon

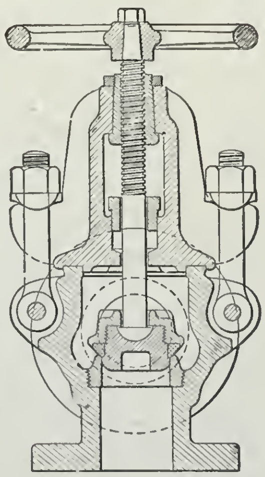
A concise description of the various types of blowoff valve of the convenient angle type, giving prominence to the features of construction and manipulation.

The angle type of blowoff valve was naturally developed largely because of the fact that, having to be placed below the boiler, an angle valve is frequently the most convenient form to use. The shape, however, has advantages that should not be overlooked, among which are ease of applying the principle of seat-washing, arranging for the joint surfaces of both valve and seat to be exposed as little as possible to the cutting action of the flow, and giving easy access for regrinding or renewal without disturbing the body. The feature of self-cleaning, as its name suggests, consists of means provided to insure that the seat will be automatically freed from foreign matter during the act of closing. It is an improvement that has been applied during the development. This end is usually attained by placing the seat ring in a deep recess in the body so that the main flow of water is nearly cut off in the act of closing, while the surface of the movable disk is still some distance from the seat. This leaves a narrow annular opening which is too small for the passage of large pieces of scale, but is still large enough to allow a stream of water to pass and wash away any sediment that comes to rest on the seat before the disk actually touches it. This feature is made a special point of in the Golden, Cadman, Lunkenheimer, Faber, Erwood, Pittsburg Foundry and Valve Company, Pittsburg Valve and Fittings Company, Kelley & Jones and Best. In all except the Faber the washing is accomplished in the same way; the water that flows past the seat is utilized to clean it.

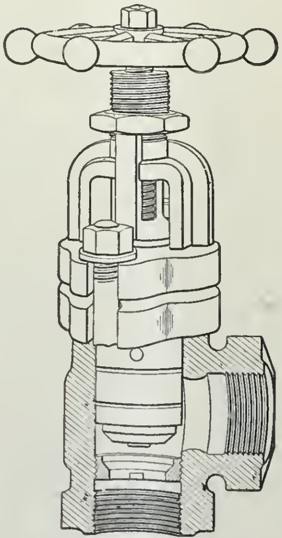
Several are alike in having seats and disks that are easily renewed. The disk in most cases consists of a heavy piston-shaped plug into which is dovetailed, or which carries, a renewable ring. The



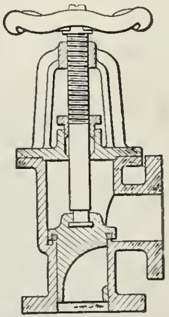
Robertson.



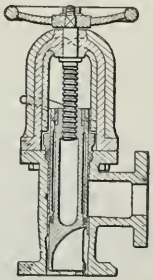
Morris.



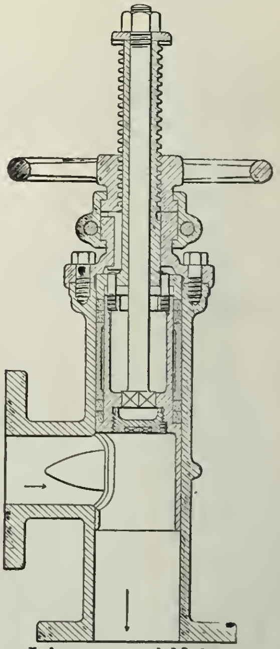
Powell.



Eastwood.



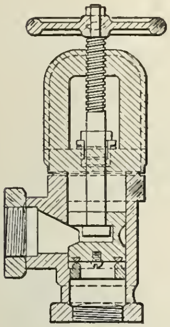
Nelson.

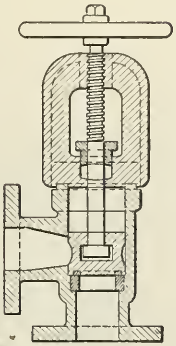


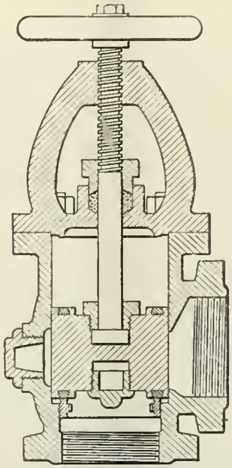
Johnston and Nolan.

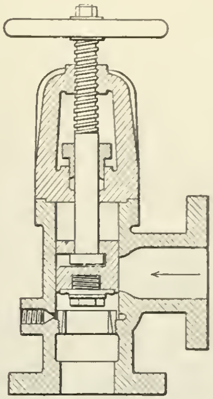
Power

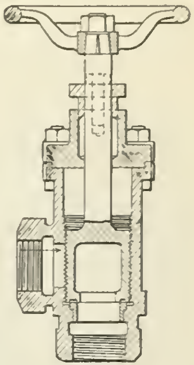
plug is made a snug fit in the cylindrical chamber of the body and deep enough to overlap the inlet opening when closed. This design, besides causing the valve to be guided to its seat, insures that any large piece of scale that may have stuck across the opening will be sheared off without damaging the working faces, and that no scale can prevent the movement of the valve by lodging between it and the bonnet.

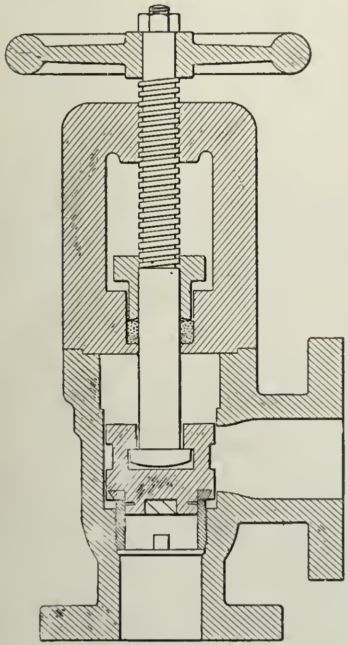

Pittsburg Foundry & Valve Company

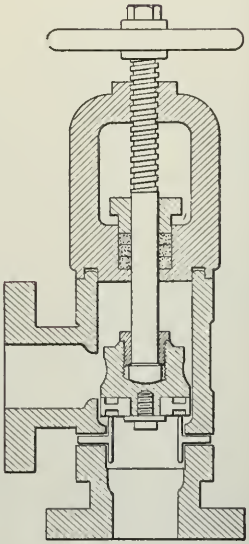

Best

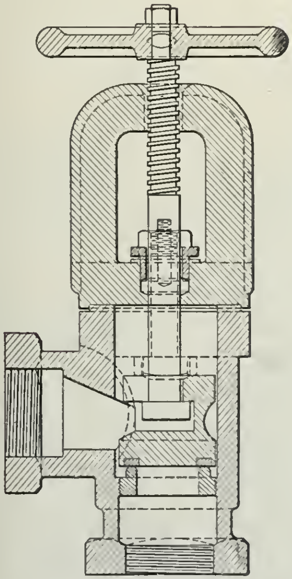

Lunkenheimer

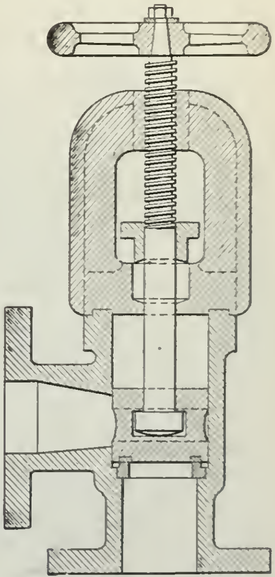

Faber


Erwood


Golden


Cadman


Kelley and Jones


Pittsburg Valve & Fittings Company

In the Cadman the valve ring is of soft copper with a hard-brass collar shrunk over it to prevent spreading. The seat ring is clamped between the lower and central castings of the body.

The Lunkenheimer, like the Cadman, has a reversible ring and a poke hole, ordinarily stopped by a threaded plug, opposite the inlet. through which sediment may be removed without dismantling while not under pressure.

In the Faber the general principle is the same but the actual cleaning is done by an auxiliary jet of clean water, steam or compressed air caused to play over the seat through a narrow annular opening just before the seat faces come together.

In the Erwood the seat washing has been reduced to the utmost simplicity. The body has an internal thread in which runs a solid plug the lower face of which has an annular groove filled with lead that makes a joint with a deeply recessed seat.

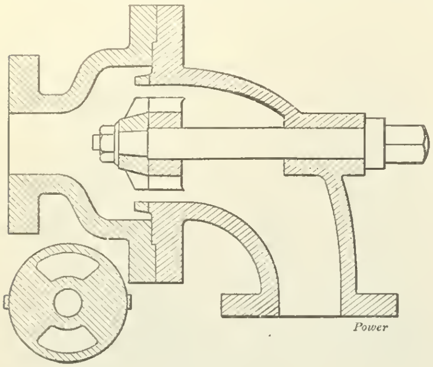
The Powell is similar in general construction to the others. The disk is renewable and reversible, and the seat ring has internal lugs to facilitate removal. There is no provision for seat washing, but the probability of the lodgment of scale on the seat is reduced by cutting it at a steep angle.

The designers of the Morris have worked on the assumption that a blowoff, from the nature of its use, is subjected to such severe treatment that serious wear of disk and seat cannot be avoided; and have turned their efforts toward making a valve that may be very easily dismantled for renewal. The body and cap are heavy castings with tongued and grooved joints held together by swivel eyebolts that hinge in the body. The disk has a renewable soft-metal ring and the seat ring is easily removable for refacing.

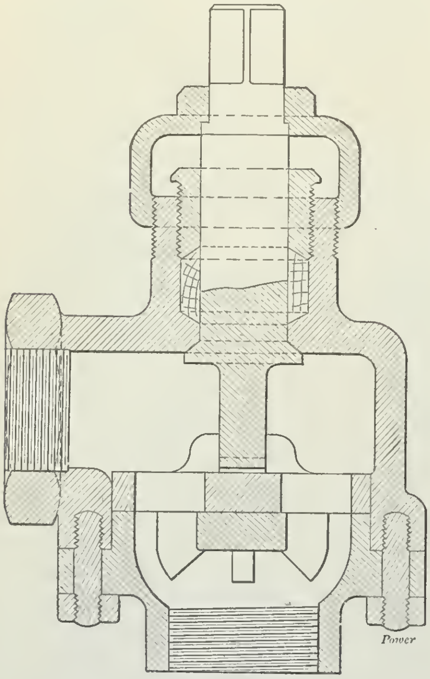
The Johnston & Nolan has a long hollow piston fitted with rings top and bottom. The body is a bored tube with the inlet about the middle of its length. In the closed position the rings make a tight joint above and below the inlet. In the open position both sets of rings are out of the flow, and an extension of the piston protects the surface of the tube. The rings may be expanded while under pressure by means of a nut at the top of the stem.

The Eastwood and the Nelson are similar in that each has a clear opening and that provision is made for protecting the seating surfaces. The Eastwood, however, is a disk valve with a projection which guides the flow past the seat.

In the Nelson the inlet and outlet are controlled by the reciprocating piston which in the open position provides a curved port. The piston has a V-shaped reccss directly over the curved port to retain the packing, and is adjusted by the gland sleeve, which is also provided with similar V-shaped recess. The gland sleeve is carried by the piston and adjusted by an octagonal nut which is screwed onto the upper end of the piston and is rotated by the double-handled wrench. This wrench cannot be removed from the valve without disassembling, but may be lifted so as to re-engage the octagonal nut in any position.



Penton



Riggin

The gland sleeve has a flattened portion to engage a similar flattened portion below the thread on the end of the piston, and engages with the guide ribs on the yoke which prevent the rotation of the piston so that the curved port of the piston will register with the inlet connection when in an open position.

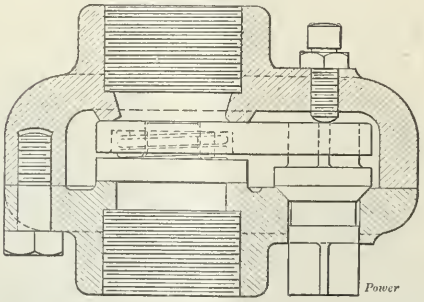
The travel of the piston is limited in its open position by a collar, and in its closed position by the lugs at the outlet connection.

The Penton and the Riggin have sector-shaped passages in the disk which register with similarly shaped openings in the seat when the stem is rotated to the open position. These valves, designed on the principle of the slide valve, are not unlike, except in outward appearance, to the Everlasting and the R. C., in which a port in the seat is opened and closed by a sliding disk operated by a lever on the outside of the case and requiring but a slight movement for the full range of travel of the disk.

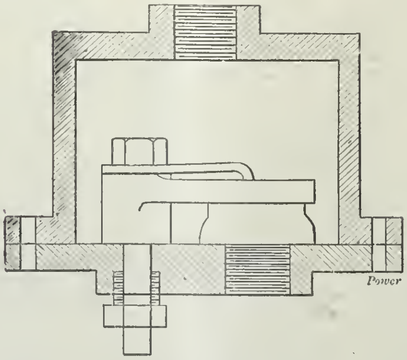
There is also another type in which the passage is opened and closed by a sliding disk or cover or by a rotating cap. In these the cover is held to its seat and the joint made tight by the pressure of the water.

In the Rothchild there is a hollow perforated plug rising in the center of the case, covered by a closely fitting split cap which is rotated by a forked stem connected to lugs on the outside of the cap. Through a small pipe rising through the stem of the rotating fork the oil reservoir in the top of the central plug may be replenished

In a class by itself is the Eclipse, in which the inner end of the stem is expanded to a sort of mushroom head which must be moved against the pressure in opening. It is so constructed that it may be ground while under pressure.



Everlasting

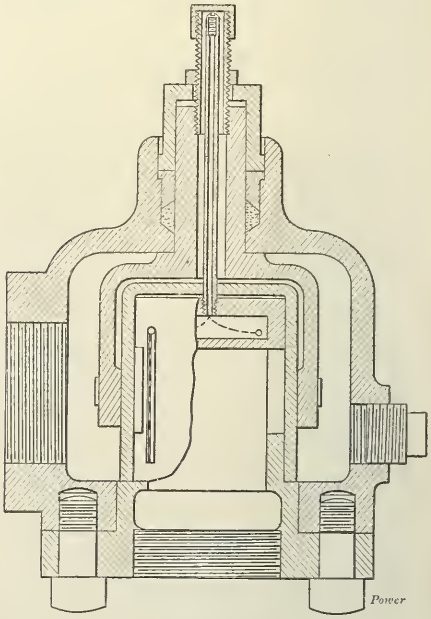


R. C.

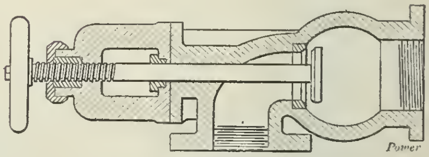
Nearly all the metals have been used in the manufacture of blowoff valves and cocks. The body castings are of brass or cast iron in small sizes, and cast iron, malleable iron or steel in the larger ones.

In many types the seat ring is a separate piece, screwed into the body and capable of renewal or repair. Owing to the greater difficulty of removing or regrinding this part, as compared with the disk, the ring is often made of a very hard material that will withstand the cutting influences to which it is subjected. The metal usually chosen is bronze, though hard brass is also used.

Much greater variety is seen among the materials chosen for the disks. In some instances a hard metal, such as bronze or iron, is used. In such cases provision is usually made for grinding the two together in place. In the majority, however, a soft material is used. This may be soft copper, rubber composition, or hard metal with rings of lead or babbitt metal cast into annular dovetailed grooves. Whichever form is chosen the intention is to give a surface that will adapt itself to any irregularities in the surface of the harder seat ring, and to cause as much of the wear as possible to take place on the easily removable disk instead of the seat, which must of necessity be a tight fit in the body. Such disks have the advantage of being very cheaply replaced.



Rothchild



Eclipse

With so many manufacturers, who are specialists on valves and all problems and conditions pertaining to their use, offering such widely different patterns it would be presumptuous to venture an opinion as to whether this or that particular valve is the best. Conditions of service and the user's judgment must determine the choice. It must not be forgotten that a valve is not gifted with sense or volition, and that none invented is fool proof. Much depends on the operator and the care which he uses.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Lightning Investigations

Ever since the experiments made by Franklin with his kite and key, investigators have endeavored to measure the amount of energy stored in a flash of lightning. For a long time about all that was known was that a lightning flash represents a transfer of energy from cloud to cloud or from cloud to earth, the transfer of energy taking place by means of a current of electricity. Later, by means of viewing the lightning flash in a revolving set of mirrors, and, still later by photographing a lightning flash on the film of a moving-picture camera, it was found that the stroke instead of being a discharge with a steady flow is pulsating in character, the frequency being usually very high. Faraday's discovery of the principle of induction furnished an explanation for the damage done to persons and property near the objects struck.

Last spring the Engineering Experiment Station of the University of Illinois began an investigation on the subject of protection against lightning. On a large open field on the university farm were erected twelve 40-foot telephone poles. Of these six were equipped with special devices for indicating the effect of lightning strokes at or near this system of poles. It is hoped by means of these instruments to gain some further insight into the laws underlying the phenomena of lightning. During the past summer there have been very few storms accompanied by lightning, and no strokes have hit any of the poles, yet indications have been given of the effect of strokes, at a distance. It is planned to add special recording devices to the present equipment and to continue the tests.

The equipment when complete will consist of a series of spark gaps shunted by resistance, inductance and capacity in series. The lightning may jump across the gaps or pass round them through the shunt circuits. Any action due to lightning will be recorded on a moving strip of paper driven by clockwork, so that the exact time of the electrical disturbance can be told from the record of the apparatus. From a knowledge of the length of the spark gaps and the record of which gaps are jumped by the lightning it is hoped that various characteristics of the stroke may be determined.

Three of the poles have been equipped with sharp needle points, and three with 6-inch brass balls. The first points were of steel and rusted badly. Silver-plated points were substituted, but they rusted nearly as quickly as the first set. On inspection all the points were found to be bent over as if softened by heat. The points were all bent so that they pointed to the northeast. Several theories have been advanced to explain this curious phenomenon. The work is being carried on by T. D. Yensen, of the Engineering Experiment Station corps.

Electrical Restrictions in Cleveland

The city of Cleveland on November 21 adopted an electrical code placing all electrical construction, interior wiring street lines for signal, fire alarm, power and other purposes under the control of the building department. This division of the department will be in charge of an electrical engineer with the title of city electrician, at a salary of $2000 per annum, and will comprise a chief inspector of electricity at $1500 per annum; a permit clerk at $1000, and six inspectors of electrical wiring and appliances. All of these officials are to be placed under bonds.

All wires except the trolley lines and arc-lamp poles are to be removed from a number of streets in the central portion of the city before June 1, 1911, and on another list of streets, further out, the removal and placing underground of lines must be commenced on January 1, 1911.

Largest Electric Motor in the World

The 6500-horsepower motor for driving the 60-inch universal mill at Gary, described in this department some months ago, was considered the largest motor in the world. A report comes from England, however, of a motor being built by the Siemens Brothers Dynamo Works, Stafford, which completely overshadows the Gary unit. The British motor is designed for direct current and is intended to develop 10,000 horsepower at 60 revolutions per minute with current at 920 volts. It is so designed that it can be operated at 1400 volts, which will be done later, when it will carry peak loads of 15,000 brake horsepower at 90 revolutions per minute. It will be coupled direct to a 36-inch cogging mill and a finishing mill, and will roll 2¼-ton ingots down to rails in one operation without reheating. It is stated that the operator will be able to reverse the motor from full speed in one direction to full speed in the other nearly 30 times per minute.—*Compressed Air Magazine.*

Automatic Generator-Motor

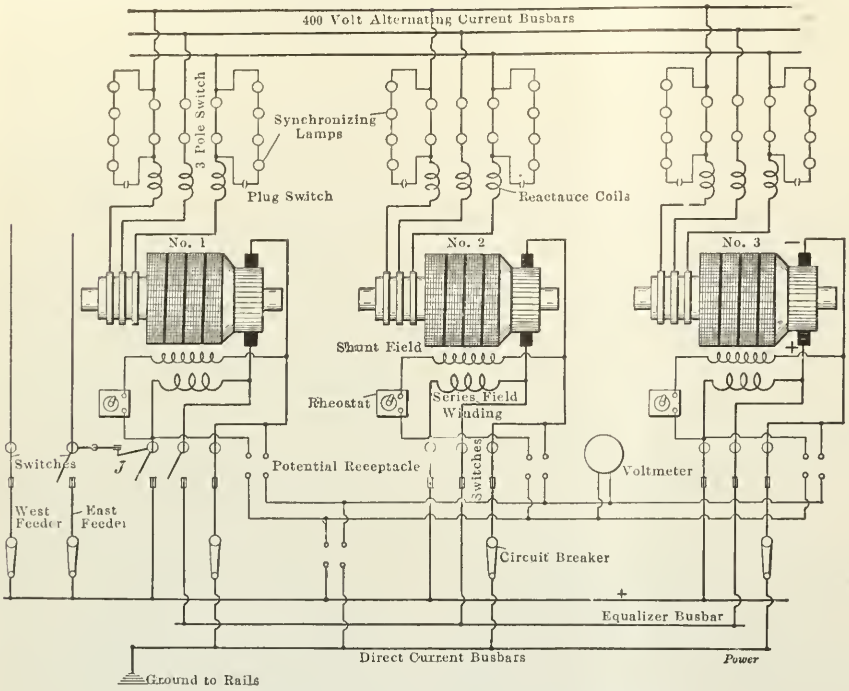
At the Laconia mills, Biddeford, Me., the superintendent, W. S. Mitchell, is using a 200-kilowatt three-phase alternator as a link between a limited water power and a larger generating plant to transfer power from the one to the other, in either direction, according to existing conditions. The alternator is belted to a pulley on the main shaft of a weaving mill which is driven by water power, and it is connected in parallel with the generators driven by other prime movers and with the large turbine-driven generator of the neighboring Pepperill mills. When there is ample water power, the generator at the weaving mill delivers power into the general system, but when the water is low or fails, the generator operates as a motor, taking power from the general system and either assisting its water wheel or driving the weaving mill entirely, according to the condition of the water supply.

LETTERS

Mr. Greer's Puzzling Converter Trouble

The phenomena cited in C. L. Greer's article published in the November 15 issue were due to the facts that the three converters are operated on the alternating-current side from a common set of busbars instead of from separate transformers for each rotary, as is the more common practice, and that the direct-current negative terminals of the three machines are tied together through the negative switches and circuit-breakers. With all positive switches open there is then a complete electric circuit between No. 1 and No. 3 as follows: from the negative brush of No. 3 machine, through the negative lead to the negative busbar, thence by way of the negative lead of No. 1 machine to the negative brush, through the armature to the collector rings, from the collector rings through the alternating-current leads to the alternating-current busbars, thence through the leads of No. 3 machine and through the armature winding to the negative brush of No. 3, completing the circuit. This same kind of circuit exists also between Nos. 1 and 2 converters.

A potential tending to cause a flow of current in these closed circuits may exist due to inequalities in the brush setting, the size of reactive coils and field strength of No. 1 and Nos. 2 and 3, and of a value depending on how closely these adjustments correspond. Therefore, with Nos. 2 and 3 running on the west feeder a local flow of current through these closed circuits between No. 1 and Nos. 2 and 3 might take place when the negative switch and circuit-breaker of No. 1 were closed. Should no disturbance occur when the negative switch and circuit-breaker of No. 1 were closed it might develop when the connection *J* was closed, due to the increase in the field strength of No. 1 converter caused by the flow of the load current of the east feeder through the series field winding of No. 1. No trouble would be expected between Nos. 2 and 3, because these machines were being operated directly in parallel and their voltages would therefore be carefully adjusted. No. 1, being considered as operating independently of Nos. 2 and 3, probably did not have its voltage closely adjusted to equal that of Nos. 2 and 3.



Connections of Mr. Greer's Rotary Converters

As the three converters equalize on the positive side, the ammeters would, of course, be in the negative lead and would show the current exchanged between No. 1 and the other converters. Had these machines been equalized on the negative side, with the ammeters in the positive leads, the ammeters would have given no indication of trouble and would have shown only the current taken by the feeders.

The phenomena cited by Mr. Greer would not have occurred had each converter been operated from a separate bank of transformers.

L. McKenney.

Wappinger's Falls, N. Y.

The armature of a rotary has only one winding, which is connected both to the commutator and to the collector rings; consequently, there is a continuous metallic circuit between the collector rings and the commutator. Now, keeping this fact in mind and remembering that the three converters are connected in parallel at the alternating-current terminals, without the intervention of transformers, and that all three machines were connected to the grounded side of the railway circuit, what Mr. Greer described as happening would be just what I should expect to happen.

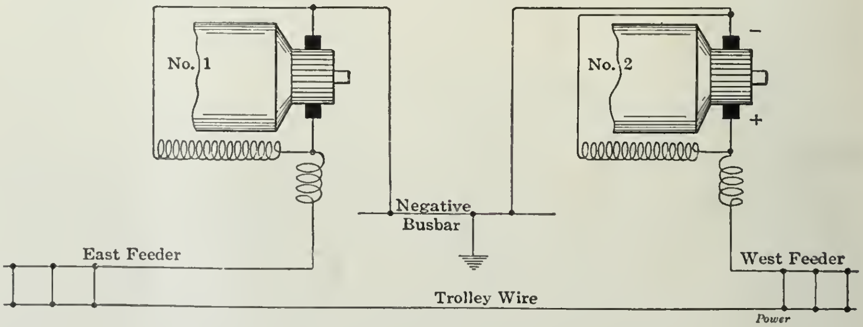


Diagram Illustrating Mr. Filkins' Theory

Nos. 2 and 3 converters were tied together with No. 1 at the positive direct-current brushes just as truly as if the positive switch had been closed, the tying together being accomplished through the armature connections to the alternating-current terminals, and it is evident that all necessary conditions for parallel operation existed except that the equalizer switch on No. 1 converter was open. Had this switch been closed it is likely that No. 1 machine would not have "hogged" the load to as great an extent as it did, but, of course, the machines would not have operated satisfactorily because, as Mr. Greer states, they would not do so even with the positive switch closed.

The result which Mr. Greer observed would not have occurred, of course, with direct-current generators or motor-generators, nor would it have occurred had separate transformers for each rotary converter been inserted between the alternating-current busbars and the converters.

F. M. Farwell.

Duquesne, Penn.

As Nos. 2 and 3 are connected directly in parallel they may be considered as one machine so far as No. 1 is concerned, or one of them can be ignored. In the accompanying diagram I have done this and have also omitted the alternating-current connections because they have nothing to do with my explanation.

Since all of the machines are connected to the track or grounded side of the railway circuit, and all of the feeders are connected to the trolley wire, which I assume to be continuous, the machines are connected in parallel even though the regular positive switch of No. 1 is open. The main connections, stripped of the "side issues" and unimportant details, are as shown in the accompanying diagram, so that the converters are paralleled on the direct-current side through the moderate resistance of the feeders and trolley wire. When No. 1 was connected to the east feeder, its voltage was evidently higher than that of the other converters and it forced current backward through them by way of the trolley wire and feeders. The series field windings made matters worse because the equalizer switch of No. 1 was open.

Earl R. Filkins.

Milwaukee, Wis.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Modifications of the Humphrey Pump*

By Herbert A. Humphrey

Two-stroke Cycle Pump

One other important modification of the single-barrel pump depends for its successful operation upon the fact that the burnt gases are so well cooled at the end of the long expansion stroke that they will not ignite an incoming combustible charge. It therefore becomes possible to have a true two-stroke-cycle pump. The action will be understood by reference to Fig. 10, which shows only the top of the combustion chamber. The combustion chamber has to be specially shaped so that the incoming charge, which may be preceded by pure air, displaces the burnt products and mixes as little as possible with them. Thus, in the figure, the admission valve *A* is located at the top of a tall but narrow part of the chamber *B*, in which the full-charge volume extends down to the level *c c*. A number of exhaust valves *E* lead to a common exhaust outlet *O*, which may be fitted with a nonreturn valve, or each exhaust valve may carry a light nonreturn valve on its spindle as shown. The level at which expansion reaches atmospheric pressures is, say, *f f*, but this level having been reached by the water, its further movement draws in fresh combustible mixture till it occupies the space down to *c c*, and the liquid level has fallen to *g g*. The column of liquid then returns and drives the exhaust products through the valves *E* (which opened by their own weight) until these valves are shut by the water. The kinetic energy acquired by the column is then spent in compressing the fresh charge, which is ignited to start a new cycle. Thus each outstroke is a working stroke, and no locking gear is required on the valves. A higher compression pressure is obtained with this arrangement than with the simple pump, and consequently higher efficiencies with the same lift.

The Two-cylinder Pump

Fig. 11 illustrates the elemental parts and arrangement of the double-barrel pump, corresponding to a two-cylinder engine. There are two combustion chambers *A* and *B*, in which explosions occur alternately. In each barrel the usual cycle of compression, explosion, expansion, exhaust and the taking in of the new charge, is carried out, with the four strokes of unequal length. In this case, as in that of Fig. 10, the water column has only one outward and one inward movement per cycle, but the expression "two-stroke" would here be misleading, unless applied only to the water column, and not to what occurs in the combustion chambers.

For example, assume that a compressed combustible charge exists in the top of chamber *A*, and that *B* is full of water. Explosion and expansion in *A* cause the outward propulsion of the water column, and when atmospheric pressure is reached the exhaust valve of *A* opens, water flows in to follow the column and to rise in *A*, expelling burnt products; there being insufficient pressure to maintain the water in *B*, the level therein falls, causing an intake into *B* of a fresh combustible charge. All this takes place during the outward movement of the column of water, and then the column coming to rest commences to return toward the chambers. The remaining burnt products in *A* are sent out through the still open exhaust valve by the returning water, whereas the new charge in *B* cannot escape because the admission valve is now shut, under the action of its spring. Consequently the column rises at first in *A* until the exhaust valve is shut, and then in *B*, compressing the new charge. Ignition in *B* starts a fresh cycle with the functions of *A* and *B* exchanged.

There are important differences between the action of this pump and of the single-barrel pump first described. The new charge is taken in merely by water falling under the action of gravity, and does not depend upon the compression and expansion of an elastic cushion. The clearance spaces at the tops of the combustion chambers may therefore be reduced to the very small volume needed to bring easily to rest that portion only of the column which rises in the chamber, the main body of the column being brought to rest by the compression of the new charge. Again, for a given delivery head, the compression pressure at-

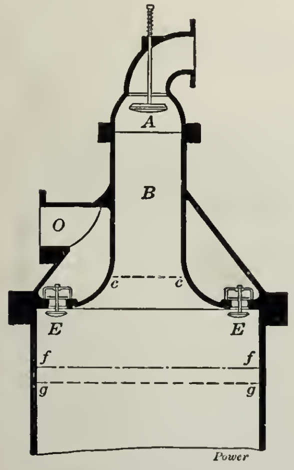


Fig. 10. Two-stroke Cycle Pump

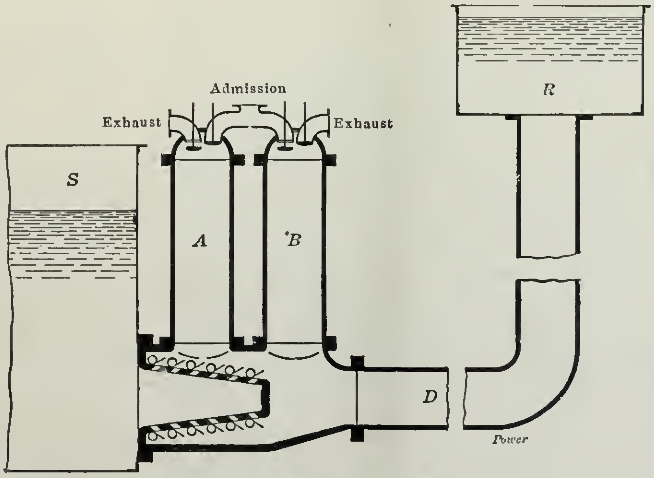


Fig. 11. Two-cylinder Pump

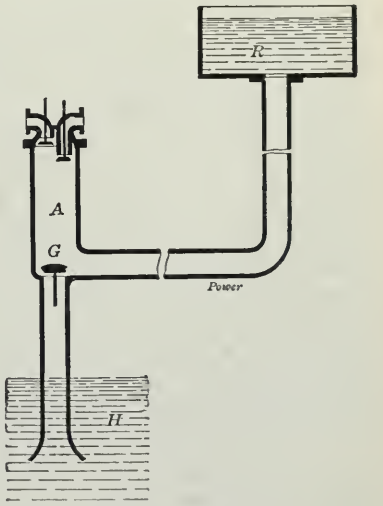


Fig. 12. Suction-lift Pump

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*Second part of abstract from a paper read before the Manchester Association of Engineers. First part appeared in the December 13 number.

tained is higher in the double-barrel pump, because the returning column gains speed while exhausting the last part of the burnt products, and thereby obtains a store of kinetic energy before starting the compression stroke proper.

The advantages of the two-barrel over the simplest of the single-barrel pumps, briefly stated, are higher explosion pressure for the same water lift, due to reduced clearance resulting in less contamination of the new charge by residual burnt products, and to higher compression pressure; more cycles in unit time, and greater output for a given total weight of pump and play pipe.

Suction-lift and High-pressure Pumps

So far we have considered only those pumps which have a submerged suction and a limited delivery head of 40 feet. We will now take a brief glance at a few of the types capable of working with a suction lift, and others for delivery against high heads.

Fig. 12 is a diagram of one kind of suction-lift pump, the cycle of which is as follows:

1. Explosion of compressed combustible charge in *A* driving water to the high-level tank *K*; continuation of expansion stroke below atmosphere owing to the existence of nonreturn valve in exhaust pipe. Consequent lowering of pressure till water is raised from the supply tank *H*, past the valve *G*, until the water comes to rest and the valve *G* shuts.
2. Return movement of the water column toward the chamber *A*, giving exhaust and cushion stroke.
3. Cushion expansion, and inlet of a measured combustible charge into *A*, and expansion of this charge below atmosphere.
4. Second return of the water column toward *A*, giving the compression stroke.

The measured charge involved in step 3 of the cycle may be obtained in many ways, one of the simplest being by the use of gas bags, the diaphragms of which are attached to valves, and on being drawn in actuate the valves to close the outlet. The use of a measuring device can, however, be obviated by employing the "rejected charge" device to be described in connection with air compressors.

A two-barrel suction pump is illustrated by diagram in Fig. 13. The combustion chambers *A* and *B* are connected to the play pipe *D*, but the water-suction pipe is at the opposite end of the play pipe, and in addition there is an air chamber *C*, with check valves *N* and *M*. The cycle will be readily surmised. On explosion in *A* the water column moves outwardly from the chambers and compresses the air in *C* until the pressure is sufficient to lift the valve *N* and pass water into the high-level tank. On the column coming to rest the valve *N* shuts and the compressed air in *C* expands again, imparting velocity to the column which carries the expansion below atmosphere; this lifts water from the low-level tank at the same time that exhaust from *A* and compression of the new charge in *B* take place. The cycle is then repeated.

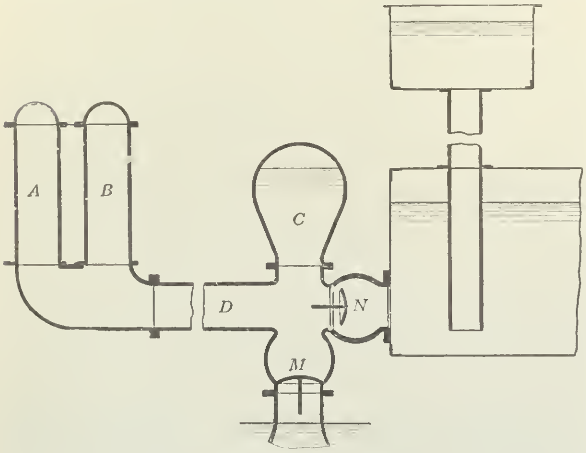

Fig. 13. Suction Lift and High-pressure Delivery

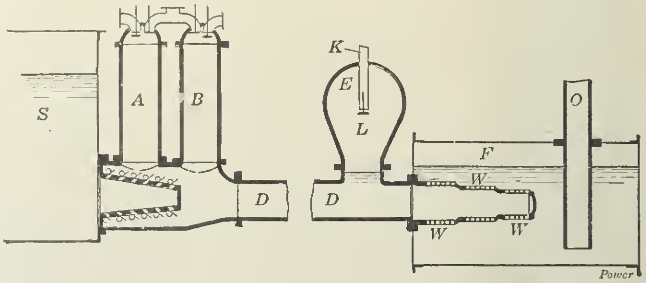

Fig. 14. Two-cyclinder Pump for High Delivery Head

Any Humphrey pump, whether single or double barrel, may be converted into a high-pressure pump by means of an air vessel fitted with valves and called an "intensifier." The idea is to first allow the water column to gain velocity and then to utilize its kinetic energy to compress an elastic fluid and deliver water under the pressure to which the elastic fluid has been compressed. In Fig. 14 *A* and *B* are the barrels of a two-barrel pump, and at the end of the play pipe *D* there are two air vessels *E* and *F*, the latter being large enough to give a continuous flow at the outlet *O*, and to maintain a practically uniform pressure. The smaller air vessel *E* is fitted with a downwardly projecting pipe *K*, open to the atmosphere at the top and carrying a valve *L* at its lower extremity arranged to close under the action of the rising water. The cycle starts with explosion, all valves except *L* being shut and the water level as shown. While the water level in *E* is rising, air is merely discharged into the atmosphere, and as no work is being done by the column of water it gains velocity until the valve *L* is shut by impact. Imprisoned in *E* there is now a definite quantity of air which suffers compression until its pressure reaches that at which the high-pressure water valves *W* can open and allow the remaining kinetic energy of the column to force water into *F*. The valves *W* close when the column comes to rest, but there remains enough energy in the compressed air in *E* to give, by expansion, the return flow, which causes exhaust in *A* and compression of the fresh charge in *B* to start a fresh cycle. When the water level falls below the valve *L* it opens and air is admitted into *E* for the rest of the return stroke.

It is easy to see that if the pipe *K* is made vertically adjustable with regard to *E*, the point of the cycle at which *L* shuts can be varied, and more or less air can be entrapped in *E*. But the amount of energy stored in this air will also vary with its quantity, for we assume that the degree of compression remains constant, and is indeed fixed by the pressure maintained in *F*. Consequently the ratio of the total energy of the working stroke to the energy stored in the compressed air in *E* can be made anything desired; in other words, any compression pressure of the new charge in *B* can be obtained, independent of the water lift. The advantage is obvious, for compression pressures equal to those in modern gas engines can be employed with a corresponding increase in thermal efficiency. Further, by manipulating the position of the pipe *K* a given pump can be made to meet any conditions as to hight of delivery, for if the head increases, *K* can be raised so that the energy stored in the air in *E* remains the same, there being now less air but at a higher pressure.

An important development of the arrangement shown in Fig. 14 is of practical interest. Remember that at each cycle air is drawn into and rejected from the vessel *E*; then suppose *K* is connected to a supply of combustible mixture instead of opening into the atmosphere. This makes an automatic auxiliary pump for taking in mixture and discharging it under pressure, and if the discharge is into a reservoir from which the combustion chambers *A* and *B* can be supplied, we have a means of quickening the cycles and greatly increasing the output of a given size of apparatus. It is convenient to replace the vessel *E* by two vessels, one for air and one for gas, so as to maintain the combustible constituents separate until they enter the combustion chambers. If the first portion of the outstroke of the water column is allowed to reject the surplus air and gas back to the sources of supply, then the action throughout the cycle is precisely that described when using the single vessel *E*, except that a larger proportion of the total energy is absorbed in the compression of air and gas but the excess is given out again during the expansion of the precompressed charge in either *A* or *B*.

The chief advantage of the form just described lies in the more rapid working; there is no need to wait for the water level in *A* or *B* to fall under the action of gravity when the charge is being taken in. In fact, the apparatus becomes practically independent of the water level on the supply side. A 1000-horsepower Humphrey pump now under construction in Germany will operate in the manner just described, the result being that the dimensions are very moderate, and the pump itself occupies no more space than a 1000-horsepower tandem gas engine.

Notes on a Large Gas Producer*

By Nisbet Latta

Generally speaking, heavy-duty gas engines have been confined to operations where either natural gas or blast-furnace gas was available; hence, before they can be generally adopted some method for the economical manufacture of a satisfactory gas in large volumes must be assured. The difficulties of such manufacture have been many, and a few of them are herewith cited.

The initial introduction of gas producers into the United States was unfortunate and the failure of the greater number of the first installations cast upon the producer a cloud of discredit which has never been removed entirely from the public mind. This failure was due, in most part, to the wrong hypothesis that because this apparatus had been successful in England and on the Continent it must necessarily be so in this country. The fallacy of this idea was due to the single variable, the nature of the fuel used. The easily fusible ash contained in the English and German coals permitted a high rate of combustion, averaging 18 to 20 pounds per square foot of cross-section. When operation at this rate was attempted here, it was found to be impossible to control the fire bed, the resulting temperature being such as to flux and finally fuse the ash and to produce other conditions inconsistent with satisfactory operation.

To avoid this condition of affairs, the rating of American producers has steadily been decreased until the best practice now averages from 8 to 9 pounds of fuel gasified per square foot of grate surface. This change has rectified to a great extent the difficulties met with in the introduction of the apparatus upon this side of the Atlantic. Producer-gas plants, however, are relatively small in size, while the heavy-duty gas engine for power purposes is essentially an apparatus of large capacity and demand. The average gas producer in common practice today does not exceed 500 horsepower in size. Even at this size, and invariably beyond it, mechanical troubles occur which are difficult if not impossible to remedy. For instance, in producers of great diameter there are different heat zones or strata, due to the difference of temperature between the center of the apparatus and the sides, which are cooled to some extent by radiation. Moreover, large diameters cause almost insurmountable difficulties in stoking and prevent the possibility of maintaining the fuel bed at a high efficiency.

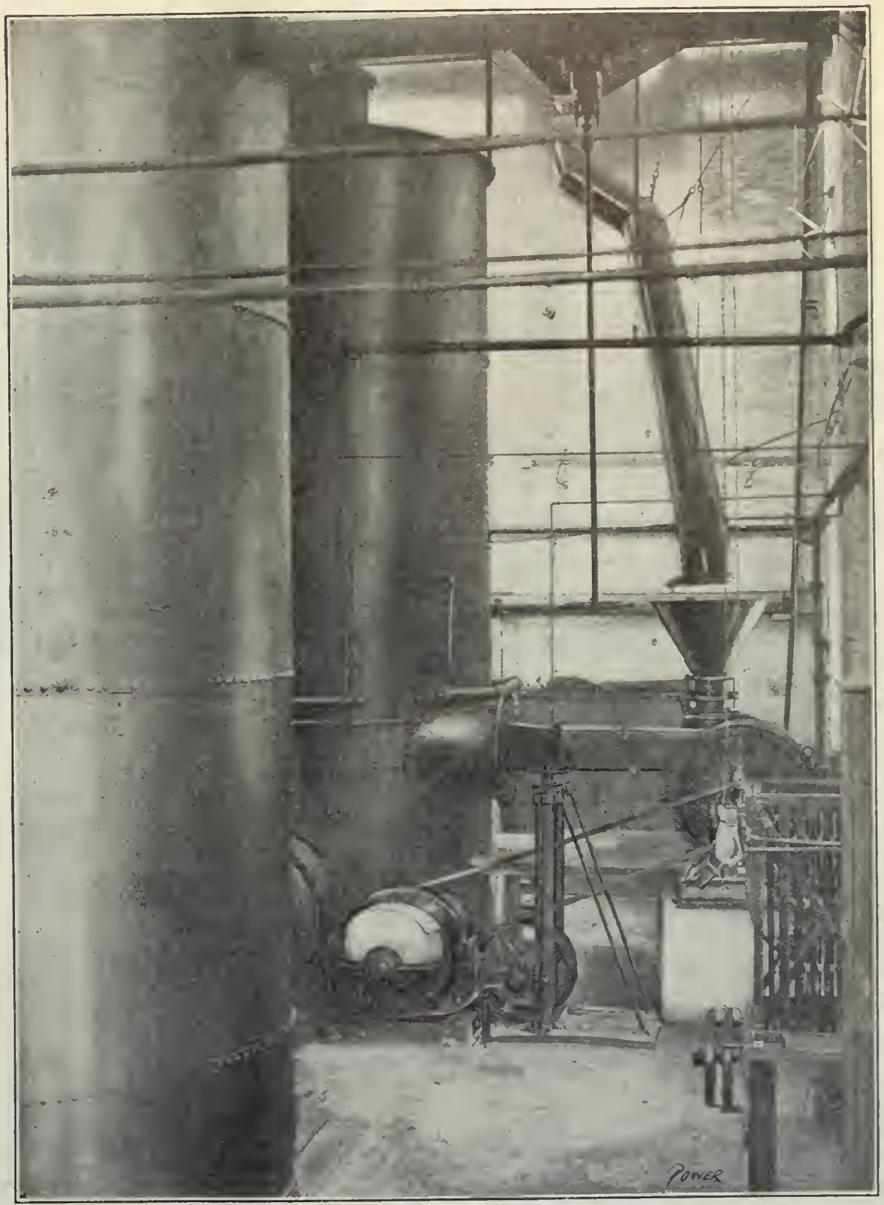

Fig. 1. Part of Hirt Apparatus, Showing Pulverizer at the Right

Again, the labor of cleaning is immeasurably increased. This condition of affairs is peculiarly emphasized in the case of caking or coking coals containing a high percentage of fines (coal under ¼-inch mesh) which tend to form a homogeneous mass, packing or caking the bed. The limit of fines in the case of pressure producers seems to be about 15 per cent. for caking coal and about 40 per cent. for free-burning coal. With the latter 65 per cent. of fines is the maximum obtained in commercial operation. In some coals the ash fuses at an early stage of combustion, while others form a densely packed or "mucky" bed, almost like molten asphalt.

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*Abstract of a paper read before the American Society of Mechanical Engineers, December, 1910.

The difficulty of operation under these conditions has hitherto been met by some expedient such as mechanical agitation; agitation by a reverse blast of compressed gas through the generator is also used. The backward blast lifts the bed and redisposes it, blows out the ashes from the interstices of the fuel and makes the fuel surface more active. This method, especially in a producer of moderate size, can unquestionably be commended.

Nevertheless, the fact remains that, although the medium-sized producer, of, say, 750 horsepower, has been developed with a reasonable degree of success, there is as yet no gas-producing apparatus which may be said to be a consort for the heavy-duty steel-plant type of gas engine of 3000 to 5000 horsepower.

It has been claimed that the solution of the matter consists in the use of a number of producer units connected in batteries or multiples, analogy being made with steam-boiler practice. This analogy fails by reason of the essential difference in the character of the service. For instance, within certain limits the question of boiler operation means merely the delivery of steam at a certain pressure, the pressure regulation between the boilers being a matter which is comparatively simple. The efficient operation of a gas engine, however, necessitates the delivery not only of a continuous supply of gas, but of gas of uniform composition. This condition is made even more complex by a variable load, and the process of operating in harmony a multiplicity of producer units is more difficult in practice than it would seem theoretically.

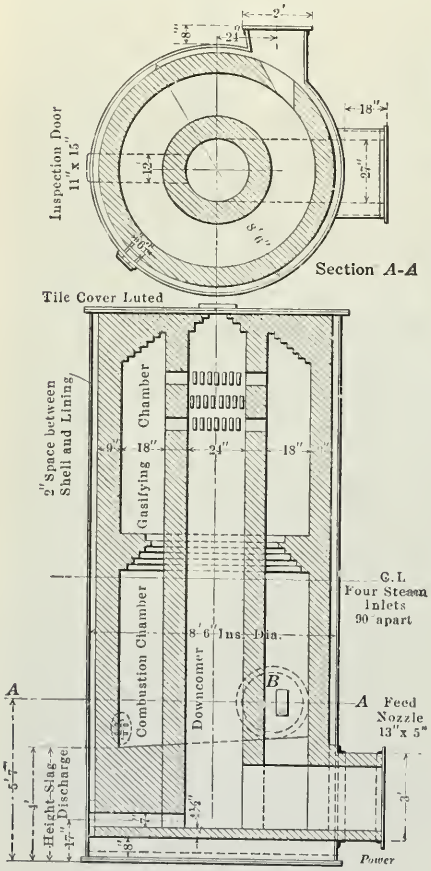


FIG. 2. HORIZONTAL AND VERTICAL SECTIONS OF GENERATOR

The foregoing facts explain the reason for starting, about a year ago, a series of experiments on a large mechanically operated producer. Although incomplete as yet, they tend to show the possibilities of the manufacture of gas on a large scale at a low cost.

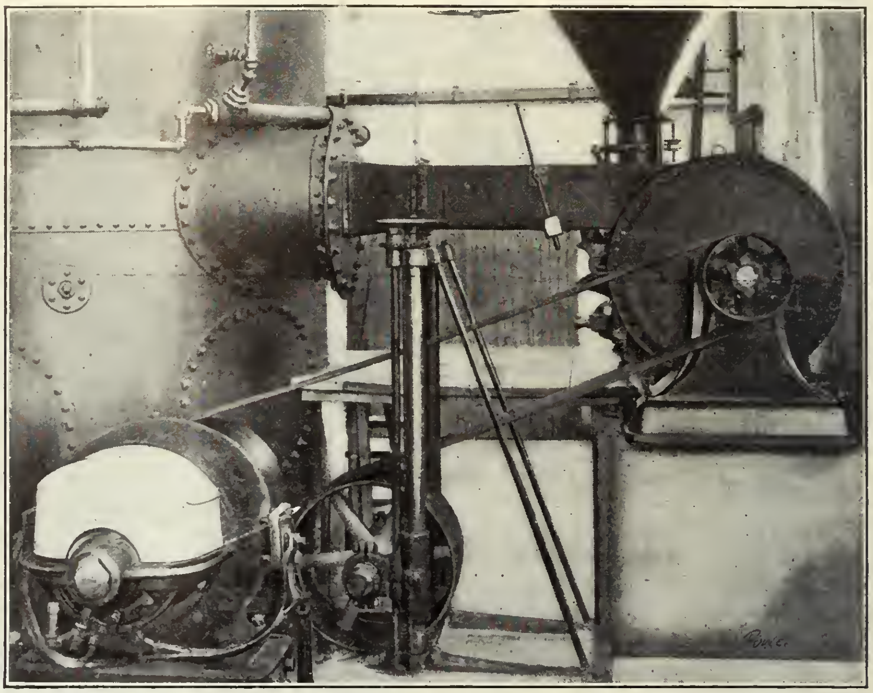


FIG. 3. CLOSE VIEW OF PULVERIZER

The apparatus selected was of the pulverized-fuel type, the one herein described having originally been designed by L. H. Hirt. This type of producer is hardly an innovation, having been used in France by Georges Marconnet, a French engineer, to supply gas fuel to engines for the generation of power and to open-hearth furnaces.

TABLE 1. ANALYSES OF PRODUCER GAS.

VOLUMETRIC PERCENTAGES.

											*
Carbon monoxide	23.7	27.5	26.5	27.0	24.5	15.85	13.52	12.20	18.2	13.8	18.9
Carbon dioxide	3.9	2.8	2.9	5.0	3.9	9.20	8.10	7.60	4.9	8.0	9.4
Hydrogen	10.0	11.5	11.0	9.5	9.8	6.17	11.51	10.50	12.2	10.4	12.2
Methane	......	......	......	......	3.4	4.09	5.17	3.20	2.1	2.5	......
Oxygen		......	......	......	..	1.40	0.30	0	0.1	0	
Nitrogen	.	......	......	......		63.29	61.40	66.50	62.4	64.8	......
Illuminants	......	......	......	......	..	......	......	......	0.1	0.5	......
B.t.u. per cu.ft., total	......	......	......	......	......	......	......	......	128.89	119.70	

*Steam admitted to upper chamber.

The Hirt apparatus consists of a pulverizer, a retort or generator and a tower scrubber or other gas-cleaning device. Fig. 1 shows the pulverizer and part of the generator and scrubber. Fig. 3 is a nearer view of the pulverizer. The fuel is first broken to a reasonable uniform size, of say ¾ inch, by passage through a crusher or griffin mill, from which it usually passes through a storage hopper to the pulverizer. Here it is ground to a satisfactory degree of fineness in the presence of air. Not only is a high degree of intimacy on the part of the elements secured, but by this arrangement coal containing a maximum of 12 per cent. to 15 per cent. of moisture may be ground without predrying. The mixture lacking sufficient air to permit complete combustion, is then admitted tangentially into the producer or retort through opening *B*, Fig. 2. After combustion, the gas takes an upward course through the gasifying chamber and passes down the central passage, escaping at the bottom to the scrubber.

Heretofore the difficulties experienced in operation with pulverized fuel have been largely the high temperature of combustion and the impossibility of maintaining a refractory material against such intense temperature. In the operation of this producer, however, the combustion is incomplete. In starting up the producer it is necessary to permit complete combustion for a sufficient period to raise the linings to the temperature of gasification, this temperature being subsequently maintained by the reaction of gasification in the natural process of manufacture within the apparatus. According to Alexander M. Gow,* 4450 B.t.u. are liberated when a pound of carbon is burned to CO; as the weight of the entire products of combustion of one pound of carbon burned to CO is 6.76 pounds, on the basis of their average specific heat, the theoretical temperature attained would be practically 2240 degrees Fahrenheit. The oxidation of carbon begins at the low temperature of 400 degrees Centigrade (752 degrees Fahrenheit), the reaction being complete at 995 degrees Centigrade (1823 degrees Fahrenheit). These facts merely serve to indicate that the heat supplied by the exothermic reaction

$$2\ C + O_2 = 2\ CO$$

would be sufficient for the operation of the producer.

TABLE 2. PROXIMATE ANALYSES OF COAL.

PERCENTAGES BY WEIGHT.

		*	
Moisture	0.52	1.20	0.73
Volatile matter	17.50	32.95	17.52
Fixed carbon	72.48	59.64	76.80
Ash	9.50	6.21	4.95
Sulphur	1.24	......	......
B.t.u. (dried coal)	....	13,925	14,550

*Analysis of ash from this coal: silica, 43.62; aluminum and iron oxide, 42; lime, 5.74; magnesia, 2.22.

Perhaps the only serious difficulty met with in the continuous and uniform operation of the apparatus was caused by the construction of the producer upon the hypothesis of the inventor that all coals could be carried up to a temperature at which the ash would fuse, so that it could be removed in the form of slag. This was an error which soon developed. It has been demonstrated that only fuels containing easily fusible ash can be handled in such a manner, and that these fuels are the exception rather than the rule. The apparatus having been constructed upon this premise, the removal of ash was a clumsy and inconvenient performance. It was necessary to accumulate an ash bed sufficiently deep to seal the producer and then gradually and carefully withdraw it from the bottom. This difficulty, however, would be easily remedied in a producer designed with the knowledge of the foregoing experiments. Either a water-sealed bottom or a conical dumping bottom would be entirely practical.

In the experiments herein cited the pulverizer was driven by an electric motor. In commercial operation, however, the sensible heat of the effluent gases would be more than sufficient to generate steam to drive all auxiliary apparatus. The interposition of a waste-heat boiler between the producer and the scrubber, such as is used in the Loomis-Pettibone apparatus or in connection with water-gas sets, would unquestionably give satisfactory results.

Bituminous coals of a wide range were used and the operation of the producer with fuels containing up to 30 per cent. of ash and 15 per cent. of moisture was demonstrated. Some typical analyses of the gases, as made by the Pittsburg Testing Laboratories, are given in Table 1.

There is, however, a definite ratio between the fuel used, the degree of fineness to which it is ground and the time the gas remains within the producer. That is, adjustment should be made to suit the chemical analysis and the physical nature of the fuel. The length of the downcomer should depend upon both of these elements, the length varying, in all probability, inversely as the percentage of volatile matter contained in the coal.

Transactions, Engineers' Society of Western Pennsylvania, May, 1903, p. 14.

A New High Speed Diesel Engine

By S. Snuyff

There was exhibited at the Brussels exposition a 600-horsepower Diesel engine, built by the Holland Engineering Works, Amsterdam, which embodies two important and several minor departures from previous practice in the construction of this type of oil engine. The chief differences are in the use of a crosshead and guide similar to those of a steam engine, and a short, hollow water-cooled piston, which is, of course, closed all round. The piston is secured to a flange on the piston-rod end with studs and is provided with twelve packing rings to keep in the high pressures of compression and combustion. The interior of the piston is kept in communication with the circulating-water supply and discharge by means of two little brass pipes, which are fastened to the under side of the piston and slide up and down in two stationary brass barrels mounted on the engine frame. The barrel through which the water is delivered to the piston is provided with a stuffing box, because this water is under slight pressure, but the delivery barrel is not packed.

The cooling of the piston has been adopted because of the large diameter of the piston and the high speed. There are several reasons why the designers have found it desirable to adopt the cross head and guides, but the principal objects were to avoid lateral pressure between the piston and cylinder walls and to make the wristpin end of the connecting rod fully accessible and free from the influence of the heat in the cylinder.

The crosshead has one block or slipper running between guides on both sides, and the connecting-rod end is forked, straddling the crosshead block. It is rather interesting to note that the first Diesel engines, designed by Herr Diesel several years ago, were built with crossheads and guides.

The valve motion too is of improved design. Instead of using cams to actuate the short rocking levers which open the inlet and exhaust valves, eccentrics are used. This was found advisable with the new high speed engine in large powers in order to obtain silent operation. The engine compares favorably with the steam engine as regards quiet, steady running with varying loads. The cyclic irregularity is $\frac{1}{200}$.

The engine at the exposition was direct connected to a direct-current generator of 350 kilowatts output supplying electrical energy for power and lighting purposes to the Holland section of the International Machinery Hall. There are four cylinders 19$\frac{11}{16}$ inches in diameter; the stroke is 25⅝ inches and the speed 210 revolutions per minute. The piston speed, therefore, is 900 feet a minute.

The guaranteed fuel rate of the new engine, using oil of 18,000 B.t.u. per pound, is 0.37 pound per brake horsepower-hour at full load, 0.429 at three-quarters load, 0.495 at one-half load and 0.771 at one-quarter load. The corresponding thermal efficiencies are 38.2 per cent., 33, 28.6 and 18.3 per cent.

They hev a little Irish feller ilin' over t' th' power house by th' name uv Jimmy Maloney. Th' brush holders uv wun uv th' dinymos got t' runnin' hot tother day an' th' super'ntendent sed th' "pig tails" wuz busted an' they'd hev t' git sum new wuns. He told Jimmy t' chase hisself down t' th' shop an' see ef he cud scare up 'bout haf a dozen. Jimmy wuz gone 'bout two hours an' kum back puffin' like a foundered steer. He hed th' pig tails all rite, an' they wuz th' real thing. Sed he'd went t' every dumd butsher shop in th' town an' cudn't find any an' hed t' go out in th' kentry t' a farmer's pasture t' git 'em. Sed he hed a heluva time ketchin' sum uv th' pigs but he'd maniged t' round up six uv 'em an' sep'rate 'em frum their tails arter he'd purty dumd night killed hisself runnin' the pesky critters down.

Readers with Something to Say

Home Study

Now that the long winter evenings are with us, the engineer should settle down to hard work and set aside a certain number of hours each day for study.

Successful home study requires that the reading be done in a systematic manner and that the student select the branches of engineering in which he particularly needs instruction, and then exhaust the subject from every point of view. He may decide on a purely theoretical course, including physics, chemistry, the elements of steam and electrical engineering, or he may select a mathematical course, analyzing and applying the formula of engineering, from the coal pile to the apparatus where the energy is finally utilized.

While employed in a plant containing quite a variety of machinery, I went through such a course, and so impressed the formulas and constants on my mind that it is only necessary to recall the particular piece of apparatus to remember the method of making various calculations.

If a practical course is decided on, it should start at the boiler room and systematically take all apparatus in order, dealing with design, construction, operation and repair and the description of all prominent makes of machinery and the work for which each is best suited. It may be desirable to take some one subject such as "boilers" and study it from a practical, mathematical and theoretical point. In preparing such a course, a complete outline should be made forming a synopsis from which to study. It can be made up by dividing and subdividing the subject down to the smallest detail, using the index of textbooks and engineering magazines to select the items. The outline can best be kept in a notebook, using a line for each item and skipping every other line for filling in items found later.

Starting with such a guide, take each item and look it up in all available literature, making a note of the article or articles which best cover the subject. It is well to read all the descriptions of central stations and isolated plants possible and note the types and capacity of the apparatus selected. Catalogs of manufactures should be obtained and the design of the apparatus studied. Such a course carefully carried out should enable anyone with an ordinary education to become master of any such subject if he employs his time industriously and with intelligence.

Lewis C. Reynolds.

Willard, N. Y.

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Safety of Dished and Flanged Heads

Owing to the failure of a number of dished and flanged heads during the last ten or fifteen years, an unwarranted prejudice has sprung up against this form of head. I can advisedly say prejudice, for, as a matter of fact, there is nothing wrong with this type of head when properly made. I believe that it can be demonstrated that the whole trouble is in the flanging and comes from an effort on the part of the workmen to make the head enter the shell easily when assembling. To do this, too much taper is given the flanged portion of the head, and when riveted in place this produces a strain tending to straighten back the flange, as shown in Figs. 1 and 2, and it is liable to start a fracture as at *A*, Fig. 2. This fracture may not show up for some time, but the constant flexure, caused by the variation in pressure, will gradually cause crystallization. Another cause of fracture is flanging the head too far over and then knocking it back, and especially this is generally done in hand flanging and when the material is not sufficiently heated.

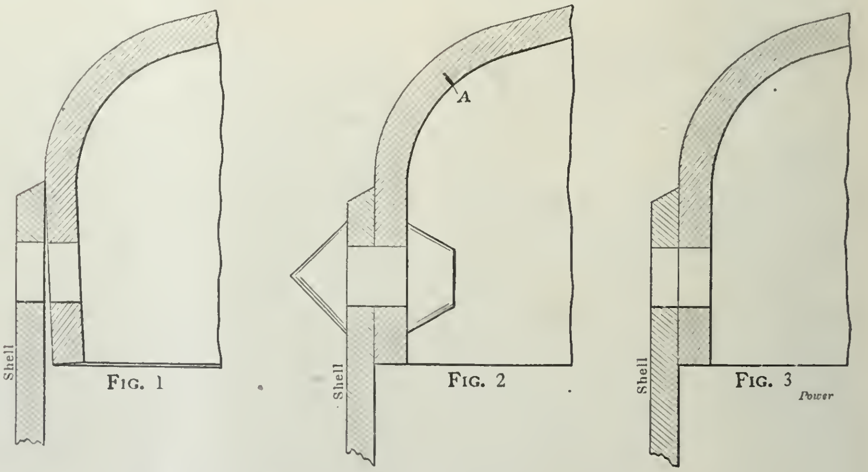


Fig. 1 Fig. 2 Fig. 3

Improper and Proper Flanging of Head

There is only one way to avoid this trouble and that is to flange the head to a good-sized radius and just far enough to make the flanged part parallel with the shell, as shown in Fig. 3. To do this, I prefer turning the flange on a flanging machine that is designed to roll the flange. Some engineers specify angle braces or through rods, but either one is objectionable and absolutely unnecessary if the head is thick enough and properly flanged. Double riveting is also sometimes specified when it is not required and only adds expense.

The radius of flange knuckles is usually made less than that shown, in which case the liability of fracture is greater when the heads are not properly flanged.

John J. Hoppes.

Springfield, O.

Hydrostatic Boiler Test

Some time ago I saw a boiler tested that had been allowed to freeze up during a cold snap. The engineer had no force pump or apparatus of any kind, yet he tested the boiler to the desired pressure, 100 pounds.

The boiler was filled with water, leaving no air space whatever, and then a small fire was built under the boiler, and as the cold water warmed up and expanded, the pressure gradually increased until the gage registered the desired testing pressure.

Then the try cocks were opened, allowing a small quantity of water to escape and relieve the pressure. Having found the boiler in good condition, the water was drained from the boiler to a proper level and the engineer then proceeded to get up steam.

H. D. Chapman.

Washington, D. C.

[While the foregoing method of conducting a hydrostatic test may be used as stated, extreme care should be taken not to build too hot a fire, and above all to keep a careful watch on the gage so that an excessive pressure will not be put on the boiler.—Editor.]

How to Make a Flash Boiler and Magneto

I would like to ask if any reader can give me the necessary directions for making a flash boiler for a 3x4½-inch engine to be run at a speed of 200 revolutions per minute.

I would also like directions for making a magneto for gas-engine ignition.

C. E. Anderson.

Minneapolis, Minn.

Correcting a Packing Trouble

Packing waste is often due to too much open space around the piston rod in the gland and cylinder head. When this condition exists, fibrous packing will invariably disintegrate and be drawn into these spaces to be blown out onto the crosshead and guide or into the cylinder, eventually finding its way through the exhaust passages to the heater and sometimes causing trouble by clogging the exhaust drains. Prolific waste from this cause has undoubtedly gone on for years in some plants, the real cause remaining unknown, while many different kinds of expensive packing have been experimented with to no avail.

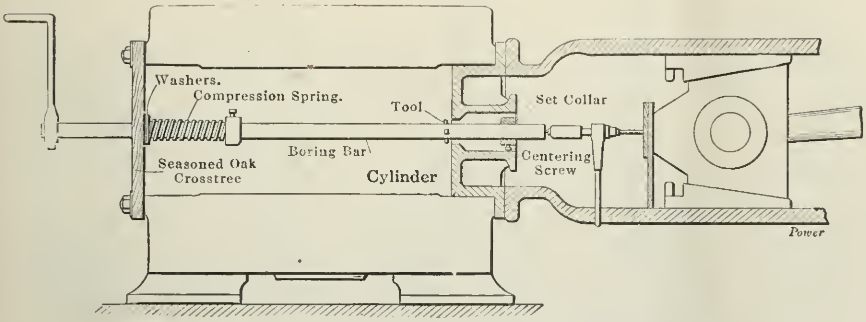


Fig. 1. Boring Tool in Place

A certain Corliss engine having a 4¼-inch piston rod had been mishandled for a long time, no care having been taken to set the piston central with the cylinder, with the result that the holes in the head and gland were worn so badly that there was nearly ⅛-inch space all around the rod. When freshly packed, this stuffing box would sometimes begin to blow within two days, and by the end of a week would need several new rings of packing. This condition was partly overcome by using a very hard packing ring at each end and soft packing in the middle until an opportunity came to make permanent repairs.

A piece of shafting, drilled and tapped for a tool and setscrew, as shown in Fig. 1, was utilized for a boring bar, and a set collar for centering the boring bar in the stuffing box was drilled for three setscrews. Another set collar, a stiff steel compression spring and two large washers completed the boring bar. A seasoned oak strap was bolted across the center of the cylinder and bored to receive the boring bar at the head end. The boring bar was then centered to the counterbore at both ends of the cylinder by means of pointed pine-wood trams about ⅜ inch in diameter and having a pin driven into each end for lengthwise adjustment. The boring bar was then pushed forward so that the disengaged tool was at the extreme end of its outward travel. The crosshead was moved up to a convenient position and a ratchet drill inserted between it and the boring bar. The set collar and spring were then shoved up to the washers against the oak crosstree, as shown; grease was placed between the two washers and the setscrew tightened. The ratchet feed was then unscrewed until the tool was at the beginning of its travel and the tool was set for the first cut.

The hole was trued and enlarged to the required size and countersunk at the cylinder end and a brass thimble, shown at *A*, Fig. 2, was riveted into place.

The stuffing box has given no further trouble and has now been running for months without the addition of new packing.

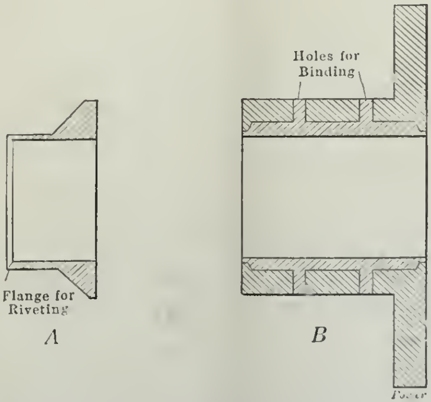


Fig. 2. Brass Thimble and Babbitt-lined Gland

The gland was sent to the shop with the intention of having a brass bushing inserted but no suitable casting for the purpose was at hand; and, as it would have required some time to secure one, it was decided to bush the gland with babbitt. This was done as shown at *B*, Fig. 2. This has run successfully for an entire season, and in one instance, where an unskilled attendant tightened the gland to a cramped position, no harm was done to the rod, whereas if a harder metal lining had been used the rod would have been heated and perhaps scored.

F. C. Holly.

Yazoo City, Miss.

Automatic Nonreturn Valves

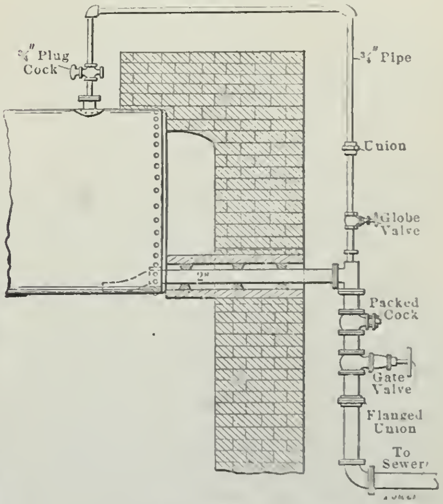
I would like to hear from engineers as to their experience with automatic nonreturn valves. Are they as successful as manufacturers claim them to be, and will they chatter and hammer so as to become useless? Will the engine or the vibration of the header cause the valve to hammer, and, if so, is there any way of stopping the trouble?

Louis J. Corilla.

Ironwood, Mich.

Position of Blowoff Pipe

The accompanying sketch shows the way in which the blowoff is connected to my boiler, a method which has given no trouble for the past ten years, except for renewing the disks on the valves about once a year. The pipe enters the boiler through the back head with a long thread to which, inside the boiler, is connected an offset pipe and coupling. The pipe is protected by a cast-iron sleeve one inch thick, having lugs cast inside to center the pipe and provide an air space around it. The opening through the brickwork is considerably larger than the sleeve and the intervening space is packed with loose asbestos. Outside the brickwork the pipe is connected to a reducing tee the small



Blowoff Connection

end of which is connected to the steam dome of the boiler; this keeps up the circulation in the pipe. The lower end is fitted with a packed cock and a gate valve and leads to the sewer.

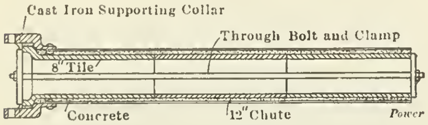
EDWARD HAMILTON.

Ridgefield Park, N. J.

Lining Coal Chutes with Tile

At the plant in which I am employed we had a number of 12-inch coal chutes which had been patched with sheet metal until there was hardly enough left of some to hold them together. We then decided to line our chutes with virtified tile, as shown in the accompanying sketch, and have found them to be almost everlasting.

The tile is assembled and clamped with the bolt, then wrapped with expanded metal for the whole length. The tile is then inserted in the chute, and is centered and concreted while held in a vertical position. After the concrete has set the bolts and clamps are removed.



COAL CHUTE LINED WITH TILE

Where the chutes have been badly eaten away ½-inch iron rods may be used to reinforce the tile, and the outside may be given a good appearance by grouting over with cement. If the tile lining were used on new chutes, only a light steel shell would be necessary; hence the first cost would not be increased very much. For hard coal an 8-inch inside diameter is sufficient for most any requirements.

H. B. SCHAMBERG.

Scranton, Penn.

A Homemade Friction Drive

The accompanying illustration shows a double-friction drive which I have tried and found fairly successful. The driven shafting runs in synchronism with the engine shafting two-thirds of the time, and there is an emergency clutch for quick action.

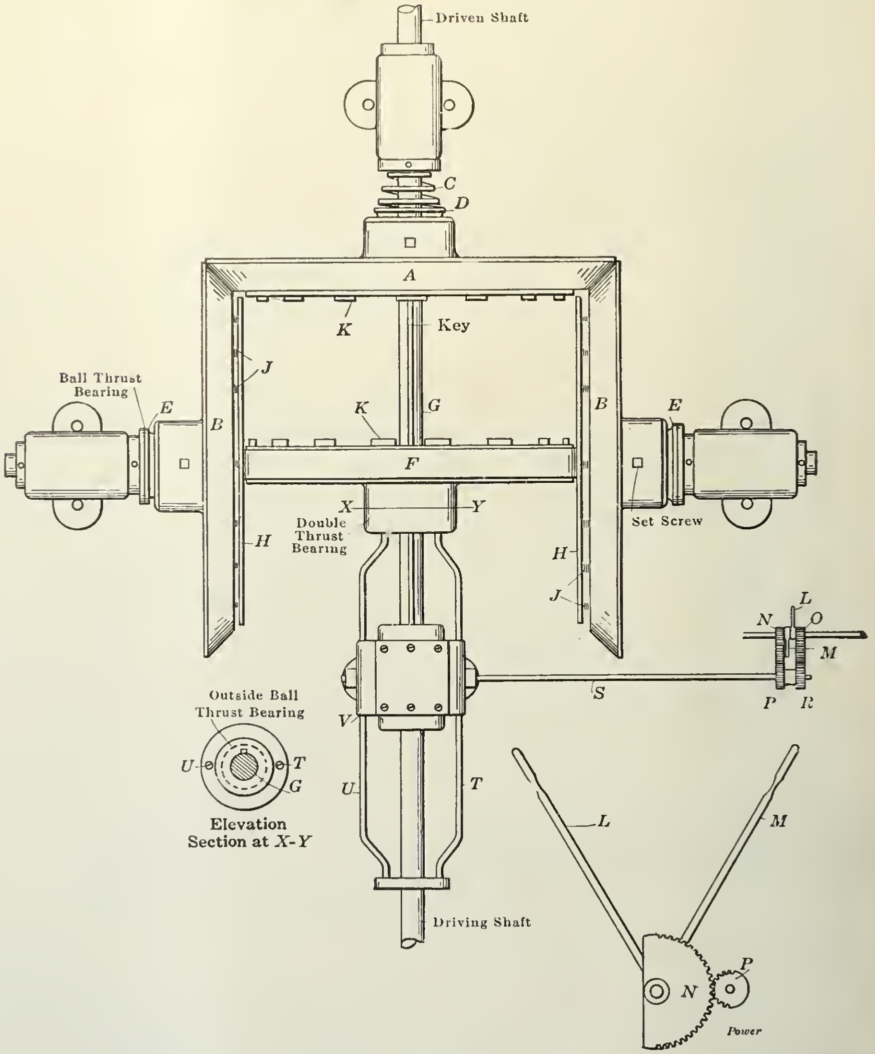
The main driving pulley *A* is feather-keyed to the engine shaft, and is set to bear against the pulleys *B B* by the spring *C*, the tension of which is regulated by the threaded steel bushing *D*. The pulleys *B B* are also feather-keyed to their respective shafts, but are set out by backing the threaded steel bushings *E E* against a thrust bearing. The pulley *F* travels on the shaft *G*, and is revolved by contact with the aluminum-faced friction disks *H H*. These disks are set out by the short, stiff independent springs *J*, which are wound around steel driving pins. This method precludes the tendency of the bearing pressures of the disks *H* to throw the pulleys *B* from contact with the pulley *A*.

As the pulley *F* slides toward the pulley *A*, the speed of the driven shafting increases, so that when they are within an inch or so of each other, the difference in speed is very slight. If, therefore, the lugs *K* on the pulleys be brought together, the driving and the driven shafts will be coupled together, and the contact will be made without danger of shock, and the pulleys *B B* and *F* are relieved of all wearing strains on their frictional surfaces.

The shaft *G* has a hardened-steel roller bearing contained in the wood filler of pulley *A*.

The method of control is as follows: The two controlling levers *L* and *M*, one for the forward, the other for the reverse, operate the semi-gears *N* and *O*, which in turn revolve the small gears *P* and *R* and the shaft *S*. The side rods *T* and *U* have teeth cut on their lower faces which mesh with small gears keyed to the shaft *S*. The casting *V* is bolted to the journal of the shaft *G*, and forms a bearing for the side rods and the shaft *S*. Each of the controlling levers *L* and *M* actuates its respective semi-gears by means of two pawls, one for the forward movement and one for the reverse. When at their extreme throw, the levers have a slight lost motion, which allows a small cam to engage with one of the pawls on the other lever. Thus when one lever is engaged the other is inoperative.

I am aware that the design of the friction drive could be improved, but, as it is homemade, the patterns had to be very simple, such as I could turn out myself, and one pattern had to do for the four main bearings, and one for the four friction pulleys.



DETAILS OF FRICTION DRIVE

R. O. RICHARDS.

Framingham, Mass.

Pipe covering which has become unsightly and soiled can be made to look as good as new by first removing all loose dust and giving it a coat of whitewash in which enough salt has been dissolved so that it will not rub off.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Uncle Pegleg on Velocity

Uncle Pegleg's lectures upon the philosophy of mechanics are always interesting and instructive, as well as amusing. I have just run across one of his points, however, which has set me to figuring. It was published in the August 23 issue.

In this lecture, Uncle Pegleg was endeavoring to fix in mind the fact that it is only *change* in velocity, not velocity itself, which absorbs force and energy—which is true enough. To bring this home he called attention to the contrast in the work apparently to be performed by two baseball pitchers, one using the North Pole as his box and the other standing on the equator and pitching east. The latter would be sending balls through space at a velocity greater than that of rotation of the earth's surface—which would be a pretty hot ball for a batter to find.

But, Uncle Pegleg pointed out, the pitcher at the equator does not send the ball out of the box, nor over the bag, at this enormous velocity. The only velocity in which pitcher, batter and catcher are concerned is that of the ball *relative* to that of the earth's surface.

So far very good. But here comes the puzzle. The energy required to alter the velocity of a body, such as a baseball, is not proportional to the difference between original and later velocity only, but to the difference between the original and later velocities *squared.*

Thus, proceeding to figures, the mass of a baseball, or its weight divided by the acceleration of gravity, 32.16, is just about 0.01. If we assume the "muzzle velocity" of the ball to be 60 feet per second; that is, the speed which the pitcher imparts to the ball, the usual formula for energy absorbed is,

$$\tfrac{1}{2}\,M\,V^2 = \tfrac{1}{2} \times 0.01 \times 3600 = 18$$

foot-pounds of energy.

From this we can figure the power developed by the pitcher, by assuming that his hand makes a stroke of 6 feet in starting the ball into motion. If its motion increases steadily the *average* velocity of the ball, while it is in his hand, will be

$$\tfrac{1}{2} \times 60 = 30 \textit{ feet per second,}$$

and the time taken in starting the ball will be

$$6 \div 30 = \tfrac{1}{5} \textit{ second.}$$

Hence, the power developed equals

$$18 \div \tfrac{1}{5} \div 550 = 0.1635 \textit{ horsepower.}$$

This is only the power absorbed in starting the ball. As the pitcher also has to start into motion his hand, arm and the rest of his body, the real power must be several times this figure—probably about one horsepower.

But, the above figures apply only to the pitcher at the North Pole. How about the one at the equator, facing east? He has to impart to a ball already possessing the velocity of the earth's surface, or about 1500 feet per second, an *additional* velocity of 60 feet per second, bringing its final velocity up to 1560 feet per second. How much energy will this take, and what power must the pitcher develop?

Using the same formula as before, the energy equals

$$\tfrac{1}{2}\,M\,V^2 - \tfrac{1}{2}\,M\,U^2 = \tfrac{1}{2}\,M\,(V^2 - U^2)$$

wherein, $U = 1500$ and $V = 1560$ feet per second respectively. According to this, the energy equals

$$\tfrac{1}{2} \times 0.01 \times (2{,}433{,}600 - 2{,}250{,}000) = \tfrac{1}{2} \times 0.01 \times 183{,}600 = 918$$

foot-pounds.

But this is 51 times as large as before!

The power developed by the pitcher must have increased in like proportion, so that the power going into the ball is now 8.34 horsepower, while that developed by the man must be about fifty horsepower! They must use stalwart baseball players on the Orinoco Giants! Eleven innings of exertion at the rate of fifty horsepower would try a man.

This does not look quite right, Uncle Pegleg, but it is figured according to "Hoyle." What is the explanation?

Sidney A. Reeve.

New York City.

The explanation is that you charge the pitcher with the velocity which the ball already has but do not credit him with the velocity which *he* already has.

The question is, how much energy is required to get the ball into motion at a certain velocity *relative to that of the pitcher* or to that of the earth on which he stands.

To get the ball into motion from rest at a velocity of 60 feet per second would take, as you say,

$$\frac{M\,V^2}{2} \text{ or } \frac{0.01 \times 60^2}{2} = 18 \textit{ foot-pounds}$$

You assume that the "stroke" is 6 feet, in which case the average force which he exerts on the ball must be

$$\frac{18}{6} = 3 \textit{ pounds}$$

To get the ball into motion at the velocity of the earth's surface at the equator, say, 1500 feet per second would require

$$\frac{M\,V^2}{2} = \frac{0.01 \times 1500^2}{2}\ 11{,}250 \textit{ foot-pounds}$$

and at 1560 feet per second

$$\frac{0.01 \times 1560^2}{2} = 12{,}168 \textit{ foot-pounds}$$

The difference is

$$12{,}168 - 11{,}250 = 918 \textit{ foot-pounds,}$$

or 900 foot-pounds more than when the motion of the earth was not considered.

The 18 foot-pounds was developed by the movement of the pitcher's arm through 6 feet relative to the earth's surface, with an average pressure of three pounds. This action required, as you figure,

$$\frac{6}{60 \times \frac{1}{2}} = \tfrac{1}{5} \textit{ second}$$

In one-fifth of a second, the earth and the pitcher, *and his hand, bearing on the ball with a force of 3 pounds,* will move through

$$\frac{1500}{5} = 300 \textit{ feet}$$

and this pressure exerted through this distance is the additional 900 foot-pounds which you introduce when you refer the movement to a point with reference to which the earth's surface is moving at a velocity of 1500 feet per second instead of to the surface of the earth itself or to the pitcher standing upon it.

Uncle Pegleg.

Central Station vs. Isolated Plant

I noticed in a recent issue an account of a meeting for the purpose of discussing the encroachment of the central station upon the field of the isolated or individually owned power plant. The speakers dealt in statistics, the results of carefully conducted experiments tending to prove that the isolated plant could furnish power cheaper than the central station. There is where the owner of the individually owned plant has the engineer at his mercy (although he may not yet know his power to hold him to the mark), for he must make good and in order to resist the encroachments of the central-power plant and hold his job. Although one or more experiments may not "prove all things," yet it is enough to show the mark which every engineer must strive to attain in order to hold not only his position but the standard of living he has set for himself. As I cannot intelligently or otherwise dispute the statistics submitted, the individually owned plant seems to have the vantage ground. I look upon the encroachment of the central station as irresistible, it is the usual development—or centralization, but, of course, it is of vital importance to the large body of engineers to keep this encroachment back as long as possible, for it means for them a subordinate position and a lower wage scale, and consequently a lower standard of living. In my opinion the large central station of the future will be operated by a small staff of experts in their line, men who can demonstrate both scientifically and practically the highest specialization of the art of engineering and who will have under them men who are now masters (but not owners of the plants they operate)—therefore it is, as I stated before, up to the engineering vocation as a whole to fight for its preservation as long as it can, and by giving to the owners of isolated plants a maximum of power for a minimum of outlay demonstrate that it is the better investment.

But the question that puzzles me is how long will this, the very foundation of all other industries, remain without the pale of centralization.

H. A. D'ORSAY.

Lynn, Mass.

Reverse Drive

In the issue of November 1, I noticed on page 1949 that S. McRae recommends a reverse drive to take the place of a cross-belt drive. The way in which he has it laid out will reverse the drive and will, as he says, be a straight running belt, provided the pulleys are in line. It seems to me, however, that he does not realize that the first pull comes on the idler, which makes it hard on the bearings, creating friction and decreasing power.

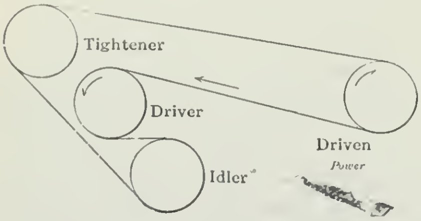


DIAGRAM OF CORRECT REVERSE DRIVE

A belt should never pull on the idler if there is any way to get out of it, as such an arrangement will always give trouble and require 50 per cent. more oil to keep it in running condition.

The accompanying figure shows a correct layout for a reverse drive. The pull is direct from the driver to the driven. The slack side of the belt runs to the idler and then to the tightener. This arrangement is light on the bearings and does not create half as much friction as Mr. McRae's arrangement. Also, the belt will last much longer and the drive will give less trouble.

With my arrangement the contact area on the driven pulley is even more ample than that on the driver. I have used a drive of this design for eight years and have never had any serious trouble with it.

T. J. MCNULTY.

Brookhaven, Miss.

Water in the Ashpits

In the October 18 number, Henry Perlman asks the opinions of other readers about the practice of having water in the ashpits. On several sugar plantations where I acted as assistant engineer, we had trouble with clinkers sticking to the grate bars, on account of dirt in the coal, and we had to pull the fires every three or four hours. I suggested to the chief that water be put in the ashpits, as I had done the same thing in the American navy and got good results. He had the same objections to offer as did the superintendent Mr. Perlman speaks of; but, at last, he told me to go ahead.

The practice was kept up for the subsequent two years that I remained there and for three or four years after that, until oil firing was adopted. The original bricks are still in the settings. So much for the softening effect on the brick.

Now, as to the effect on the clinkers. When I was in the navy, on a ship doing service in Hawaiian waters, we burned coal which came from Sydney. It was the dirtiest coal I ever handled. We had to blow the tubes out every half hour and break up the clinkers about five times in a watch of four hours.

I went to the chief engineer and asked him to let me try putting water in the ashpits, as I had heard that then the clinkers would not form so quickly, neither would they stick so hard to the grate bars. He told me to try it because, as it was, the fire-room force could not keep up steam. It took two men with a strong slice bar to break loose some of the clinkers before we tried the water in the ashpits. After that, by running the bar through the fire once an hour, we could keep going for a whole watch without cleaning fires.

W. ELLERBROCK.

Honolulu, Hawaii.

Writers among Engineers

I noticed the inquiry in the October 25 number, "Why Are There Not More Writers among Engineers?"

I think that one principal reason is that we simply do not sit down and go at it. It seems to be quite a task for some people to write letters. I know that I am in that class.

I am in charge of a plant in which there is one 18x42-inch Hamilton Corliss engine, two return-tubular boilers, one an 80- and one a 150-horsepower. This equipment furnishes part of the power for the factory. The balance is furnished by a three-cylinder vertical gas engine, operating on a four-stroke cycle with make-and-break ignition. This engine is belted to a 100-kilowatt generator. And, then, there are the usual auxiliaries necessary for both steam and gas prime movers.

Naturally, I am on the lookout for any information pertaining to such a combination of prime movers in general, and especially to gas engines, gas producers, etc. We are using natural gas now, but we may some time have to install a producer.

W. D. HOLLISTER.

Perry, N. Y.

Having read Mr. Gage's letter in the October 25 issue, I would like to say that I have often wondered why there are not more writers among engineers.

It is true that there must be men all over the country who could teach a good many of us quite a few things by relating little experiences encountered by them from time to time.

Strange to say, there are comparatively few engineers who show any desire to put their experiences on record, even when the subject is mentioned to them. It is not because they cannot, but because they will not talk upon the matter. Hence it is that the young man who is starting out as an oiler or fireman meets with so many obstacles.

It would seem from this that there is a certain feeling existing that cannot be called by any other name than jealousy. This is probably due to a feeling that the man who is seeking information is also seeking the other fellow's job. Allowing this to be so, is it not of equal benefit to both parties, for then the man who is being questioned should sit up and take notice before the other fellow takes his job.

Mr. Gage mentions the mixing of cement and asbestos in equal parts for pipe coverings, etc. It has been my experience that in cases of this kind where a leak develops, the water traverses the pipe for some distance from the place where the leak occurs before showing through, and when the covering is removed, the leak is found not at the place suspected but somewhere else.

In regard to the business end of a plant, I do not consider any plant is being run properly unless the engineer keeps a daily record of the work the plant is doing. It is his place to be able to tell his employer what is being done when the question is put to him. The method of keeping records can best be determined by a study of the plant and its needs.

Referring to Mr. Vedder's letter in the November 22 issue, which I read with interest, I want to say that I do not agree with him when he says that every man, including the first assistant engineer, should have a definite amount of bright work, floor area, etc., to clean. The duties of the first assistant engineer as I understand them are to keep the apparatus in the plant in first-class running order. Now, if in addition to this he is expected to clean bright work, floors, etc., I would not like to be in his position.

I do agree with him, however, when he says that the chief engineer should have no "pets," for if he does, the plant is bound to suffer.

H. H. Burley.

Brooklyn, N. Y.

Treatment of Subordinates

In regard to H. R. Rockwell's letter in November 15 issue, I think, as he does, that the engineers who find fault with their men simply to show their authority, lack good sense and principle.

As a sample of this class, I recall a man who was operating engineer on the shift on which I was fireman. The first night we worked together he took special pains to inform me that he was the boss and cautioned me not to forget it; this, without having even given me an opportunity to become acquainted with him. One glance was sufficient for me to notice his size and nationality, and convince me that I had better not dispute his claim. And I did not until one night after I had reported to him that one of the boilers had developed a leak under a rivet head. He came downstairs with a 12-pound sledge and a calking tool to calk the leak with 125 pounds steam pressure in the boiler. For a few minutes I forgot that he was the boss, and it was only after I had threatened to telephone to the chief that he agreed not to attempt to do the job.

Unless this man acquires more common sense and better judgment before he becomes "the boss" in reality, he will not be a safe man to work under.

To illustrate the right kind of engineer, here is an instance which came to my notice recently.

A young man, ambitious to get experience in another branch of the profession, had an opportunity to go to another plant and was about to take leave of the engineer under whom he had served as fireman for about six months.

The young man, taking the hand of his superior, said, "Well, Mr. ———, you are the best engineer I have ever worked for and I shall consider myself fortunate if ever I have a chance to work for you again."

The engineer replied, "You have been a good man for me, you have always done your work to the best of your ability and, while I am glad to see you advancing, I am sorry to lose you. If you need a recommendation, I shall be pleased to say a good word for you. I want you to keep in touch with me and, if ever I have a vacancy which you can fill, I shall certainly give you a chance." And, having said good-bye and exchanged best wishes, they parted. To the man who sees everything from the viewpoint of dollars and cents, this may seem like foolish sentiment, but it is not; it is a feeling of good fellowship which exists between men who have used each other right, and is never the result of tyranny or mean treatment.

During my experience I have worked for two chief engineers who were considerate of the men under them and had the courage to let that fact become known even to their employers, but I have worked for enough of the other kind to enable me to fully appreciate the right sort when I meet one.

J. A. Levy.

Greenfield, Mass.

I read with interest the letters of Mr. Westerfield and Mr. Rockwell, in the November 15 issue of Power, on the treatment of help.

I always try to treat my help in the same way that I would expect to be treated if I were in their places.

I never believe that a "grouch" gains anything, for the majority of men will do good work if you give them credit for what they do properly and not scold them if they make an occasional mistake. We all make mistakes. It happens sometimes, though, that men cannot stand humane treatment and get to thinking that their superior is not of much importance if he is not mean to them. The consequence is that they get to doing as they please. If you give such men an inch they will take a foot. If you try to correct them in their mistakes, they become offended and "fly off the handle" right away. Therefore, as Mr. Westerfield says, the best thing to do with men of that kind is to get rid of them, for there are plenty of men who can be induced to do what is right.

I have a man with me now, who put his first fire in a furnace for me a little over two years ago. He is now second engineer and giving good service. The first day he was with me he said that he did not know anything about a steam plant and that I would have to "show" him. I told him that he was just the man I wanted and if he would listen to me and do as I told him we would have no trouble. He did, and is still of the same attitude that he was on the first day.

He looks to me for advice and expects me to tell him, and when I tell him he listens and tries to do as he is told. He never acquired a "swelled head" nor got to thinking that he knew it all. We have never had a misunderstanding.

Walter Carr.

Harrisburg, Ill.

Hotel Power Costs

Having read the editorial in the November 8 issue relating to power cost in hotels, I beg to submit some figures on the cost of power at the Hotel Rochester.

The plant is two and one-half years old and has been under my charge since it was installed. The equipment is as follows:

Three 207-horsepower water-tube boilers. Three high-speed automatic engines, one of which is 11x12; one, 16x14 and one, 18x16 inches in size. To these are connected a 50-, a 100- and a 150-kilowatt generator respectively. One 14, 24 and 12 by 20-inch and one 16 and 10 by 16-inch elevator pump. Two 14 and 8¼ by 14-inch house-service pumps and two 7½ and 5 by 6-inch boiler-feed pumps. A 9x15-inch refrigerating machine driven by a 13½x12-inch engine.

The peak load on the electrical machinery is about 1400 amperes.

COST OF POWER FOR 1909.

Packing, water, repairs, sundries, etc. . $	918.02
Oil and waste.	277.88
Salary and labor, including wages of "repair man for the rooms" and man to handle ice in ice plant.	6,110.95
Coal at $2.60 ton.	9,706.80
10 per cent. interest and depreciation on power plant.	2,803.00
Total.	$19,816 65
Less heat, light and ice sold outside. .	3,379.45
Net cost.	$16,437.20

Output of dynamos for year, 514,228 kilowatt-hours.

$$\frac{16{,}437.20}{514{,}228} = 3.2 \textit{ cents per kilowatt-hour}$$

These figures include the cost of steam and repairs for the kitchen and laundry equipments, elevators and elevator pumping plant, ice plants, vacuum-cleaning apparatus, ventilating and heating equipment and hot water for the entire house. They do not, however, include the cost of lamp renewals, as this is charged directly against the rooms.

The following is the power cost for October, 1910, which is the best month we have had:

POWER COST FOR OCTOBER, 1910

Packing, oil, waste, water and incidentals, which item includes all cost of material except coal. $	55.00
Coal at $2.60 ton.	947.88
Labor.	550.00
10 per cent. interest and depreciation.	231.00
Total.	$ 1,786 88
Less amount sold (heat, light and ice)...	562 23
Net.	$ 1,224 65

This cost includes, as in the previous case, the cost of steam and repairs for the kitchen and laundry equipment, elevators, ice plant, etc.

FIRST COST OF POWER PLANT EQUIPMENT.

Engines and dynamos complete.	$13,600.00
Boilers, stokers, stack and feed pumps complete.	14,430.00
Heating and ventilating complete with pipe covering connections to and including work in kitchen and laundry—everything complete	22,970.00
Ice plant complete.	12,500 00
Elevators and elevator pumping plant.	29,250 00
Electric wiring and fixtures.	17,620 00
	$110,370.00

Of this amount the only sum for interest and depreciation that could be charged against the lighting outfit would be that on the cost of the engines, dynamos and boilers complete, or $28,030.

William J. Bedard.

Rochester, N. Y.

An Engineer's View of "Graft"

In the November 22 issue appeared an article entitled, "The Perquisities of an Engineer," having for its object the exploitation of a recently formed organization of engineers, the purpose of which appears to be the segregation of engineers into two classes—the "blacklist," composed of those who accept what is termed a bribe, and the "honor roll," composed, presumably, of members who are to be listed as immune.

The article further states that the organization does not aim to interfere with or injure other organizations but has for its principle the abolition of "graft."

It would seem that had the founders of this organization had the redemption of the engineer from the dangers of grafting solely at heart, some established association could have been found whose aims and social status would be sufficiently high for its members to consent to incorporate into its teachings the tenets and principles of this new order, thereby enjoying whatever benefits might accrue.

The subject of "graft" has been pretty well threshed out by correspondents in Power and it is generally conceded to be a vicious habit. It will be of interest to learn just how the new organization interprets the word. If accepting cigars or the usual tokens that are given in return for honest patronage are included in the category, I predict that the list of survivors on the "roll of honor" will be short.

The owner of a local factory during a heart to heart talk with a newly hired engineer expressed a somewhat original view of the subject. He said: "When buying supplies I expect you to use your best judgment as to what supplies are necessary, the quality, price and quantity. I want you to keep on hand everything in reason that pertains to the satisfactory operation of the plant in such quantities as will be economical. Don't buy a large quantity of something that will dry up or spoil because a large order costs a little less. And don't let the salesmen 'get a rope on you.' "

The engineer hastily assured him that he was not in the habit of accepting "graft" from salesmen.

"You needn't call it graft," the employer resumed. "If you order right and see that you get what you order you need not be afraid to accept any present the salesman may offer. The price to me is just the same and I keep as close a watch on this department as I do on the rest, so if the orders are not right or if the plant is not kept up, you will surely hear from me. If you feel any compunctions regarding what you call graft, you can accept it and turn it in at the office. That will be better than leaving it in the hands of the salesmen."

In justice to the engineer, I will say that up to date no returns have been made to the office, and the best of feeling prevails. The engineer and employer invariably consult as to what and where to buy and the price to be paid.

The organization and the editorial referred to may have been inspired by an effort to uplift the benighted engineer, but it looks as though the sales people and not the engineer or the employer are the ones to benefit by a too critical view of what constitutes a bribe.

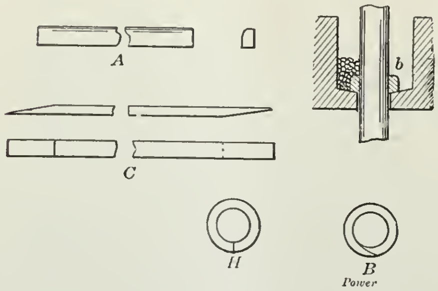
Amos Skeg.

Saugus, Mass.

Trouble with Metallic Packing

I read with interest Mr. Chapman's account in the November 1 issue of the trouble he has been having with metallic packing.

When an engine is shut down for any length of time, the piston should be taken out and the cylinder given a heavy coat of oil or grease. I suppose that this was done in this case. When the piston was taken out the copper wire was probably moved a little, thus giving the packing a chance to work out through the clearance between the rod and the back of the stuffing box. A piece of copper wire is not very satisfactory as a bushing as it will not follow up the wear of the rod and thus it leaves a space through which the packing may work out. A small copper bar with one corner rounded off, as shown at *A* in the figure, will work better. The pressure of the packing on the curved surface *b* of the ring will cause it to stay close to the piston and thus prevent any leakage of packing past the ring. The bar should be annealed and the ends filed down as shown at *C*, so that they will fit together as shown at *B*. The ends can then slip a little as the wear is taken up around the rod, while if the ends were left as shown at *H*, the ring would not follow up the wear of the rod.



Bushing for Metallic Packing

R. L. Rayburn.

Kansas City, Mo.

Furnace Design

In Power for November 22, D. G. B. asks for a rule for finding the proper distance between the grates and the shell of a horizontal tubular boiler. In the answer given it is stated that this distance should not be less than the diameter of the boiler. I think that the distance should be about equal to the radius, or half the diameter of the boiler, or, at most, not over three-fifths of the diameter of the boiler for hand-fired furnaces and bituminous coal. Even with a distance equal to only two-fifths of the diameter of the shell good results have been secured when the chimney draft was not less than three-tenths of an inch, water gage.

In the November 15 issue there is an article by D. F. Randall which was very interesting to me. Mr. Randall did not give his opinion as to whether the use of full arches from the boiler front to the back end of the shell is a detriment to the boilers. I know of many engineers who say that it is, based on the experience they have had with leaky tubes.

G. A. Gustafson.

Chicago, Ill.

Feeding Boilers

My article under the above heading, printed in Power for October 18, received a severe raking over by Mr. Willie in the November 22 issue.

Be Mr. Willie right or wrong, these facts cannot be disputed: This type of feed has been in use for two years on my boilers; no such trouble as Mr. Willie describes has happened so far. Furthermore, this feed was suggested by an inspector from a reputable insurance company which carries quite a few risks in that part of the country where Mr. Willie resides; namely, Massachusetts.

However, I would like to see this matter thrashed out. I do not want any cracked plates or burnt sheets. As Mr. Willie says, the danger exists and it is good practice and common sense to guard against it. As for the coolest part of the boiler, is it not at a point close to the uptake? Take the temperatures of the gases in the front part of the tubes and in the rear; a blind man could "see" the difference in temperature.

Any suggestions as to the merits of this method of feed will be welcomed. What has T. T. Parker, of New York, to say?

A. Rathman.

Chicago, Ill.

Annual Meeting Mechanical Engineers

Most successful meeting of recent years. Large attendance listens to a number of valuable papers. Col. E. D. Meier elected president.

The fall meeting of the American Society of Mechanical Engineers was held at the Engineering Societies building, New York City, December 6 to 9, and was one of the largest and most successful meetings of recent years. Increased importance was given to the opening session on Tuesday evening by holding the formal reception of the president and president-elect after the session at which the president's address was presented, a concert being furnished and a collation being served.

Retiring President's Address

The address of the retiring president, George Westinghouse, was of particular interest, in that it contained the story of his invention of the air brake. Having been delayed by a collision, his mind was directed to the important saving which could be effected in life, property and time by the application of effective mechanically operated brakes to the trains. He worked at first with chain brakes operated by a steam-actuated cylinder on the locomotive, but this was ineffective for a sufficient length of train. He then tried placing a cylinder beneath each car, furnishing the steam through a flexible coupling, but this failed because even in warm weather it was found impossible to transmit the required pressure to the rear end. He finally saw in a mechanical paper an account of the use of compressed air for operating the drills in tunnel construction, and this suggested the use of that medium for the brake. An account of various tests of the device, the organization of the Westinghouse Air Brake Company and the various developments which have followed the use of the brake followed.

Mr. Westinghouse also earnestly urged uniformity in the electrification of railways, pleading not for the adoption of any particular system, but for uniformity and interchangeability. With the growing use of electricity by the railways, this is a subject which will have as much importance in the early future as the adoption of standard gages and equipments did in the early days of railroading.

The secretary then announced the election of Col. E. D. Meier as president for the ensuing year. Following this introduction, the meeting, which was held in the auditorium, adjourned to the rooms of the society in the same building, where an opportunity was afforded to meet the retiring and newly elected president at the formal reception above referred to.

Officers Elected

The business session of the society was held on Wednesday morning. The tellers of election announced the election of the following officers for the succeeding terms: For president, E. D. Meier; for vice-presidents, H. H. Vaughan, E. M. Herr, George M. Brill; for managers, D. F. Crawford, E. B. Katte, Stanley G. Flagg, Jr.; for treasurer, William H. Wiley.

Wednesday Morning

Past-president W. R. Warner reported regarding the Land and Building fund and an amendment was adopted bringing the constitution into accord with the articles of incorporation. The subject of the importance of an engineering museum was brought up by Doctor Hutton and discussed, and Mr. Gilbreth suggested the importance of a department or committee of industrial management. President Meier commented favorably upon the address of the retiring president and a rising vote of thanks to Mr. Westinghouse was passed. President Meier also announced the death of William H. Bryan, of St. Louis, one of the active members of the society. Mr. Bryan had recently been appointed as engineer to the school board of Chicago, after a competitive examination, and the news of his sudden removal was universally deplored.

Prof. F. R. Hutton, honorary secretary of the society, and one of the official representatives of the society at the English meeting, presented a verbal account of the visit, and Calvin W. Rice, the secretary, presented, with a few supplementary and explanatory remarks, his printed account of the occasion. James M. Dodge presented a progress report for the committee on public relation.

The paper by George A. Orrok on the transmission of heat in surface condensation, abstracted in our issue of December 13, was then presented and discussed.

Transmission of Heat in Surface Condensation

Prof. W. F. M. Goss: It seems to me that in general there is no more prolific field for engineering research today than that touched on by this paper of Mr. Orrok's, that which deals with the effect on heat transference and of the philosophy of the fluids involved. The time is near at hand when we are going to very much enlarge our conceptions as to what may be done with a given area of heat-transmitting surface through controlling the velocities of flow over these surfaces. As a contribution touching this very important field, I look upon the paper of Mr. Orrok as one of very great value.

Professor Ennis: I should like to ask Mr. Orrok, first, as to how he rated the flow in comparison with the size of the apparatus compared with that in a modern large-surface condenser?

Mr. Orrok: I think we kept within the condenser limits—that is, I went as high as 10 or 11 feet a second, and kept down as low as one foot a second; that is, all of the experiments are within the condensing range. We did run one set of experiments at 4 or 5 inches vacuum, and succeeded in condensing 220 pounds of steam per square foot of surface, and I think that is double what has been done before, but so far as the conditions are concerned, they are within absolute practice.

Professor Ennis: I could wish that a little discussion had been put in the paper on the question of tube thickness, and two or three other practical questions. Take the matter of oxidization. Mr. Orrok lays some stress on oxidation as a determining factor, although Joule regarded oxidation as of light influence. The question is both technical and commercial at this point. We have oxidation on the one hand, and the transmission coefficient on the other hand, and on top of that a large number of purely commercial factors. The question of oil cutting in reciprocating-engine installations would be one of the factors presumably taken care of by the cleanliness coefficient *d*, but for which we have no data in the figures. What I would like would be to have Mr. Orrok say what kind of a tube he would use, for example, in his own practice here in New York with salt water for condensing, but he does not tell us quite that.

H. H. Suplee: Mr. Orrok has called attention to the important influence of the presence of air, and he has quoted also the paper of Mr. Morrison before the Institution of Naval Architects. Mr. Morrison, if I recollect, called attention to the fact that it was not only the presence of the air that exerted the influence, but its location. The tendency of air which leaks in is simply to crowd out the steam, and also the vacuum, but the influence of air contained in the water is to clinker the tube. Mr. Orrok speaks of a condenser consisting of a single tube, through which the cooling water is flowing, surrounded by an atmosphere of steam. Condensation, he continues, will take place at the outer surface of the tube and the steam will rush in to the tube from all sides. That is what Mr.

Morrison said, but that the air of the steam will rush to the tube, and the result is that the outside of the tube is immediately covered with bubbles of air, as we see bubbles of carbonic-acid gas clinging to the side of a siphon. That forms a nonconducting surface and cuts down the condensing value of the tube, and that does more harm than the air leaking through by degrees—more harm than anything else. Mr. Morrison said that the proper thing to do is to keep the tube swept clean of air, as we have scrapers on boilers to scrape the soot off. He thought if there was some moving back and forth through the length of the tube, brushing away the bubbles of air, the efficiency would be increased greatly. It is the location of air, and until you know about that your other determinations are almost certain to be irregular.

Thomas C. McBride: The writer is particularly interested in the author's experience as to the effect of air mixed with the steam, and the fact that this air, apparently from an unknown cause, varied in amount so greatly in the apparatus as at first constructed and connected that the results did not seem to be constant enough to have any scientific value. The conditions of these first tests were apparently, however, those which we meet in low-pressure turbine practice, and it seems therefore to the writer that they should have a practical value. It is hoped that the author will explain more fully the origin of the exhaust steam used in the apparatus as first constructed, and his conclusions as to what might have been expected from it as revealed by his tests, if used on low-pressure turbine work.

If, as the author states, the partial air pressure in his apparatus approached ¼ inch of mercury, the results obtained can hardly be considered as applying to practical conditions in surface condensers. In the writer's opinion, the heat-transfer results obtained indicate very much less air, this opinion being based on the heat-transfer results obtained and air condition existing in surface-condenser plants. Any condenser having as much as ¼-inch air pressure would require a vacuum pump of a size that would be prohibitive. As an example, if we presume the condition usually existing in surface condensers carrying 26-inch vacuum where it is presumed that the temperature of the water, steam and air passing to the vacuum pump is 110 degrees, one volume of air at its individual pressure of ¼ inch and temperature of 123 degrees, to which the steam temperature has been reduced by its pressure, will become after passing through the condenser at its temperature of 110 degrees and individual pressure of 1.4 inches:

$$\frac{0.25}{1.40} \times \frac{459.5 + 110}{459.5 + 123} = 0.174$$

Similarly this one volume of steam at its individual pressure of 3¾ inches before reaching the condenser has a volume 11,687 times that of the corresponding water.

The ratio of the effective displacement of the air pump and the boiler-feed pump required to handle this air will be

$$\frac{0.174}{\frac{1}{11,687}} = 2033$$

The usual practice of manufacturers of these conditions is an air-pump ratio of 13 to 1.

Similarly presuming ¼-inch air pressure in the exhaust steam with 28-inch vacuum in the steam-turbine work and that a separate cooler and rotative dry-vacuum pump are used, assuming also that the mixture of air and steam passes from the condenser to the rotative dry-vacuum pump at a temperature of 75 degrees, one volume of air as it exists in the exhaust pipe at its individual pressure of ¼ inch and temperature of 97 degrees will have, after leaving the condenser, an individual pressure of 1.13 inches and a temperature of 75 degrees and therefore become

$$\frac{0.25}{1.13} \times \frac{459.5 + 75}{459.5 + 97} = 0.212$$

volumes. The volume of steam in the exhaust pipe at its individual pressure of 1¾ inches will in its condensation be reduced in volume about 23,875 times, or the ratio of the effective displacement of the rotative dry-vacuum pump to the boiler-feed pump must be

$$\frac{212}{\frac{1}{23,875}} = 5060$$

This, again, is prohibitive, representing a vacuum-pump displacement 200 times or 300 times that usually found necessary in steam-turbine condenser work.

Accepting Mr. Smith's statement that the effective displacement of the rotative dry-vacuum pump must be about 20 times the displacement of the feed pump, we can calculate the amount of air presupposed by this statement to be mixed with the steam on its way to the condenser. Using the same assumptions as above for 28-inch vacuum, one volume of water in the feed pump becomes about 21,500 volumes on its way to the condenser; 20 volumes of air passing from the condenser to the rotative dry-vacuum pump at its temperature of 75 degrees and individual pressure of 1.13 inches would have, when mixed with the steam before condensation, a volume of 21,500 and temperature practically 101.4 degrees. Neglecting the slight increase in volume by change of temperature, the individual pressure of this air in the exhaust pipe must then have been

$$\frac{20}{21,500} \times 1.13 = 0.00105 \textit{ inch}$$

The relative volumes of the air and steam in the exhaust pipe if separated would therefore be as 0.00105 to 2, or a ratio of 0.00052. In other words, Mr. Smith's conclusions as to the rotative dry-vacuum pump capacity that should be used involve 5.2 volumes of air mixed with 10,000 volumes of steam in the pipe leading to the condenser and a considerably lower air pressure than he has stated when operating under the conditions assumed.

These questions suggest the advisability of the society adopting some sort of a standard code or standard method of testing large condensers, particularly as refers to the question of the amount of air present and the influence of this air on the results obtained, and it is believed by the writer that much useful work could be accomplished by the society in establishing some code.

Mr. Orrok: Professor Ennis speaks about the thickness of the tube—most of the tubes which we tested were of the same thickness, 18 gage; I had some 16 and some 20. I could not determine any difference between 16-, 18- and 20-gage tubes. In fact, my own idea is that I do not think I could have detected much difference between 12 and 20, and perhaps even larger. I think we never used a tube up to the limit of its capacity.

Regarding the oxidation problem, I suppose we tested fifteen or twenty tubes in various stages of observation. On page 2205 of the December 13 number of Power the two lower curves in Fig. 3 represent, one a new Admiralty tube, that is, the one that U equals 308, and the one below, at 224, an old Admiralty tube that has been oxidized. How much oxidized, I would not like to state. If you should plot every heat transmission experiment there, and then try to draw a mean through the center, you would probably get somewhere within 0.05 on either side of the one-half, that is, it would go from 0.45 to 0.55 for the exponent. Professor Ennis thinks my variation there is quite large. I acknowledge it is large, but it is more than we were able to get in any other way, and I hope some day to make it smaller yet, but at present do not see my way to do it.

Mr. McBride wishes to know where our exhaust steam came from. That is the exhaust steam from blower engines and such auxiliaries turned into a large main running the length of the station, and tubes taken from that into open heaters. The chances for air are large, and I have no doubt there was air. Why do I say ¼ inch at the main? That is an estimate. I tried thermometers and pressure gages and almost every other means at my disposal to tell how much air I had there, and, while I could be sure of it for one minute, I could not be certain of it for more than one minute. I found by reading Mr. Smith's paper that he had exactly the same condition, and worked six months to get the galvanized-iron tank tight, and after he soldered the edges and tarred them, he found the soldered tank would not hold tight and air would get in. In our case the boiler, so called, was made of a couple of 14-inch 200-pound steam-heater steel plates. I have no doubt air came through the pores of that steel at a greater or less degree at that vacuum. As to the pipe that I used for the condenser and the connections, I have no doubt that this leaked air under vacuum conditions. I had two 1-inch air lines running to a big dry-vacuum pump and a condenser, and I was pretty certain of the flow of steam through the pipe, so that there was a continual current parallel with the tube to sweep that air out of the condenser, and even then I think I had air. We tried to find whether there was any variation of temperature within that fifth temperature tube that was the shell to the condenser. We put some thermometers in it and raised them up and down and put them close to the tube and close to the edges, and put the three checks along the length, and we could not find any difference in temperature any larger than about one-tenth of a degree. Now, I have found very much greater differences on a big condenser, but I think in the tube space, where the steam is, you can pretty nearly always find practically the same temperature.

Wednesday Afternoon

The session of Wednesday afternoon was presided over by Vice-president R. C. Carpenter, the first paper presented being that upon combustion and boiler efficiency by Edward A. Uhling, abstracted in our issue of December 13.

Discussion on Uehling Paper

O. H. Bathgate thought that a proximate analysis of coal is insufficient. A separate determination of sulphur is necessary to associate the quality of coal with efficient steam generation.

He agreed with the author that for daily boiler-room practice when the maximum CO_2 is likely to be below 12 or 13 per cent., the omission of the remaining gas constituents is not serious, and for the average fireman the simplifying of such matters tends to greater efficiency.

He believed that a plant where the CO_2 is determined either manually or automatically, should also provide a means for determining the carbon in the ash. Gas and ash analyses are equally essential to a complete knowledge of combustion, for in actual experience, he had found firemen sacrificing coal to the advantage of a high CO_2 record. This indicates that gas analysis in itself is not necessarily a true index of boiler efficiency.

He looked upon the various groups of the tabulated results with as much suspicion as does the author himself, who states that correct gas analysis, indicating furnace and boiler efficiency, depend on the composition of fuel, stack temperature and temperature at which analyses are made, and yet none of these important factors appear to have been given consideration. For this reason, inconsistent results would be expected, these varying with the conditions under which the analyses were made. If fewer results had been used and these corrected to a basis of standard fuel, temperature, etc., more reliable conclusions could have been derived.

Boiler-room efficiency can be increased by using a reliable CO_2 recording apparatus, but his experience with three different types and investigations of as many more had led to the conclusion that there is at present no practical automatic CO_2 apparatus available for continuous boiler-room operation.

To develop the highest efficiency, he would recommend running a single boiler on continuous test under strictly operating conditions, noting the quantities of steam generated and coal consumed and recording all conditions entering into the operation. Continuous gas-sampling vessels, coal and water meters; steam, water and gas temperature recorders are instruments whose practicability has been proved. With one man looking after these instruments, numerous records covering any desired period of time may be obtained. The result would determine the conditions necessary to effect the highest furnace and boiler efficiency. A CO_2 indicator and recorder would then be useful as an index of comparative operations and would be of material value only so long as the quality of coal and other controlling factors remained essentially the same.

Prof. William D. Ennis, of the Brooklyn Polytechnic, said: I do not believe that we can overestimate the importance of this subject. It is a vital thing in the power plant today, beside which such a matter as the steam turbine is a mere detail—but from an operating standpoint it is largely a matter of air leakage. How wholly it is a matter of air leakage I have recently had opportunity to determine. We took one of our boilers (making an effort to change nothing else) in its ordinary operating condition, and that was not very bad, and carefully plugged up all the air holes we could find in it. The result was that we increased the percentage of CO_2 from 4 to 11, simply by reducing the air leakage alone, no change in the method of firing, and this I estimate corresponds with an increase in boiler efficiency represented by the ratio 57 to 85—not that we got 85 per cent., but that is the ratio.

There are reasons—some were mentioned by Doctor Uehling—for measuring the CO_2 rather than the oxygen. One is that at reasonably good combustion efficiency, the percentage of oxygen gas is difficult to measure. There is another reason—that stratification in the furnace, that always occurs, results in the simultaneous presence of oxygen and carbon monoxide, and you might get one or the other, and any deduction you would form from the presence of one would be an unsafe deduction; but if you get an average sample of CO_2 across the furnace, you get something that represents the facts. CO_2 measurements usually tell that something is wrong, but not what is wrong. It is best to determine this by the oxygen and carbon monoxide, and we have an opportunity to check, since, as shown, the sum of this will be a constant, and if we had a coal containing no moisture, and air containing no moisture, the constant would be equal to the percentage of oxygen in the air by volume.

I must dissent from the opinion which has just been expressed that there is a comparatively low best percentage of CO_2. That may be technically and theoretically true, but in the power plant it has been my unvarying experience that the percentage of CO_2 of maximum economy is the highest we can get—that we cannot reach the point where we go the other way. If I am wrong, I would like to be corrected, as it is a practical point, but that is the state of my religion on the subject. We do not need to be afraid of going too far, or getting too high a percentage of CO_2. I would pay a fireman a bonus based on getting a high percentage of CO_2.

Professor Kent: Do you carry it above 15?

Professor Ennis: I have not gotten up to 15. You must remember that when we get to these high percentages, the increase in CO_2 from 15 to 16 does not mean much. A jump from 4 to 5 per cent. means six and a half times as much as would a jump from 11 to 12 per cent.

We have been trying some devices to get the CO_2 without using expensive automatic apparatus. We find in these machines the pipe gave considerable trouble. If the coal contained sulphur, we found a deposit in the pipe line, probably iron sulphate. That has been remedied by using copper pipes.

One of the claims made for automatic machines is that they give a constant indication of what is going on, yet in one plant we measured the pipe line, a ¼-inch pipe, 120 feet to the nearest boiler, and 180 feet to the farthest boiler, and by an ingenious method we measured the time required for the gas to travel from the different boilers to the machines. We found it took from five and a half to six minutes. In another plant, a ¾-inch pipe was used, 130 feet to the boiler, with a different type of instrument, intermittent sampling, and it took from four to sixteen and three-quarter minutes for the gas to reach the machine, and seventeen minutes for it to get through. I do not call that a constant indication. The fireman said that when they had the best fires the machine showed the poorest CO_2.

We have not given up the notion of judging everything we need to know about boiler economy from the temperatures at the furnace and stack. Our present line of investigation is to take samples in a large container. First we used a glass bottle, and collected the gas over water. We got the average composition, and we found if we tested it at once there was 10 per cent. CO_2, but if we let it wait a couple of hours, we would get a smaller percentage of CO_2. We tried tin cans, shellac joints, etc., and we came to the conclusion that the water absorbs the CO_2 rapidly, and in looking up the information we could get from the builders of automatic machines, we found it was their practice to put salt in the water, so that it would not absorb the CO_2. We tried that, but we found that the water absorbed the CO_2 as much as ever. We made some experiments to see how rapid the absorption was. We found it very rapid at first, and then it slacked off somewhat, gave a sort of parabolic curve. The absorption was greatest if the initial percentage of CO_2 was high. By using a film of kerosene over the surface of the water we have cut down the rate of loss about one-third, and by making the analysis quickly after taking the sample, there is no loss.

Some of the automatic machines confine the sample in contact with water for twenty-four hours, and you can imagine how corrrect the determination of CO_2 is under those conditions. After working with two or three types of these machines, ever since they first appeared on the market, I have drawn a mental specification of what constitutes an ideal CO_2 recorder. It must be automatic; not too delicate; it must not be easy for the fireman to fake; must be continuous and not intermittent; must record and indicate its readings, and, I think, most important of all, it ought not to cost over 25 cents a boiler horsepower, and one ought to be put on every boiler in the United States.

Prof. William Kent: I am rather surprised that Mr. Uehling did not in his paper mention the improvement in boiler economy that can be obtained by autographic registration of the furnace temperature. Fifteen years ago I borrowed one of Mr. Uehling's instruments for that purpose, to determine continuously the furnace temperature. With the Uehling pyrometer and the Bristol recording gage attached to it, I could see at all times what the temperature was, as close as the individual recorder would give it, and that was pretty close. Whenever the pyrometer showed in the neighborhood of 3000 degrees in the furnace we had good economy, and when it went down we had rather poor economy.

There are some other principles of boiler economy which Mr. Uehling has not emphasized, and that is, if you want the best economy you must have dry coal. You must not have coal which will clinker and change the condition of the fire. This is rather more important than any theoretical efficiency regarding gas analysis or coal analysis.

Another thing I notice in the recent literature on the subject of boiler economy is that too much emphasis is placed on heat units, especially with anthracite coal and semi-bituminous coal. The semi-bituminous coals are usually uniform in quality—that is, they have between 15,000 and 16,000 B.t.u. per pound of combustible, and if you take 15,758 B.t.u. you are more nearly correct than if you send the coal to a chemist to have the B.t.u. determined. With the anthracite coal the heat value per pound of anthracite does not vary over 2 per cent. from the anthracite with 7 per cent. volatile matter down to 2 or 3 per cent. volatile matter, and it is not necessary to analyze, to get the approximate value of anthracite coal, except to determine the characteristics of the coal as regards ash. The salient features are how much moisture is there, and how much ash?

When you go beyond the semi-bituminous coals, the case is different. With the bituminous coals of Pittsburg and the far West the situation is different, and it is necessary to make an analysis of the volatile matter before we can make any general conclusions. The most important things, for a proper boiler test, are the continuous autographic record of the temperature of the furnace and a continuous autographic record of the CO_2 in the gases. After we have these, we have about all the indexes we need for boiler economy.

Prof. D. S. Jacobus: We have, as has been pointed out, more to lose and gain in the boiler plant of the power house of today than in any other place, and this idea of recording the analysis of the gases is all a move in the right direction. It should be remembered, however, that the gas analysis is not the only thing, and I will say a little bit more on that later on, but before leaving the gas-analysis part, I want to say several things. We have been talking and Mr. Uehling has shown that the CO_2 reading is the important item, yet we have found out, where we have to reach the ultimate in getting efficiency out of a boiler, in a test, that the hardest thing to do is to dodge the CO, that is, when we can readily get the CO_2 with a good fireman or stoker up to 12.5 to 14 per cent., with only a trace of CO, it is entirely a different proposition to get higher percentages without making CO. We did not appreciate that fact so much, until we were driven against it in this way. We started in by making ultimate analyses of the flue gases, where we give the hydrogen and the hydrocarbons, and we found we have been thinking all along of determining the CO_2; and where we record the CO_2, very often we might have 0.2, or we might have had 0.4, or sometimes as high as 0.75 of a per cent. of CO, and that 0.75 of a per cent. means a great deal in the economy.

At the present time in determining the CO_2 we use two absorption pipettes, running first through one, which is used continuously, and taking out what is left in the second—and we mix a fresh solution on the ground before we start to work, and make sure that the solutions are right.

Now, to consider some of the other places where you can lose and gain in boiler-room practice. There is as much loss in the average boiler room by running along with the boilers at a very low load during certain parts of the time as there is through excess air. There are two problems in the generation of power, one in the plant which is running continuously at a certain uniform load; the other, the ordinary power plant, where, we will say, you have a load running over a period of five hours, where the load starts up and goes to a peak early in the day, going down again, and then you will have another heavy peak, and you do not know when it is coming, and you must have enough boilers to carry you over the peak. The great problem, and where a good man will save expense as compared with another man, is in getting the boilers ready to meet these peak loads, and in that one particular as much can be lost and gained as in cutting down the air, or getting up the efficiency in the boilers while they are actually running. The modern practice tends more and more to run the boilers hard during the peak-load periods, because if you can carry over two or three hours of the peak, even at the sacrifice of some little efficiency, and thereby cut off one-third of the boilers in the plant, the efficiency of the plant, that is, the daily or weekly or monthly efficiency, will be better in running the boilers in that way; and at present the modern practice tends to run the boilers during peak loads at double the so called rate of 10 square feet of grate area per horsepower, and putting in boilers that will come up to that class of service, and the whole tendency is to drive the boilers even harder during that period, and I would not be surprised to see them driven considerably harder than that during these peak-load periods, with a gain in the efficiency of the plant as a whole. That does not apply to a plant where you have a nearly steady load, like a cotton mill or some load of that character. In that case we have a different proposition, and as we go deeper into the matter we appreciate the fact that a certain boiler arrangement that will apply in one case will not fit into another, and you have to study the conditions of the particular plant from which you desire to get the best results.

H. C. Abell: There is one other point not brought out forcibly enough, and that is the hydrocarbons not being properly burned. Fifty per cent. of free air will give you about 240 cubic feet of gaseous products of combustion. Assuming the 15,000 B.t.u. per pound Professor Kent was speaking of, and assuming that only 1½ cubic feet of marsh gas is given off, which gives less than 0.7 of 1 per cent., we have 10 per cent. less in thermal efficiency—1500 B.t.u. escape in the 1½ cubic feet of marsh gas. In some of the boiler efficiencies I have followed up I have made tests by following the Orsat apparatus with an explosion pipette, and I have found a large per cent. of the efficiency lost and unaccounted for was due to the unconsumed hydrocarbons, and I think that occasionally the flue-gas analysis should be continued through to the explosion pipette.

Professor Kent: I used the explosion pipettes a good deal and the results were negative and I could not tell whether I had any hydrocarbons or not. I have my doubt about their use, unless they are used in connection with more refined analysis.

John C. Parker: I called up a power house yesterday, where they burn about 200 tons of coal a day, and I said, "Have you a CO_2 recorder?" The answer came "Yes." "Are you using it?" "No." "How long since you used it?" "Four months." "What is the matter?" "Needs repairs." "Anything else?" "Yes, I have to give it my personal attention." As far as I have had experience in going through the power stations of the country, I have usually found the recorders out of service; they tested them and then removed them. The CO_2 recorder is an interesting device; it is very clever, but it is more of a laboratory device than it is an instrument for the boiler room. It seems to me that, so far as I am personally concerned, if I can get control of the fire, get control of the grate so that I can burn the fuel, and if I can have some method of knowing just how much steam output I am making, I do not care whether the temperature of the fire is high or low, the CO_2 high or low, or anything else—all I want to do is to produce a large amount of steam with a small amount of coal.

We ran an 88-hour test recently, different grades of work, and we were surprised to find on the single-ended test, where there was undoubtedly more surface air than with the double-ended test, that the efficiency was higher. I have heard it stated quite frequently that a high CO_2 does not mean a high efficiency, and my experience justifies that view. Mr. Uehling says: "Air infiltration lowers the stack temperature in proportion to its ratio to the actual products of combustion." In one of my first boilers I varied the amount of air above the grate, and as I admitted more air my stack temperature went up. Of course, if you let enough air come in you can undoubtedly reduce your stack temperature. On one test I was taking the flue-temperature readings, and I watched the pyrometer from the time the man covered the fires until he covered them the next time—I wanted to find the difference in the flue temperature, the effect of different periods between the firing. I told him not to fire until I gave him instructions, and I watched the pyrometer for a half hour, and the flue temperature kept going up for a half hour. I got tired then and went to look at the fire, and it was burned out and was all ashes.

A. W. K. Billings: I do not think we should lose sight of the fact that what we are after is not efficiency in combustion but economy in the total operation of the plant. With chain grates it is possible to get high CO_2, and yet if you continue to operate them you will have increased maintenance expense and difficulties generally that will offset the ostensible gain in fuel. The fact that CO_2 recorders in general are insufficient to bring economy has come to my attention in my work, in which we have to deal with low-grade firemen and in which the cost of fuel is expensive.

I have been struck with the facility with which a fireman can beat the recorder. The percentage of CO_2 rises to a maximum and drops off when you get imperfect combustion. When the men are not subject to close supervision, the firemen, by unduly thickening the fuel bed and shutting off the draft, can shirk the work and force the load on other firemen and get a high CO_2 record, but at the expense of clinker troubles and other troubles generally.

I have noticed, especially in the case of large and important plants, that their experience with CO_2 recorders was a good deal such as I had on one occasion when I visited one of the largest and most efficiently operated plants in the United States. I was interested in a certain type of recorder, which I understood they used, and I asked where it was. They led me to a certain aisle where it was supposed to be, and it was not there. We asked the foreman where it was, and we went up another aisle and found one-half of it. It was not accomplishing the purpose for which it was designed.

There is a tacit assumption in this discussion that we must confine ourselves to the CO_2 in determining the efficiency of a plant. I think that is an error. The small fractional differences of boiler pressure show a more practical means of checking the performance of the boiler, and after all it is necessary to make a practical study of how you want your firemen to fire and how the details of the plant shall be handled. It is immaterial whether we observe the CO_2 or adopt other means of watching how the firemen handle the fire.

Under conditions which precluded other successful methods of determining the efficiency of the plant, I have tried the plan of taking the difference in air pressure through the fuel bed, and the difference in air pressure through the passages, taking a simple device, such as a Bristol recording instrument, to show that ratio. I have found that it could be developed into something that would be a very practicable means of watching efficiency, because it gives instantaneous readings and the apparatus is simple.

There is another point to be dwelt upon in watching fuel economy, by using a mechanical device which will tell what the fireman does—a point seldom mentioned—and that is under uniform conditions of regular tests, the gas analysis may be correct, but no matter how correct the analysis, if you have theoretical conditions, there is an error which may enter the calculations, and that is the time element. A continuous sample may not be representative of the average of the gas, which is the true measure of the losses, and while under normal conditions that amounts to little, under many conditions it can amount to 10 or 15 per cent. of the loss.

The paper of E. Viola upon the automatic control of condensing water related principally to the use of condensers in connection with vacuum pans for sugar manufacturing, etc. The discussion evolved nothing of interest to power-plant engineers. An abstract of the paper will appear in an early issue.

The paper upon a "Test of a 10,000-Kilowatt Steam Turbine," by S. L. Napthaly, and upon the "Test of a 9000-Kilowatt Turbine Generator Set," by F. H. Varney, were discussed together.

Discussion on Turbine Papers

An abstract of Mr. Varney's paper was published in the December 13 number, page 2206, and a review of the paper by Mr. Naphtaly appears in this number, page 2221. The discussion on both papers follows:

J. H. Lawrence: Mr. Naphtaly gives a table which is a summary of the tests made on the 10-000-kilowatt unit; one column is headed B.t.u. per kilowatt-hour.* His results vary considerably from those which I have calculated, and give the turbine a much lower heat consumption than I think it should be credited with. I have tried various methods and have used both the old and the new steam tables, but have not been able to check the figures given in the table.

The heat consumption of a turbine is the difference between the heat content of the entering steam and the heat of the liquid in the exhaust steam, multiplied by the pounds of steam per kilowatt-hour. Accordingly, for run B with 168 pounds gage pressure, 59 degrees superheat, 28.18 inches vacuum and 14.427 pounds of steam, a heat consumption of (1232.7 — 66.1) 14.427 or 16,830 B.t.u. per kilowatt-hour is obtained. This result is almost 12 per cent. higher than that given by Mr. Naphtaly. The corresponding thermal efficiency is 20.27 and not 22.65 per cent.

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*For the table see page 2221 of this number. The column on B.t.u. per kilowatt-hour was omitted, but is reproduced in the accompanying table.

Mr. Naphtaly's figures for Rankine cycle ratio check very closely with those which I have calculated.

The following is a table showing the revised results, calculated with the Marks & Davis tables:

REVISED SUMMARY OF TESTS ON 10,000-KILOWATT TURBINE.

	Load, Kw.	Steam Pressure, Lb. Gage.	Superheat, Deg. F.	Vacuum, 30″ Bar.	Water per Kw.-hour.	B.t.u. per Kw.-hour.	B.t.u. per Kw.-hour from Naphtaly Table.	Rankine Cycle Ratio.	Thermal Efficiency, Per Cent.
A . . .	7,972	171	58	28.28	14.581	17,035	15,378	66.8	20.04
B . . .	8,563	168	59	28.18	14.427	16,830	15,062	68.2	20.27
C . . .	8,198	169	60	28.10	14.596	17,017	15,414	67.6	20.05
D . . .	9,173	167	59	27.90	14.572	16,926	15,360	68.9	20.15
E . . .	5,333	173	54	28.34	15.655	18,274	16,493	61.8	18.67
F . . .	8,148	167	60	26.16	15.855	18,093	16,782	69.5	18.86
G . . .	5,401	174	56	26.16	17.611	20,075	18,587	62.2	16.99

Dr. D. S. Jacobus: I wish that some of the gas-engine men would get onto their feet and say how much better they could do on the B.t.u. consumption. I have not checked up these B.t.u. figures myself, but he may have had a heater in his exhaust, and possibly done something there—I am not sure of that, but the figures as given here take about 15,000 B.t.u. per kilowatt-hour as they stand in the table, and one test would correspond with a boiler efficiency of 70 per cent. to about 22,000 B.t.u. per kilowatt-hour, which is a very low figure. The lowest figure for actual work, day after day, is that at Oakland, where, with an oil plant, the record was made of a kilowatt-hour for 25,000 B.t.u., including everything else about the plant. It would be interesting to hear some of the gas-engine men say how close they could get to these two figures—first, 22,000 B.t.u. for a steady load, and, secondly, 25,000 B.t.u. with a curve in which the load fluctuated greatly, and where there was a layover of 4.5 hours per day, and where the 25,000 B.t.u. included everything about the plant, such as the lighting, and represented the B.t.u. in the oil as compared with the net kilowatt output of the plant, and as paid for by the consumer.

Irving E. Moulthrop: Referring to Mr. Varney's paper, it is interesting to note that the steam turbine has operated satisfactorily when paralleled with a large number of water-power stations, and its service has been more satisfactory in regulation and maintenance than that from either steam engine or gas engine. The engineers of the Pacific Gas and Electric Company are to be congratulated on the speed with which a 9000-kilowatt steam-generating station was completely built and also on the very low kilowatt cost of this station.

It is interesting to note that on the test the water rate of the turbine was inside the manufacturer's guarantee, but it seems to me that this actual performance was not especially creditable to the Curtis turbine of 9000 kilowatts maximum capacity. Of course, the vacuum during the test was not especially good and with injection water of 51 to 52 degrees it should be possible to reduce the absolute back pressure by about one inch, which would very much improve the water rate of the turbine.

The superheat of 70 to 75 degrees on the maximum-load test is also rather low for the best results. One hundred and fifty degrees is safe for superheaters and piping from our experience and would better the economy some 6 per cent.

Both of these changes would probably have increased the investment. A gain of about 1¾ pounds in the water rate per kilowatt-hour would seem to be easily obtainable and would justify quite an investment charge.

It is interesting at this time to call attention to a test of a 3500-kilowatt horizontal Curtis turbine in a plant of the Great Western Power Company at Oakland. At 3464 kilowatts load, 194.35 pounds pressure at boilers, 127.5 degrees superheat, 1.17 absolute back pressure, inches mercury, the water rate was 13.62 pounds per kilowatt-hour. For the size of the unit this seems very good.

At the L street station of the Boston Edison Company, a 5000-kilowatt five-stage, Curtis turbo-unit tested in 1907 shows the following results:

Load, kilowatts . .	2,558	5,195	7,526
Steam pressure, lb.	173.0	173.7	169.7
Superheat, degrees Fahrenheit.	149	142	134
Absolute back pressure, inches Hg.	1.13	1.18	1 35
Pounds water per kilowatt-hour . .	15.24	13.57	13.73

This same machine has been tested once a month since being installed and the best record of actual water rate came with the highest superheat and the best vacuum: 5095 kilowatts, 170.4 pounds steam pressure at throttle, 179.1 degrees superheat, 0.52 inch back pressure, Hg., 12.71 pounds water per kilowatt-hour. The corrected economy to standard conditions of this test is slightly poorer than in the test given in the table above, so any possible error in this test is in the direction of making the water rate too high rather than too low. These figures seem to show conclusively the advantage of high superheat and vacuum, at least on the impulse type of turbine.

A glance at the correction figures used with the various units spoken of is most interesting, as they suggest the possibilities of the various kinds of turbines under different conditions of superheat and vacuum:

Type	G. E. Vertical	West. Double-flow	G. E. Vertical	G. E. Vertical
Company	Pacific G. & E.	City Elec. Co.	N. Y. Edison	Boston
Station	Oakland	San Francisco	New York	L Street
Capacity of unit maximum.	9,000 kw.			
Capacity of unit normal.		10,000 kw.	8,000 kw.	5,000 kw.
Corrections for pressure, 1 per cent for every.	10 pounds	20 pounds		10 pounds
Superheat, 1 per cent. for every. .	12½ deg.	10 deg.	12 deg.	12 deg.
Vacuum per inch normal load . .	1.8 pounds	0.45 pound	0.985 pound	1.15 pounds
one-half load . .		0.91 pound	1.35 pounds	1.60 pounds

Mr. Naphtaly has contributed some very valuable data upon the Westinghouse double-flow turbine, a type of machine about which very little information has been published. The unit is large, having a normal rating of 10,000 kilowatts and an overload capacity of 50 per cent. As the rise of temperature on the generator is only 60 degrees Centigrade after twenty-four hours of 50 per cent. overload, this unit might be given a rating of 15,000 kilowatts maximum.

It seems from the test that this unit is very efficient, especially at the low superheat and vacuum used.

These figures would indicate that the impulse turbine received more benefit from an increased pressure than the combined impulse-reaction machine, which is very curious, to say the least, as the high-pressure elements are of the same type. The benefit from an increase in superheat is marked in both the types, with the advantage slightly in favor of the double-flow type. The benefit from an increase in vacuum is very much less in the double-flow combined impulse-reaction turbine than in the straight impulse turbine, especially at the heavier loads (42 per cent. as much at full load, and 62 per cent. at one-half load).

The designer should provide for moderately high superheat and as good a vacuum as the inlet-water temperature will warrant, and then the operating engineer should continually keep after the condensing apparatus in order to get every bit of vacuum obtainable with the apparatus furnished him. This is true of both types of turbines discussed but especially true of the straight-impulse turbine.

George A. Orrok, of the New York Edison Company, presented the accompanying table in which are the results recalculated on the same basis of tests which have been available during the last few years, using the new Marks & Davis steam tables. The Rankine cycle efficiencies are neglected since in most cases the efficiency of the generator and the mechanical efficiency of the turbine are not given and cannot be easily obtained.

It is much simpler to compare the various machines on the basis of the B.t.u. in the steam at the throttle valve, and to credit the turbine with that portion of the heat turned back into the feed-water system at the condenser temperature.

The table gives actual test results, pounds of steam per kilowatt-hour, vacuum, superheat and steam pressure not corrected to the contract conditions.

He regretted not having at his disposal the latest Parsons performance in England, nor the latest A. E. G. performance on its 12,000-kilovolt-ampere machine, which it is reported is much better than any of the figures given. There is no doubt that from the larger-sized apparatus better results are being obtained.

ECONOMY TESTS OF STEAM TURBINES.

Turbine.	Load Kilowatt.	Steam Pressure, Pounds Absolute.	Superheat of	Vacuum.	Pounds Steam per Kw.-hr.	B.t.u. per Kilowatt.
Rummelsburg A. E. G.	4,179.6	179	289.3	29.19	11.95	15,665
Carville Parsons	5,164.1	207	125.2	29	13.15	16,122
Chicago Curtis	10,816	190	147	29.47	12.9	16,206
Berlin Moabit A. E. G.	3,150	185	225	28.5	13	16,352
Boston Curtis	5,195	179.5	142	28.8	13.52	16,578
City Electric San Francisco Westinghouse	8,563	183	59	28.18	14.427	16,850
N. Y. E. Westinghouse	9,830	192.2	96	27.31	15.15	17,778
Brown Boveri Rheinisch Westfall	5,128	176	193.7	28.47	14.32	17,790
N. Y. E. Curtis	8,880	192.5	108.5	28.1	15.05	17,940
Pacific Curtis Oakland	8,775	194	72.95	28.03	15.95	18,735

E. D. Dreyfus: In the days of the Corliss engine, much interest was centered about discussion of initial condensation, cylinder ratios, jacketing, reheating receivers and other features affecting steam economy. The experience obtained from over 2000 steam turbines in service should now reveal the operating conditions and design which realize the highest final efficiencies. Uniformity of operating conditions has already become evidently, and very probably a generally accepted type will be prevalent eventually.

The author is to be commended both for having selected moderate working conditions by means of which he obtains the highest ultimate economy and for having indorsed an increase in rotative speeds which obviously conduces to higher efficiencies.

While the water rates which Mr. Napthaly obtained with his turbine (14.57 pounds as tested, and 13.88 pounds corrected) somewhat exceed other steam consumptions that have been recorded, there has, nevertheless, been established a new mark in the efficiency of conversion of the available thermal energy between the throttle and exhaust pressures into electrical energy at the switchboard; viz., 69 per cent. as referred to the Rankine cycle.

Until the turbine came to be extensively used, the Rankine cycle measurement of efficiency was but seldom employed in general practice. This may be ascribed to the fact that the efficiency of the compound or triple-expansion reciprocating engine does not improve proportionately with the amount of added energy made available by the higher vacua and superheats. Manifestly high vacua, so beneficial in the turbine art today, were not regarded seriously in reciprocating-engine practice. With many of us, the value of these factors has been insufficiently considered, and therefore greater familiarity with the temperature diagram is desirable. Such diagrams were formerly used in the critical analysis of steam-engine indicator cards, but they will in this case forcibly emphasize the influence of operating conditions. For example, with initial conditions of 190 pounds and 100 degrees Fahrenheit superheat, a one-half pound reduction in pressure at the exhaust end (from 28 to 29 inches vacuum) increases the available energy as much as the addition of over 100 pounds at the inlet of the first stage. This, of course, is of purely academic interest. In practice, the addition of an equivalent amount of energy would be made available by a further increase of 140 degrees Fahrenheit, or 240 degrees total.

Doubling the rotative speed reduces the steam consumption from 4 to 6 per cent., depending upon size. In 1000-kilowatt turbines of 3600 and 1800 revolutions per minute, under United States Government test, a difference of 4.5 per cent. was shown. Large turbines, such as those under discussion, would probably improve somewhat more.

The above efficiency, as already stated, has not been equaled in any other published test. It is to be remembered that this percentage includes all armature and windage losses of the generator, as well as the field losses which are commonly omitted.

It is believed that the efficiency ratio consistently improves until 100 degrees (Fahrenheit) superheating is exceeded, when further delay of the "dew point" or occurrence of saturation is offset by the greater amount of heat in a pound of steam passing out to the exhaust. Hence it is not improbable that the turbine would have accomplished a slightly better record with additional superheating—60 degrees being the average throughout the test.

The effect of vacuum is more indefinite and is subject to the areas and angles provided in the final stages of the turbine.

It is sometimes indiscriminately stated that the higher the vacuum the better the over-all plant results. This is plainly a narrow perspective, as the power consumption of the condenser auxiliaries is thus entirely ignored. Condenser temperatures fall off very rapidly with the increase of vacuum, as is shown by the standard steam tables, and the corresponding capacity and power required by the air and circulating pumps become proportionately greater. There is, consequently, a point where the improvement in the turbine fails to exceed the increased power demanded by the auxiliaries, and in the latitudes we are accustomed to we generally find the most economical yearly average vacuum varies between 28 and 28.5 inches (referred to 30 inches barometer), depending upon the source of water supply. While fuel economy is an important factor of the total cost of power, manifestly investment and operating expenses must also be reckoned with. Fortunately I have been able to obtain what I believe may be considered conservative installation and monthly operating costs from Mr. Napthaly's plant. These results plainly indicate efficient management, both in its construction and subsequent operation. The entire station consists of two 2500-kilowatt, and one 6000-kilowatt turbine, in addition to the 10,000-kilowatt unit.

A typical monthly payroll was $3565. Oil, waste, supplies, water and maintenance material averaged for three months, $769. The corresponding oil-fuel consumption was 20,800 barrels for the entire station, and the output was 4,908,800 kilowatts. The smaller units increase the amount of wages, and therefore only three-fourths, or less, is strictly chargeable on the basis of the use of large units.

Considering the new addition provided by the installation of the 10,000-kilowatt turbine which has been described, the cost to the City Electric Company may be conservatively quoted at $339,413, or $33.94 per kilowatt, including building, foundations, boilers, piping conduits, oil storage and burning equipment, auxiliaries and electrical apparatus—in fact, everything from the cost of real estate up to and including the switching gear.

The low total cost of a complete expansion turbine (less investment and cost of labor), it is believed, settles the question definitely as to the advisability of introducing a combined engine-turbine unit in a new station to accomplish a small saving in fuel. By employing the higher speed of 1800 revolutions per minute instead of 900, the unit has been reduced in length about 17 per cent. Mr. Napthaly developed his plans providing for two rows of turbines in the engine room, having them alternately faced with respect to one another. The 10,000-kilowatt turbines of the type described demand an engine-room floor space of only 0.18 square feet per kilowatt of normal capacity installed, which figure, it is thought, also constitutes a new record.

I am somewhat at variance in opinion with Mr. Moultrop on the advantage of vacuum. Every type of machine may be designed for the same degree of efficiency with a given vacuum. The change in economy of any particular type, with the change of vacuum, depends on the number of stages or rows of blades which it contains—therefore the machine with the fewer rows of stages is more sensitive to changes under operating conditions and will more rapidly decline in efficiency, if the auxiliary station equipment is not maintained up to the original standard.

As regards the question asked by Professor Jacobus, in relation to the gas engine, the large gas engine would not be a competitor for the large steam turbine. When you come to the smaller sizes, where the gas engine maintains its efficiency and the turbine falls off, there is a field for the gas engine.

Francis Hodgkinson: I had the pleasure of being present at the time this test was made, and while it is true that this was not a scientific test, in the sense that losses were all segregated, and that sort of thing, I do wish to state there was not a stone left unturned to insure its accuracy. I am in thorough accord with Mr. Moultrop when he says that it is unfortunate that the load conditions of the plant were such that they could not go to the heavy overloads, as the paper would then be of greater interest.

Mr. Moultrop quoted a number of tests, in which he showed some figures of steam consumption. We know that the pounds of steam per hour is not the whole thing. You must look at the whole thing, and consider the operating conditions. Mr. Jacobus brought out the fact that high superheat costs money to get, and is not an unmixed blessing. In the table in the paper is a column showing the Rankine cycle efficiency, for efficiency ratio. That is, after all, the best comparison we have, but it is not necessarily an absolute one. One turbine may be designed for a 26-inch vacuum and show a certain efficiency, and another turbine may be designed for a 29-inch vacuum and show another efficiency, and because the turbine with the 29-inch vacuum shows the lower figure, it does not follow necessarily that it is the worse turbine.

R. J. F. Pigott: In regard to turbine tests of recent years, it is a happy thing that manufacturers and owners of these machines are coming to feel that they can publish more data than we had several years before, but it is not an unmixed blessing, because most of them, while they probably take a great deal of pains in getting their data, they know it so well that they forget to put it in the papers, and the papers lack any data from which the readers can compute what they desire to know. This paper is open to that fault, I think.

The trouble which the other gentlemen found, I also found, in checking from the B.t.u. values. There is nothing in the paper which would guide one to a knowledge of how the B.t.u. value was computed, and this tends to throw doubt on the paper.

Mr. Naphtaly refers to the turbines at Fifty-ninth street, and states: "It is acknowledged that slightly superior efficiency exists with the combined engine and turbine unit, reported by Mr. Stott, at 72 per cent., the engine unit particularly making a remarkable showing. On the heat-consumption basis, however, the San Francisco unit is approximately only 3½ per cent. in excess of the combined reciprocating and turbine unit, but it is obvious that this advantage in steam consumption is offset by the advantage of the simple turbine unit, without the complication of the reciprocating engine, which involves greater first cost, greater floor space and cost of maintenance and necessitates shutting down the unit while the reciprocator is being adjusted, or else further complicating the system by use of an auxiliary governor and high-pressure valves on the low-pressure turbine." He is making his comparison, in his case, with 60 degrees superheat, and in the case of the turbines at Fifty-ninth street with saturated steam. If we compute the turbines at Fifty-ninth street with superheated steam, it would reduce the water rate from 13.2 to 12.3 pounds per kilowatt-hour. There is nothing to prevent the installing superheat and taking advantage of that. The real advantage of the low-pressure turbine, therefore, in this case is, as was stated in the paper by Mr. Stott, 13.5 instead of 8.5. Taking the lower figures, if the units are operated on normal load, it is about $70 or $90 a unit. The additional expense of more men to handle a turbine is about $8 a day, and the additional expense, based on the greater cost of the reciprocating unit over the turbine, is $12 a day, leaving a margin for taking care of additional repairs on the engine over what a turbine would be forced to have. These are based on a cost per kilowatt of from $83 to $100 for the engine system alone, the average figure being $85 to $90. In the case of some of the big plants, erected some years ago, it was $100. The low-pressure turbine costs $40 to $80 per kilowatt, which brings the average down to $65. The cost for the high-pressure varies from $65 to $68, so that the actual increase in cost, due to low-pressure turbine is not very great, and the interest is taken care of by the savings per day of 8.5 per cent. better economy. Mr. Naphtaly says: "But it is obvious the disadvantage in steam consumption is offset by the advantage of the simple turbine unit without the complication of the reciprocating engine." That statement is quite true in regard to new plants, because I do not think anybody is likely to design and build a large station having reciprocating engines and turbines. The high-pressure turbine has increased in economy too rapidly for that.

W. L. R. Emmet: Mr. Varney's paper relates to a machine of which a great many are installed and in operation, the five-stage Curtis turbine, operating at 750 revolutions per minute, and Mr. Naphtaly's paper relates to a newer development of the Westinghouse machine operating at 1800 revolutions per minute, the capacity of the higher-speed machine —the rating—being larger. The comparison will naturally be made by readers of these papers, who have heard them discussed together, as to the relative value of these units and the records in these papers look, upon inspection, very unfavorable to the Curtis unit. The results reported in Mr. Varney's paper are about a pound per kilowatt-hour worse than the results which Mr. Moultrop has reported on his Boston turbine, two or three of which have been tested with uniform results, and which are identical with the machine reported in Mr. Varney's paper. The machine in question had been in operation about a year when it was tested, and simply tested under station conditions, and the possibilities of considerable errors in such tests are very well known. The other machine reported in Mr. Naphtaly's paper, I had occasion to study a little bit at the time of the Edison convention, because I saw the report submitted by the Westinghouse company to that association, about the same information which is contained in Mr. Naphtaly's paper.

Isaac Harter, Jr., presented some "Notes on the Value of Napier's Coefficient with Superheated Steam," which, with the discussion, will be treated in a later issue.

In lieu of the usual reception on Thursday evening, a reunion was held at the Hotel Astor on Wednesday evening. This is the principal opportunity afforded during the meeting for social intercourse, and its opportunities were enjoyed by a large proportion of the attending membership.

Thursday Sessions

On Thursday forenoon "A New Theory of Belt Driving" was presented by Selby Haar. "A Graphical Method for Calculating Stresses in a Connecting Rod," by Winslow H. Herschel, "Operating Conditions of Passenger Elevators," by Reginald P. Bolton, and "Modern Shoe Manufacture," by M. B. Kaven and J. B. Haddaway. In the afternoon the session in the auditorium was devoted to machine-shop practice, and the gas-engine section held a separate session in another hall.

Mrs. and Mr. Westinghouse received the members of the society and guests from 4 to 6 at the Hotel St. Regis.

Graphical Method for Calculating Stresses in Connecting Rod

W. H. Herschel called attention to the fact that while empirical formulas are sufficiently accurate for designing slow-speed engines, when it comes to high-speed engines other forces are involved which make the use of such formulas questionable. On the other hand, analytical formulas are unsatisfactory because they are, for the greater part, based upon incorrect assumptions.

To avoid the assumptions generally made in calculating stresses in a connecting rod by analytical formulas, a graphical method is proposed for finding the stress at any point of a rod of any shape. Determination of the inertia forces of material points is simplified by means of a diagram calculated from exact formulas. The bending moment due to inertia forces is calculated by the usual string-polygon method, and there is also a moment equal to the axial method, and a moment equal to the axial thrust multiplied by the sum of the deflection and an assumed eccentricity of loading. Deflections are found by Mohr's method and the eccentricity is assumed in accordance with the conclusions of Moncrieff.

The numerical examples show that in a simple shaft of uniform stress the maximum bending moment and maximum diameter will be less than 0.6 the length of the rod from the crosshead end, and the maximum bending moment at any one point will occur at about 38 per cent. stroke. The stresses are much greater than given by the usual formulas, mainly on account of considering the bending moment due to axial thrust.

Gas Power Section

The session of the Gas Power Section was held on Thursday afternoon, President Bibbins presiding. After listening to the president's address and the reports of the various committees the society proceeded to consider the papers. Two papers were presented, one on the "First Large Gas Engine Installation in American Steel Works," by E. P. Coleman, and the other, "Notes on a Large Gas Producer," by Nisbet Latta. An abstract of the former, with the discussion, will appear in a later issue, and the latter paper is presented in the Gas Power Department of this number. Owing to the untimely death of Mr. Coleman, his paper was abstracted and read by E. E. Kreiger. Also, Mr. Latta was prevented from being present and Mr. Lent presented an abstract of his paper.

In discussing Mr. Latta's paper, Mr. Chapman recalled the fact that he had visited the Hirt producer while it was in operation and had noticed that it emitted smoke which extended a distance of a quarter of a mile. He was of the opinion that a better analysis than those shown could be obtained by putting in less air, in which case the CO_2 would decrease; but when enough air was admitted to make a good commercial mixture, the CO_2 increased and the producer became more of a furnace. As for the statement that "a large producer is impracticable owing to the difficulty of its operation and the difficulty of stoking," Mr. Chapman contended that it applied only to hand-stoked producers, as the large ones must be stoked mechanically, and upon mechanical agitation with a deep fuel bed largely depends the elimination of blowholes having an intense temperature.

Mr. Winchop referred to the fact that the analyses given in the paper showed a low heat value of the gas for a high heat value of the coal, representing a rather low efficiency of the producer, the high temperature apparently working against a high efficiency.

At the conclusion of the discussion, the new president, Mr. Fernald, took the chair and after making a short address adjourned the meeting.

Friday was devoted to excursions to various points of engineering interest in the city and vicinity, a large number of which had been provided for the selection of the visitors.

Philadelphia Engineers Hold Banquet

The second grand concert, supper and dance of the Combined Associations of the National Association of Stationary Engineers of Philadelphia, Penn., was held at Lulu Temple on Tuesday evening, December 6. The combination included Quaker City No. 1, Tacony No. 9, Merrick No. 12 and Northeast No. 20. Notwithstanding the severe snow storm which prevailed, the large auditorium was well filled with the members and friends of the several organizations, many prominent personages of the engineering fraternity being present.

The festivities began with an enjoyable entertainment by the New York "Bunch," comprising Herbert Self, Peerless Rubber Manufacturing Company; Frank Corbett, Consolidated Safety Valve Company; Billy Murry, Joe McKenna, Johnny Forsman, James Devins, Jenkins Brothers; and Jack Armour, of POWER. In an interval during the entertainment Past National President W. J. Reynolds called Past President Lynch to the stage and presented him with a handsome gold jewel. It is the intention of the national body to present to all of the living past presidents a similar emblem.

At the conclusion of the musical program the guests adjourned to the dining hall, where an appetizing dinner was served. Dancing followed the banquet.

The officers and representatives from the various associations acted as a general committee and they were congratulated upon the completeness of the arrangements.

They started Jones out from the terminal in front of fifty loads with No. 5637 and a lot of leaky tubes. Jones had reported 5637 the last time he took her out so he was wise to the reason for her poor steaming and kept her going as long as he could, but there was a limit and finally Jones wired in from tower "WR":

"X5637 died at WR; what shall I do? Jones."

The master mechanic replied:

"Was death sudden or lingering? Miles."

To which Jones responded:

"Lingering and long drawn out."

His reply from the master mechanic was:

"Come in. Inquest Thursday."

Sid Diggles hez a old yaller tom cat 'round his ingin room. It fell inter th' flywheel tother day an' after swingin' 'round th' circle a few times, flew out an' landed on th' bib uv Sid's overalls. Sid dubbled up like a jack knife an' cudn't get breth enuf t' cuss th' cat fer 'bout two hours; he's bin makin' up fer it ever since tho. He says it's th' wust catastrophe thet ever happened t' him in all uv his experience.

Inquiries of General Interest

Annealing Brass

I wish to anneal the ends of some brass heater tubes before expanding them in place. How can I do it?

A. B.

It is customary in annealing brass and some other metals to heat to a dull red and cool in water. The result will be the same if allowed to cool slowly.

Effect of Extra Lap

If lap is added to a plain slide valve, will it be necessary to change the eccentric?

E. E. L.

To give full port opening the travel of the valve must be twice the width of one steam port plus twice the lap. If the travel of the valve with the added lap will not give full port opening, the throw of the eccentric should be increased.

Proper Position for Globe Valves

In putting globe valves in a horizontal pipe line, how should they be placed?

P. G. V.

Valves should always be placed as near the supply end of the line as is possible with the stem horizontal to avoid a possible water pocket and with the pressure under the disk, so that the valve may be closed and the stem packed while under pressure.

Method of Banking Fires

My fires are banked for about 14 hours. I close all doors and the damper, but sometimes the fire burns through and the safety valve blows. How can I prevent it?

M. B. F.

By opening the flue doors just enough to furnish what air leaks past the damper. In this way no air is drawn through the fire or ashpit doors.

Sulphur in Feed Water

Is water saturated with sulphur suitable for boiler-feed water?

S. F. W.

Sulphur is not soluble in water unless in chemical combination with some other element as iron or lime. With lime it forms scale and with iron it causes pitting and corrosion and is not desirable in feed water.

Difference in Tube Expanders

Is it good practice to use a tapered plug in expanding boiler tubes? Would not a roller expander be better?

D. T. E.

Good practice depends more on the manner in which any tool is used than upon the tool itself. The roller expander is convenient and more easily handled than the plain plug, but in the hands of a skilled mechanic the work done with either will be equally good.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Fusible Plug Composition

What composition is used in filling fusible plugs?

F. P. C.

Fusible plugs are or should be filled with pure tin which melts at 442 degrees Fahrenheit.

Position of Check Valve

Should the check valve in the feed pipe be placed between the boiler and the globe valve, or should the globe valve be next to the boiler?

P. C. V.

There should be a valve or cock between the check and the boiler which may be used if it becomes necessary to repair or replace the check.

Starting Cross-Compound Engine

If a cross-compound engine has no bypass to the receiver, how may it be started when the high-pressure side is on the center?

S. C. E.

By slightly opening the throttle and working the high-pressure wristplate by hand until there is pressure enough on the low-pressure piston to move the engine off the center.

Underground Pipe Covering

We have two lines of steam pipe, 400 feet long, at present laid underground in wooden box packed with shavings; this method is unsatisfactory as the shavings accumulate dampness, and the pipes decay from the outside. What do you think of laying the pipes in tile?

U. P. C.

The same practical common-sense methods should be observed in laying steam pipes underground as above. They should be protected by the most efficient nonconducting covering available and kept dry on the outside.

Water Temperatures in Engine Calculations

What five temperatures of water are often used in power-plant calculations?

T. E. C.

32 degrees, melting point of ice.

39.1 degrees, temperature of greatest density.

62 degrees, standard temperature.

70 degrees, standard temperature for condenser injection-water calculations.

212 degrees, boiling point at sea level.

Engine Duty Defined

What is meant by the term duty when applied to a steam engine?

E. D. D.

It is the work the engine is capable of doing on a given fuel or steam consumption. It is stated in pounds of fuel, combustible, or steam per indicated horsepower, or foot-pounds of work for a given number of pounds of steam, or of heat units supplied.

Direct and Indirect Engine

What is the difference between a direct and an indirect engine?

D. I. E.

An engine is direct when the motion of the valve coincides with that of the eccentric, and indirect when the motion imparted by the eccentric is reversed by the interposition of a rocker arm.

Open and Closed Feed Water Heaters

What is the difference between an open and a closed feed-water heater?

F. W. H.

In the open heater the water and the steam intermingle in the same chamber or vessel, which is open to the air, and from which the water flows to the pump by gravity.

In the closed heater the water and the steam are separate, the water usually passing through tubes which are surrounded by steam. The water is forced through the tubes on its way to the boiler by a pump or injector.

Blowback in Crane Pop Valve

How is the blowback in a Crane pop safety valve adjusted?

B. C. V.

It is not adjustable at all in the general acceptation of the term. The tension of the auxiliary spring which controls the degree of blowback is fixed at the factory and cannot be changed without dismantling the valve and changing the tension of the spring. The reduction in pressure before the valve closes will be the same regardless of the pressure at which it may be set to blow.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

Of this issue 34,000 copies are printed.

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Please do not ask us to start subscriptions before the December 6 issue. All previous issues are exhausted.

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Power and The Engineer

Editorial

Vacuum and Pressure

There is a great deal of looseness in the treatment of that subatmospheric pressure which is referred to by the engineer as a vacuum. Instead of expressing it directly in terms of the pressure existing in the vacuous space above absolute zero it is expressed in terms of the difference between the zero pressure and that of the atmosphere, as the hight of a column of mercury which will be sustained by the atmospheric pressure against the lower pressure in the vacuous space.

This would not be so bad if the atmospheric pressure were constant, but the barometer is subject to considerable variation even in the same location, and is very considerably affected by elevation.

Suppose an engine or a turbine to be designed to operate between 135 pounds absolute initial and 2 pounds absolute back pressure. If it is stated in this way there can be no misunderstanding. The pressure and temperature in the boiler or steam chest must be 135 pounds and 350 degrees, and the pressure and temperature in the condenser must be 2 pounds and 126 degrees, and this whether the plant is at Coney Island or on Pike's Peak.

The designer can determine how many heat units or foot-pounds of energy are available between these limits, and proportion his engine accordingly, and tests made under similar conditions so expressed would be comparable.

But suppose a specification or guarantee to be expressed in pounds gage for the initial pressure and inches of mercury for the vacuum.

For a gage pressure of 120 pounds there would be an absolute pressure of 135 pounds, in round numbers, at the sea level or with a 30-inch barometer; but for a barometric pressure of 27 inches the absolute pressure in the boiler would be only 133.5 pounds. This would knock off less than a degree of temperature and less than half a B.t.u. of total heat at the high-pressure end.

This is not serious, but on the low-pressure end the disturbance is worse. Two pounds absolute pressure means about 26 inches of vacuum with a 30-inch barometer, but with a 27-inch barometer a 26-inch vacuum means an absolute pressure of one inch of mercury, approximately half a pound, the temperature corresponding to which would be only 79 degrees instead of 126.

In other words, if the conditions are expressed, as is a common practice, as 120 pounds (gage) initial and 26 inches vacuum the temperature range with a 30-inch barometer will be from 350 to 126 = 224 degrees and the heat convertible into work by a perfect engine 274 B.t.u., while if the barometer is at 27 inches the temperature range will be from say 349 to 79, a difference of 270 degrees against 224, and the heat convertible into work by a perfect engine would be 348 B.t.u. against 274.

A low barometer is then an obvious advantage to the engine when the vacuum specified for it is expressed in inches of mercury. The expert who, holding a brief for the engine builder, waits for a high barometer with the idea that the engine will profit by the higher vacuum attainable fools himself, if that higher vacuum is produced by a higher atmospheric rather than a lower condenser pressure.

A vacuum of 26 inches with a 30-inch barometer is only 87 per cent. of the possible, and easily attainable. Its temperature with steam is 126 degrees and there is a wide difference between this and the temperature of available condensing water. A vacuum of 26 inches with a 27-inch barometer is 96 per cent. of the possible and the temperature of the vacuous space is with water vapor 79 degrees. There is a small difference of temperature between this and ordinarily obtainable condensing water, and the condenser man to meet this condition must furnish larger water pumps and air pumps which must be not only larger but more efficient.

Two practices have arisen, one of referring the vacuum to a 30-inch barometer and the other of expressing the vacuum in percentage of the possible, i.e., the obtained vacuum divided by the reading of the barometer. Neither of these solves the difficulty which is, however, completely disposed of by expressing the lower level in pounds absolute. The only disturbance which can come with this practice is that the air pump of a condenser designed to preserve this low pressure would have some less work to do with a low than with a high barometer, and for an abnormal case the barometric pressure expected could be worked into the specifications.

The Lap Seam Repeats

Another lap-seam boiler is added to the long list of those which have failed in Massachusetts where more time, effort and money have been spent to secure adequate boiler-inspection laws than in all of the other States combined.

In forbidding the installation of new boilers over 36 inches in diameter having lap-riveted longitudinal joints, a half step was made toward legislation for public safety, which, if completed by fixing a reasonable age at which all boilers of this type should be retired or have the allowable pressure so reduced that their operation for power purposes would be unprofitable, would spare many lives and save millions of dollars' worth of property. In this latest failure the old story of the hidden crack which can be found only by unmaking the seam is repeated, and one that will be told time and time again as these boilers one after another go to destruction.

The cause of the deterioration of the sheet just outside the outer row of rivets has been too often explained to need repetition. But the fact that all joints of this type are of uncertain strength and because of this uncertainty should be classed as unsafe cannot be too strongly emphasized.

Some years ago the idea was advanced that on account of the laminated structure of iron-boiler sheets it is difficult if not impossible for a surface crack to work its way entirely through the sheet, the crack being checked at the first lamination just as does the hole drilled stop the extension of a crack under other circumstances. Until recently the iron boiler has been immune from this kind of failure. But the Laconia boiler was of iron and the small size of the sheets in the New Bedford boiler indicates that it antedated the beginning of the period of the general use of steel for boiler plates.

Whether of iron or steel the final result of the flexure which takes place at every change in pressure will be the same. Failure is inevitable. Almost without exception the recent disastrous explosions have been in this type of boiler, many of them subject to the best obtainable inspection and high-grade, skilled attendance.

Massachusetts has taken advanced ground in the matter of boiler legislation and is to be commended for the good work done in the prevention of the perpetuation of the lap-seam boiler on the people of the Commonwealth. But it is regrettable that the hands of the Board of Boiler Rules have been tied to the extent that, although it may decide the eligibility of a new boiler for installation in the State, it may not forbid the use of one already installed on which an insurance company will take a risk. The lap seam has long been discarded in European boiler practice.

Good Machines and Good Men

When a man expends a certain amount of money for apparatus of any kind, it is generally done with the expectation of getting a machine that will make a saving over other types. Most men will pay a high price for good apparatus rather than a low price for an inefficient machine, because it is their expectation to more than save the extra cost of the better machine in operating expenses.

This is sound business sense, and the purchaser will not be disappointed if the machine is properly handled. There are cases, however, where high-grade machines are being operated in so ignorant or careless a manner that the results expected are not obtained.

In some cases perhaps the purchaser is influenced by a salesman into buying apparatus that is not suitable for the work, but even if a mistake has been made in the selection of machines, it is not good management to allow incompetent men to operate them just because they will work for a low wage. The saving in labor cost will never exceed the losses occasioned by inefficient management, such as is displayed in the following cases:

A certain manufacturing company installed a high-pressure boiler at a higher first cost than the same output could have been purchased in a low-pressure boiler. The idea was to carry a high steam pressure and attain a high efficiency in the engine.

An ordinary man, paid the usual low wage, was placed in charge of this boiler. In due time scale began to form in the tubes, and eventually they became so coated that some of them failed, and the plant was shut down until repairs were made.

Here was a high-priced boiler put out of commission and a manufacturing plant shut down because the man in charge failed to keep the tubes of the boiler clean. More than that the loss in heat transmission due to the accumulation of scale would doubtless more than cover the wage of the engineer.

In another steam plant a number of Corliss engines are installed. These engines are run seven days a week, day and night. There is no reserve unit, so that none of the engines can be shut down for repairs, and as a result the plant is in a deplorable condition. The pounding of the cranks in passing the centers is worse than might well be imagined and the wonder is that some of the crank pins have not been sheared before this. If the visible parts of the engine are so out of adjustment, what must be the condition of the valves?

Evidently there is no use of installing high-grade engines when so little attention is paid to their operation, either because provision has not been made for needed repairs and adjustments, or because an incompetent engineer is in charge. If economy is to be practised in one department of the power plant, it should be in all, and there is but little use in buying the best machines unless they are to be put in charge of competent men.

Boiler Explosions in Germany

"What man has done, man can do." The appalling number of boiler explosions constantly taking place in the United States may be ascribed by interested parties to the inherent depravity of inanimate objects. That, on the contrary, it is due to the inherent depravity of *animate* objects, is conclusively shown by comparison with the boiler-explosion statistics of Germany.

In the German Empire during the year 1909, there were only nine boiler explosions, six of which were attended by personal casualties. Altogether five persons were killed, eight seriously wounded and twenty-three slightly injured. Within the last thirty-three years there have been 518 explosions, by which 344 persons were killed, 211 severely and 509 slightly injured. By average, this is about ten explosions per year, killing ten persons, wounding six persons seriously and sixteen slightly. These figures are still more significant when analyzed, since in the recent years, when there must have been many more boilers in use, the actual number of disasters has run about the same, or even less, than in earlier years. The greatest number of explosions on the record is thirty-five, in 1894, and the highest account of deaths thirty-six, in 1879.

Of the nine explosions in 1909, five are believed to have been caused by low water; one by defective material; one by weakening of setting; one by corrosion, and one by breakage of a manhole bolt from too energetic use of the wrench, or otherwise. It does not appear that there was a water-tube boiler among them.

This is the season when the lecturer stands in the lantern's glare and descants upon everything from entropy to boiler-feed compounds for the benefit of the operating engineer. There is much to be gotten out of these lectures, and the opportunity to learn is not nullified by the fact that the lecturer may have as a motive the exploitation of his wares. One cannot know too much about the things he may be called upon to use, and discussion and a chance to hear the other man will bring out points about them that you never thought of.

The generally accepted belief in the safety of water-tube boilers seems to have received a severe jolt by the recent explosion in Brooklyn. The authors of text-books upon boilers will have to get out revised editions.

Lap Seam Boiler Explosions

At 11 a.m., December 9, the 42-inch by 16-foot return-tubular boiler in the planing mill of A. S. Allen, on North Water street, New Bedford, Mass., exploded, probably fatally injuring Mr. Allen and seriously hurting and scalding six others. The rupture started in the longitudinal seam of the rear course, tearing the sheet completely from the rear head and from the course in front.

As the initial rupture was at one side and near the rear of the boiler, the reaction of the escaping steam and water threw the boiler whirling through the side of the mill into the street. In the fire which followed, the two-story building was practically destroyed. In a later issue particulars and illustrations will be given.

Another explosion of a return-tubular boiler, resulting in two deaths, occurred at Weatherby, Penn., on Monday night, December 12. The boiler was one of five owned by the Read & Lovatt Manufacturing Company. Particulars will be given in an early issue.

PERSONAL

It is understood that Max Rotter has severed his connection with the Allis-Chalmers Company to take charge of the new works which are being built at St. Louis for the manufacture of Diesel engines.

SOCIETY NOTES

The next annual meeting of the Indiana Engineering Society will be held January 12, 13 and 14, at Indianapolis, at the Denison hotel. The feature of the convention will be the annual exhibit for manufacturers of anything of interest to the engineering profession. Space for exhibit can be secured on application to the secretary, Union Trust building, Indianapolis, Ind.

BOOK RECEIVED

An Introduction to Thermodynamics. By John Mills. Ginn & Co., New York. Cloth; 133 pages, 5¾x9 inches; 64 illustrations; tables; indexed. Price, $2.

Standard Practical Plumbing. By R. M. Starbuck. The Norman W. Henley Publishing Company, New York. Cloth; 406 pages, 6x9 inches; 347 illustrations; indexed. Price, $3.

The Indicator Handbook. By Charles N. Pickworth. D. Van Nostrand Company, New York. Fourth edition; cloth; 142 pages, 4¾x7¼ inches; 93 illustrations; indexed. Price, $1.50.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

INTERNAL COMBUSTION ENGINE. Ernest Easthope, Jr., Vancouver, B. C., Canada. 976,858.

ROTARY EGINE. John Putnam, Peoria, Ill. 976,913.

WATER MOTOR. Henry W. Yost, Springfield, Ohio, assignor to the Yost Gearless Motor Company, Springfield, Ohio, a Corporation of Ohio. 976,972.

FLUID-PRESSURE MOTOR AND CONTROLLING MECHANISM THEREFOR. William H. Caball, Chicago, Ill. 976,983.

ROTARY ENGINE. Harry A. Massey, East St. Louis, Ill., asignor of one-half to George H. Stein, St. Louis, Mo. 977,027.

TURBINE MOTOR. Thomas J. Loftus, Castella, Cal., assignor of one-half to Charles T. Loftus, Castella, Cal., and one-half to Harmon Bell, Oakland, Cal. 977,107.

ROTARY EXPLOSION MOTOR. Frederic Beck, Neuilly, France. 977,260.

ROTARY ENGINE. John Putnam, Peoria, Ill. 976,913.

INTERNAL COMBUSTION ENGINE. John Edward Sears, Jr., Newcastle-upon-Tyne, England. 977,334.

ROTARY ENGINE. John Putnam, Peoria, Ill. 976,913.

BOILERS, FURNACES AND GAS PRODUCERS

BOILER. Salvator Prebandier, Neuchatel, Switzerland. 976,912.

FURNACE FOR STEAM BOILERS. Francis Duffy, Chicago, Ill. 977,078.

BOILER CONSTRUCTION. Allison B. Stirling, Pleasant Mount, Penn. 977,144.

COMBINATION GAS AND OIL BURNER. Greene Allen Smith, Indianapolis, Ind. 977,231.

FURNACE. Burt L. Worthen, Tucson, Ariz. 977,249.

POWER PLANT AUXILIARIES AND APPLIANCES

GOVERNING MECHANISM FOR FLUID-PRESSURE ENGINES. Herbert H. Dow, Midland, Mich. 976,853.

VALVE. Thomas F. Payne, Pittsburg, Penn. 976,908.

CENTRIFUGAL PUMP. Ferdinand W. Krogh, San Francisco, Cal. 977,101.

VALVE. Henry Lippold, Erie, Penn., assignor to Lippold Valve Company, Erie, Penn., a Corporation of Pennsylvania. 977,105.

STEAM TRAP. Michael L. Richards, New Haven, Conn. 977,131.

TUBE-CLEANING APPARATUS. Clinton Robinett and John Patrick Reid, Dayton, Ohio, assignors to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 977,223.

PIPE COUPLING. Frederick Sargent, Glencoe, Ill., assignor of one-half to Olafe E. Oleson, Chicago, Ill. 977,226.

FUEL MIXER. Alfred Schmidt, Lake Hopatcong, N. J. 977,229.

COMBINATION GAS AND OIL BURNER. Greene Allen Smith, Indianapolis, Ind. 977,-231.

VALVE FITTING. Julius F. Bernstein, New York, N. Y., assignor to William J. Cunningham, New York, N. Y. 977,261.

BOILER FLUE. John Robert Crowley, Atlanta, Ga., assignor of one-half to Nathaniel Nelms Boyden, Atlanta, Ga. 977,276.

DEVICE FOR HANDLING FLUE CLEANERS. Melville O. Lewis, Peoria, Ill. 977,411.

INDICATOR FOR INTERNAL COMBUSTION ENGINES. Edward D. Meier, New York, N. Y. 977,419.

ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC ARC LAMP. Willy Legel, Berlin, Germany, assignor to General Electric Company, a Corporation of New York. 977,022.

DYNAMO ELECTRIC MACHINE. William H. Powell, Norwood, Ohio, assignor to Allis-Chalmers Company, a Corporation of New Jersey, and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 977,220.

ELECTRIC SWITCH. Percy L. Sibole, Wilkinsburg, Penn. 977,140.

OIL BREAK SWITCH. Harry J Hunsicker, Albany, N. Y. 977,193.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., Col. E. D. Meier; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

George Westinghouse, inventor of the air-brake, which practically revolutionized railroading, told the members of the American Society of Mechanical Engineers how he came to discover and perfect it.

It is a vivid picture for the man who says, "I've no time to read technical papers." It's a lesson which teaches that the *value* exists in technical papers, and whether it's extracted or not, depends wholly upon the reader.

How he conceived and developed the air-brake now in use on all railroads was told by Mr. Westinghouse in detail. Its conception, his story revealed, resulted first from an accident and second from the chance that led two young women to canvass his subscription to an engineering magazine, in which was recited the use of compressed air in the driving of Mount Cenis tunnel.

The accident was a collision of two freight trains between Schenectady and Troy in 1866, which delayed for two hours a train in which Mr. Westinghouse was a passenger. Had the freight trains been equipped with brakes of sufficient power, thought Mr. Westinghouse, the collision, with its subsequent loss of time to him, could have been avoided, and immediately he set to work to develop such a brake.

An inventor named Ambler already had perfected a chain brake which was operated by the revolving of a windlass in the engine, the chain thus being taken up and the brake levers of each car thus operated. To Mr. Westinghouse came the idea of exchanging the windlass for a cylinder beneath the locomotive, the piston of which should be of extraordinary length and connected with Mr. Ambler's chain so that the drawing in of the piston by the application of steam from the locomotive would give a more accurate control of the brakes than was possible with the windlass device.

But experiments showed quickly that the piston could not be made long enough to operate the chain on more than four or five cars, and Mr. Westinghouse overcame this difficulty by placing a cylinder beneath each car with a flexible pipe connecting each one to the locomotive for its supply of steam. This effort failed because, even in warm weather, it was found impossible to transmit the steam. Then the young ladies called.

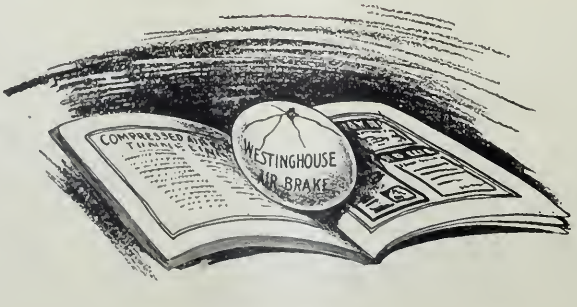
It was then that Mr. Westinghouse saw the article on compressed air, and in the use of this medium instantly saw a solution to his difficulties.

An apparatus employing compressed air instead of steam was built at once and a Steubenville, Ohio, accommodation train of the Panhandle railroad was equipped with the device. On its first test the engineer, as his train emerged from the tunnel near the Union Station in Pittsburg, saw a wagon on the tracks, and the value of the new brake was demonstrated instantly by the short distance in which the train was halted in time to avoid an accident.

Many experiments were made by various railroads, and in 1869 several of these had their running stock equipped with the new brake, but once more the inventor was to face and overcome an obstacle. It developed that it took too long to set and release the brakes, and that in the event of a break in the train the rear section would be without brakes. The first automatic air brake grew out of the need to provide against this contingency.

Of the lives which his brake may have saved in its prevention of railroad accidents, Mr. Westinghouse quoted a diplomat at the International Railway Congress in Washington in May, 1905, who declared that he felt safe in saying that "The air brake has saved more lives than any general ever lost in a great battle."

Mr. Westinghouse found the solution to his problem in the reading columns of an engineering paper.



Reading and advertising columns are twin brothers —the one tells "how," the other shows the devices which have been invented and marketed to help.

No progressive engineer can afford to overlook either any more than he can afford to cut out food and drink.

Power and The Engineer

Vol. 32 NEW YORK, DECEMBER 27, 1910 No. 52

WHEN a man is advanced from a subordinate position to one of trust and responsibility within a comparatively short time, public opinion usually attributes his success to either favoritism or luck without stopping to consider that perhaps he possesses special qualifications for the position.

Favoritism, although occasionally found in certain classes of service, is seldom practised in strictly commercial or engineering circles. Too much risk is involved in putting an incompetent man at the helm. Hence, with rare exceptions, the responsible position goes to the man who can produce the "goods."

It cannot be denied that the element of chance figures, to a certain degree, in shaping the careers of most people, yet it does not play so important a part as is generally supposed.

Those people who are born under a lucky star are nearly as scarce as hen's teeth.

At some time or another opportunity knocks at everyone's door, and indeed to many it is a frequent visitor, but whether it is taken advantage of depends entirely upon the individual concerned.

One of the surest ways to achieve success is to be prepared to sieze an opportunity when it is presented. It is too late to prepare after the opportunity has appeared, for in that case the chances are that another will get the job.

There are people who consider that if they do their daily routine work in a satisfactory manner, they are performing their duty. They may be doing their duty to their employers, but not to themselves. Time spent in picking up additional information pertinent to their work is not wasted, but is really invested at a high rate of interest.

An incident exemplifying this occurred some time ago at a large manufacturing plant in New England. One of the employees who held a subordinate position spent his spare time in picking up bits of information regarding the operation of departments other than that in which he was employed. He also became interested in electricity and when the plant finally changed over to electric drive, a slight accident gave him an opportunity to be of service. This brought him to the attention of those higher up and he was given a better position which afforded him further opportunity to become familiar with the operation of the plant.

His real opportunity came when the superintendent was suddenly stricken with paralysis. The general knowledge he had gained proved so useful at this time that a large part of the superintendent's duties fell to his lot. This gave him the chance he had been waiting for, and he immediately introduced a number of changes which decreased the cost of production. When the superintendent was permanently incapacitated for work by a second stroke, our friend was immediately chosen to fill his place.

All this happened within a little over three years, and although it is only one of many such instances, it serves to emphasize the fact that one should be always ready to meet the unexpected.

Repair of a Large Engine Bed

By F. W. Brady and C. J. Mason

Due to culm being carried over with the circulating water into the low-pressure cylinder, the guide barrel was split off from the rest of the frame. The repair was effected by cutting out the guide barrel and the attached portion of engine bed, and inserting a new section.

Some time ago in an issue of Power there was given a short account of an unusual engine wreck. The engine was a cross-compound of 2400 horsepower, driving a direct-connected street-railway generator, and the wreck was caused by coal being carried into the low-pressure cylinder by the cooling water of the barometric condenser. The water was raised by a centrifugal pump from a nearby river, the banks of which were lined by numerous culm piles, and each freshet carried quantities of the culm down stream. On this particular occasion the tail pipe of the condenser became clogged with culm; then the overflow, as the centrifugal pump continued operating, came over into the engine-exhaust pipe. The vacuum was lost and the dry-air pump, becoming choked with the culm, was stalled. It all happened so suddenly that those on watch had no idea as to what was taking place. However, they immediately proceeded to shut down the engine, and while so doing, the suction of the low-pressure piston drew a charge of water and culm into the cylinder. On the return stroke the piston struck the imprisoned mass, and the turning force of the flywheel and generator armature knocked the cylinder base from the bed. Fortunately, the cylinder itself escaped injury, but the 18-inch connecting piece between the cylinder and the bed was broken. This let the crank end of the cylinder drop, as it had no other support, resulting in a broken steam connection and a wrecked valve gear; the head end of the cylinder resting on two flanged feet clamped to the foundation capstone.

The most serious break, however, was that of the guide barrel, which had the top split off by the upward pressure of the connecting rod against the crosshead, which, in turn, bent the piston rod as well as the end of the connecting rod itself. The 3⅛-inch key in the crank disk was also partly sheared off.

The engine bed included the guide barrel in one solid casting 20 feet long; hence the breaking of the guide barrel made it necessary to procure a new bed. As it was specially desirable to continue the operation of the high-pressure side of the engine, and as this required the use of the pillow block of the main shaft on the broken bed, and also as a method of reducing the cost of making the repair, it was decided to cut out the 9-foot guide-barrel section.

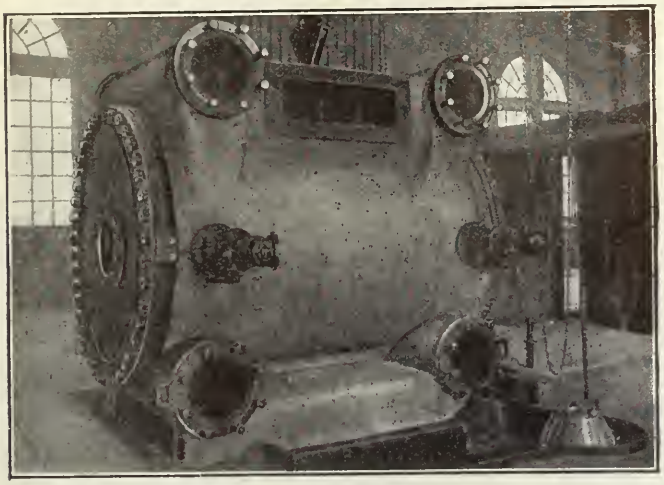


Fig. 1. Cylinder Stripped and Blocked

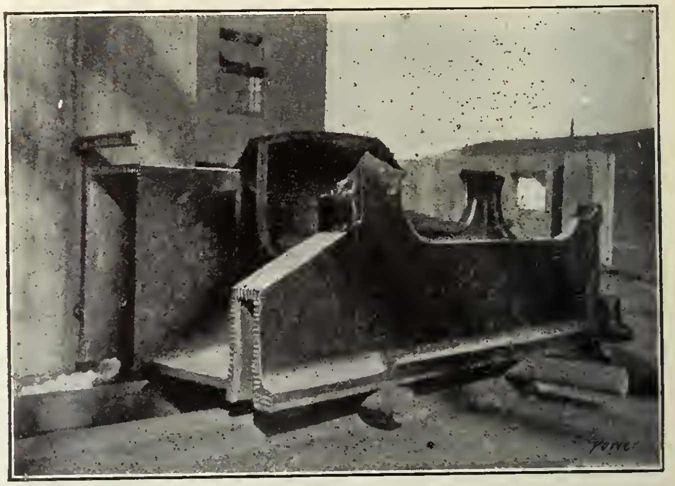


Fig. 2. Broken Guide Barrel Cut from Bed

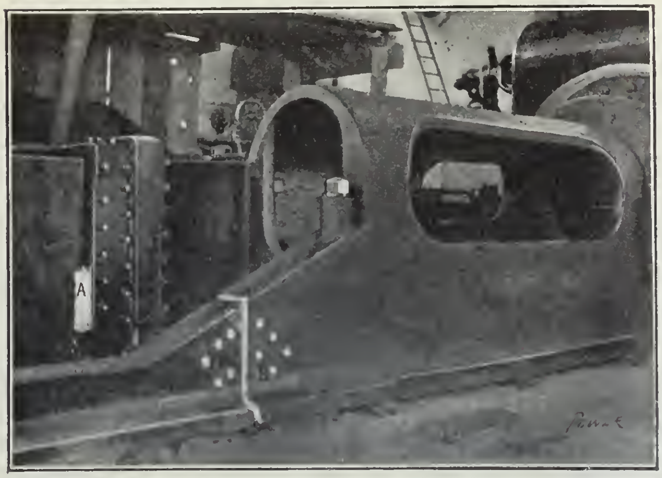


Fig. 3. New Guide Barrel in Place

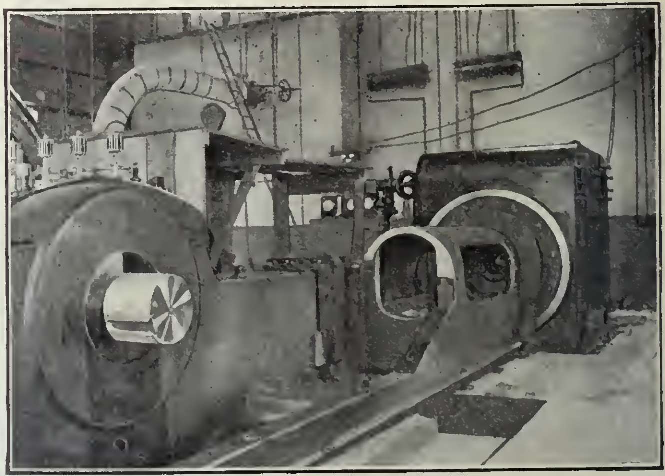


Fig. 4. Engine Before Receiving New Crank Disk

The cylinder was stripped and blocked as shown in Fig. 1, so that the work on the bed could be done as conveniently as possible. The 9-foot section, shown in Fig. 2, was then cut from the bed by drilling ¾-inch holes along the entire cutting line. It will be seen from the illustration that the bed where cut consists of two box sections connected at the bottom by a horizontal flange. The rear box is 40 inches high and 20 inches wide, the connecting flange 28 inches wide, and the front box about 10 inches wide and 20 inches high. The thickness of the metal ranges from 1⅜ to 1¾ inches and the weight of the section cut out was a little over four tons.

After removing the damaged section the concrete foundation was cut down two or three inches to give clearance. Pits were chiseled under the joints (one is shown in Fig. 4) and extended out into the floor so that the workmen could get under the bed. Fig. 3 shows the new piece set in place together with some details of the joint. The new guide-barrel casting is made with a 3-inch flange on both the rear box section and the bottom web. A similar flange having a 7-inch plate was clamped to the end of the pedestal casting by means of a double row of 1½-inch square-headed screw bolts and the two castings were held together by a single row of hexagon-headed screw bolts. Lead was run into the crevice between the bed and the plate on the flange, the dam at *A* being for that purpose.

The low portion of the box section at the front of the bed was fastened to the guide barrel by means of plates placed on the inside and held by screw bolts, five bolts being used through each end. A fitting strip, or liner *B*, about ⅞ inch thick, was used to close the space between the abutting surfaces. The screw bolts and liner were then dressed flush with the outer surface so that the bed might have a smooth surface finish when painted.

A housing of rough boards was erected over the foundation to protect the generator from the dust, etc., incident to the repair of the bed. A new crank disk was also necessary, for the original one had one-fourth of its circumference broken off and the 3⅛-inch key was sheared over about half way. The disk, which was 63 inches in diameter, was removed from the shaft by drilling a row of five holes and driving tapered pins into them until the disk opened up enough to permit its being taken off. The 10x10-inch crank pin was not injured at all.

About five months elapsed from the date of the accident until the engine was again put into service. To prevent a possible repetition of the accident a change in the condensing-water supply, so as to filter the water and remove all foreign material, was necessary.

A New Theory of Belt Driving*

By Selby Harr

Conclusions drawn from the results of over 1000 tests conducted by Professor Kammerer, of Charlottenburg, Germany.

The effective pull on a belt is not reduced so much by centrifugal force as the currently accepted method of calculation would lead one to expect. In other words, the maximum allowable tension in a belt apparently increases with the speed.

The coefficient of friction increases with the speed and reaches values far above that usually assumed.

The maximum efficiency of transmission is not limited by the speed.

The coefficient of friction is larger for large pulleys than for smaller ones.

A wooden pulley has a higher coefficient of friction than an iron one, at least when new.

Transmission is more efficient and the coefficient of friction is higher when the smaller pulley drives.

A belt operates with higher efficiency and usually with less creep when the tight side is on the bottom.

A properly placed idler improves the efficiency of transmission.

The neutral layer is symmetrically located in the belt so that the effective diameter is the sum of the pulley diameter and the thickness of the belt.

At any speed the sum of the initial tension and the centrifugal force is constant.

In concluding his paper, Mr. Haar made the following statement:

As a result of the modern experimental work on power transmission by leather belting, a great deal of light has been shed on the subject and the methods of treating it must be revised. It appears evident that the maximum allowable tension in a belt is not constant and neither is the belt velocity, even though the pulleys revolve at constant speed. The efficiency of a belt is shown to be quite high, just as high as a good gear transmission, the over-all efficiency of the belt transmission being considerably reduced by larger bearing friction losses. This shows that the improvements should come in the bearings, as by the use of ball or roller bearings, for example. The exceptional flexibility of a belt transmission has made it a favorite for many difficult problems. We shall close with the following remarkable—and, we are afraid, overenthusiastic—prophecy of Mr. Gehrckens:

Sixty meters per second (11,900 feet per minute) is no limit to belt speeds at present. Belt pulleys have run at 500 meters per second (98,000 feet per minute). Brown, Boveri & Co. have reached 375 meters per second (74,000 feet per minute); the Schuckert Company run flywheels at 100 meters per second (19,600 feet per minute) (and so do American manufacturers).* Air resistance is an important feature at such speeds. When air is excluded the resistance decreases to one-half, and in a vacuum it is almost nothing. * * * We are by no means close to the limit of belt speeds, and the capacity of the belt does not commence to increase until high speeds are reached. Steam-turbine speeds are so high that some reducing device must be provided and for this purpose the belt is the simplest and most practical connecting link. With transmissions at such high speeds the construction must be painstaking and the crowning of the pulley is an important feature.

The discussion following the presentation of the paper was in brief, as follows:

C. G Barth: I am pleased to see that Mr. Haar takes the view that we cannot hope to run belts at the tremendous velocities used in Germany. At high speeds the initial tensions would be so high as to overbalance the centrifugal forces so that high tensions could not be maintained very long.

G. M. Vanderhoff: The best belt drives are those which represent the wisest investment; therefore, all the factors entering into the investment must be considered. The narrowest belts consistent with the other factors means maximum economy. A mean between the high speed demonstrated in the German research and the moderate speeds suggested by American engineers would represent the best practice.

In designing belt drives it seems that some of the things to be kept in mind, in view of the papers presented, are: first, decreasing the slack-side tension by increasing the arc of contact; second, increasing the coefficient of friction. The arc of contact can often be increased to advantage by the use of large-diameter idlers. A high coefficient of friction can be obtained by the use of wooden pulleys. The statement that belts do not last as long on wooden pulleys as on iron pulleys does not apply to wooden pulleys with iron faces.

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*Abstract of paper and discussion presented before the American Society of Mechanical Engineers' annual meeting, New York City.

*Parenthetical expression is Mr. Haar's own comment.

It would seem from this paper that the ratio between the diameter of pulley and thickness of belt should be, for the best results, considerably greater than we have been inclined to think sufficient.

Harrington Emerson: In some respects I cannot help but differ from Mr. Haar's paper. There are two or three main facts about belts; one is that if there is a large initial stretch, owing to a high tension, much power will be lost. If the belt is stretched 15 per cent. the driving pulley will travel that much faster than the other. Also, if there is a high tension there will be a large arc of slip; and this wears out the belt.

The only legitimate wear of a belt is what occurs in the slip; except for that, a belt should last forty years.

Mr. Taylor: When anybody suggests a speed of 11,900 feet per minute for a belt, he is very near the limit. Experimental data which are the result of purely academic experiments, made on a machine especially made for the purpose, should not be followed blindly, unless backed up by experience. Five to six thousand feet per minute is a ruinous speed for belts and economy. It should neved be practised unless necessary, and ordinarily should not exceed four to five thousand feet per minute.

G. I. Rockwood: Mr. Taylor's remarks apply to small belts but when a belt is transmitting power from a 5000-horsepower engine it is different. For such practice 5000 feet per minute has been in use for years.

Burning No. 3 Buckwheat Coal

By Warren O. Rogers

This grade of coal can only be used when forced draft and shaking grates having small air spaces are employed. The fire must be handled in a certain way as all ordinary methods result in failure. The method used by one of the largest steam generating companies in the world may be of interest.

Burning their coal for them is what the New York Steam Company is doing for over thirteen hundred customers. In other words, this company, which is said to be the largest of its kind in the world, sells the product of its boilers direct to its customers unmodified by any question of engine and generator conversion.

In order to supply the demand for steam, three large steam-generating plants are maintained, two at the foot of East Fifty-ninth street, and one at 173 Washington street, New York City. The smaller uptown boiler house contains eight 650-horsepower Babcock & Wilcox boilers, a total of 5200 horsepower. The larger uptown plant contains 24 boilers with a capacity of 11,410 horsepower, or a total of 16,610 boiler horsepower for both plants. These boilers consist of three 270-horsepower Babcock & Wilcox, seven 1000- and two 750-horsepower Climax boilers and four 525-horsepower Aultman-Taylor boilers. The downtown plant contains fifty-six 250-horsepower Babcock & Wilcox boilers, eight 350-horsepower boilers of the same type, one 350-horsepower Parker down-draft boiler and one 1000-horsepower Climax boiler, a total of 18,150 boiler horsepower.

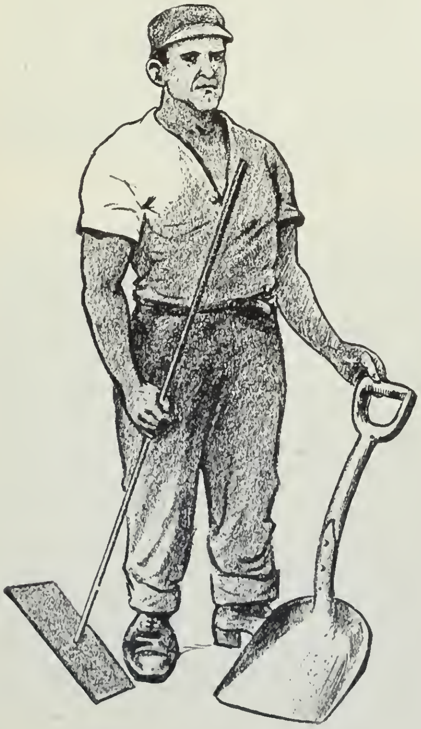

FIG. 1. THE FIREMAN AND HIS TOOLS

These boilers are mostly equipped with a grate designed by G. C. St. John, president of the company. They are of the shaking type and have a 47 per cent. air space. Fig. 4 shows the design. Forced draft is used under all boilers in the three plants with one exception, which boiler is run under natural draft.

No. 3 buckwheat coal is burned with an average draft of from 0.5 to 0.6 inch of water. No damper regulators are used on any of the boilers, the damper being manipulated by hand and kept from ¼ to $\frac{5}{16}$ open.

Eleven to 12 pounds of coal are burned per square foot of grate surface. One fireman attends to three of the 250-horsepower boilers; two of the 650-horsepower boilers are fired by one man and one Climax boiler by two men. The fires are carried at a thickness of from 3 to 12 inches, depending upon the length of time the fires have been run without cleaning, which duty is performed twice during each eight-hour shift. The fires are all carried at the same thickness, are kept level and the work of firing is practically a continuous performance.

During my visit to these boiler plants the following procedure was observed: The fireman opened a furnace door and at a glance noted the condition of the fires. The scoop shovel was then deftly handled, green fuel being thrown on the places where incandescent coal showed. The red spots in the fuel beds were not covered.

"Now watch that man," said the chief engineer of the plant, and I did. The shovel was thrust into the coal pile in front of the boiler, and, with a skill that was surprising, one-half of the amount picked up was distributed around one side of a red place, and, with a twist of the wrist and more skill, the remaining coal was distributed around the other side of the red coal. Wherever a red place showed, no coal was thrown. In this way the fuel was allowed to become incandescent and not permitted to work down into the ashes and finally be dumped into the ashpit to be carted away as refuse.

At intervals of about two hours a light hoe was used to level the fire and mix such green coal as had failed to properly ignite, with the burning fuel. Fig. 1 shows all the tools that it is necessary to use in the boiler room.

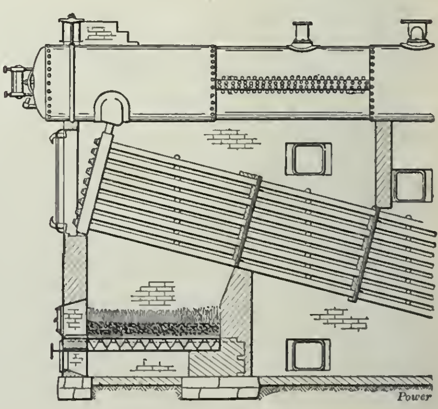

FIG. 2. PROPER CONDITION OF FIRE

"How are the fires cleaned?" I asked.

"Come down stairs," answered the chief. (The boilers of the downtown plant occupied four floors.) We went down to a lower floor where the men were busy cleaning fires, two at a time.

The beds of ashes and red-hot clinkers and coal were about 10 inches thick, but the proportion of coal was small. The fireman operated the shaking grates, until there was but 1 inch of ashes left on the grates. These were then smoothed off level with a light hoe, and about ½ inch of green fuel thrown on this bed. The furnace doors were closed and the forced draft turned on.

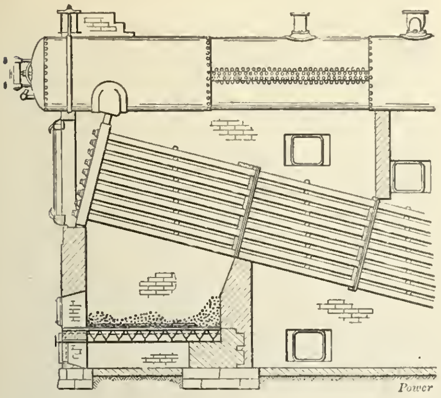

FIG. 3. BADLY KEPT FIRE

"I guess he has lost that fire," I remarked, as the fireman finished, for there was no sign of live coals after the green fuel had been thrown into the furnace, nothing but the glow of red-hot sides and bridgewall.

"In five minutes that furnace will have as pretty a fire as you want to see," replied the chief. As he spoke he opened the furnace door and sure enough little shafts of blue flame were seen shooting up through the black bed of coal. In about five minutes we returned to the boiler and, as predicted, the furnace contained a bright, hot, even fire.

"How will the fireman handle that fire for the next two hours or so?" I inquired.

"Watch him," was the reply.

I did so. The fireman took the shovel and, to me, accomplished but half a job. Instead of covering the entire surface of the burning coal with green fuel he scattered a little over the bright places in the rear next to the bridgewall, firing heaviest at the front, but heavy does not mean more than ½ inch thick.

"Why doesn't he spread the same thickness of coal over the entire grate surface?" I asked, while the inclination was strong to take the shovel and complete the work of properly (?) firing the boiler.

"It would be useless work," was the reply; "besides it would be a waste of coal. Owing to the pitch of the grates, the influence of the draft and the occasional leveling of the fires, the fuel works itself back to the rear of the furnace." Figs. 2 and 3 show proper and inproper conditions of the furnace.

"What kind of a slice bar is used?" was my next question, not noting any lying about.

"None is ever used; there isn't one in the plant," was the reply. "Couldn't find a quicker way of killing a fire or of wasting coal than by using one."

"Do you test your coal?" I next asked.

"Yes, a sample from each cargo is taken. For instance, about four cargoes are received each week. When the coal arrives one boiler is put on test and run for eight hours, a record being kept of the coal burned, the water evaporated and the ash content of the coal. More than that, the coal is graded; that is, 100 pounds of coal is taken from each cargo and sifted to determine the grades of coal contained in it. This sample is passed through screens, the first having a 3/16-inch mesh, which allows everything but No. 1 buckwheat to pass through to the second screen, which is of ⅛-inch mesh. This screen retains the No. 2 grade. The third screen has a mesh of 3/32 inch, which retains the No. 3 buckwheat coal, and all that passes through the screen is classed as No. 4 grade, which is practically dirt.

"Looks easy enough to burn barley coal," I remarked, as I made ready to depart, expecting an out and out denial.

"It is —— when you know how," replied the chief. "You have seen the whole process, and that is all there is to it. It comes hard on a new man, because he wants to throw a lot of coal up against the bridgewall, the same as you wanted to, and he wants to fire heavy and above all stir the fire. We do just the opposite here, and that is why we get good results in the three boiler plants."

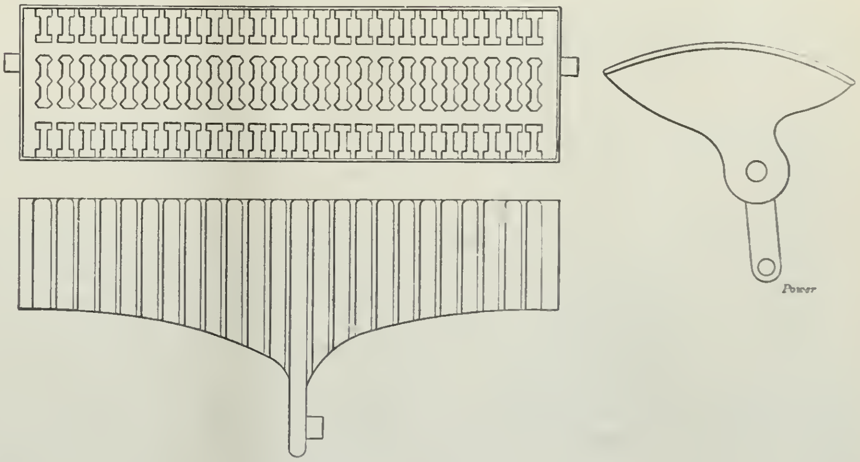

FIG. 4. DESIGN OF SHAKING GRATE

As I went away I summed up the situation in burning No. 3 buckwheat as follows:

Fire light and often.

Keep a forced draft of from 0.5 to 0.6 inch.

Keep the damper 5/16 open.

Never use a slice bar.

Level the fires about every two hours, or when necessary.

Never throw green fuel on other than incandescent fuel.

When cleaning fires keep one inch of ashes on the grates.

Always use a shaking grate.

Never handle the fire as other fuels are handled.

After noting the methods used in burning No. 3 buckwheat coal it would appear that a small steam plant containing one or two boilers would have trouble in running on this grade of fuel, because it would be difficult to force the fires in case a sudden demand was made for steam, and doubtless it would be hard to keep up steam when cleaning fires if the boilers were hard pressed.

The foregoing information was obtained through the courtesy of Fred A. Rankins, chief engineer of the downtown plant, and John Felder, chief engineer of the two uptown stations.

Pete Blowoff got drunk agin tother day an' when an' old mule kum 'round his ingin room an' begin t' brayin', Pete sed he gessed th' dumd masheen needed ile frum th' way it skreaked, so he went out an' pored 'bout a gallon uv ile in th' critter's ear; then he got 'round behind it an' got t' pumpin' its tail, thinkin' it wuz th' startin' bar uv his ingin; he started sumthin' all right, th' gol darnd critter landed on th' bib uv his overalls with both feet an' th' fust thing Pete sed when he'd kum to wuz t' ask ef th' 'sploshun wuz caused by low water. Th' docter told 'im thet it wuz caused by a combinashun uv low booze an' a couple uv jackasses.

The diameter in quarter-inches squared and divided by six equals the approximate weight, in pounds per foot, of a bar of iron.

Testing a Planimeter for Accuracy

By W. L. Durand

> *To determine the accuracy of a planimeter the user must ascertain whether the readings are uniform, whether the length of the tracing arm is accurate and whether the axis of the recording wheel is parallel. An outline of each test is presented.*

The planimeter is one of the most widely used of the measurement instruments employed by engineers, yet few understand the principles on which the planimeter is based or the methods employed to determine if a particular instrument in use is giving correct results.

The following are three tests that should be applied to all planimeters before complete reliance is placed on the readings they give:

Test for uniformity in the record-wheel readings.

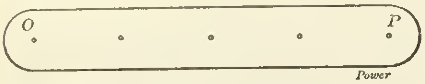

FIG. 1. THE CHECK RULE

Test for accuracy in the length of the tracing arm.

Test for parallelism of the recording-wheel axis.

To test for uniformity in the record-wheel readings a small check rule, as shown in Fig. 1, should be made of a thin strip of some metal such as brass. The rule is pivoted at *O* and the point of the planimeter is placed at one of the points on the rule, as shown in Fig. 2, and an area traced. Ten readings should be taken in this way, and then the tracings should be repeated in a negative direction, which should give the same readings as those in the positive direction. If the difference between the maximum and minimum result does not exceed 2 to 2.5 units of the vernier the planimeter may be considered to be correct in this respect. The errors are most apparent when the circle traced nearly coincides for some distance with the zero or base circle.

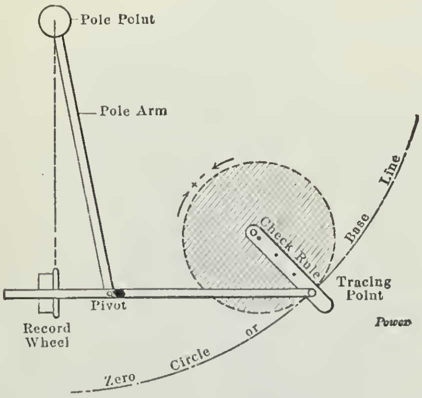

FIG. 2. TEST FOR UNIFORMITY IN RECORD-WHEEL READINGS

The zero circle is a circle around the pole point of the planimeter as its center and made by the tracing point so that the graduated wheel is continually traveling in the direction of its axis and consequently will not revolve. To find the zero circle, first a circle larger than the zero circle and then one smaller are drawn. The reading given by the planimeter for the larger circle is the area of the strip between the zero circle and the outer circle, while the area recorded for the small circle is the area between that circle and the zero circle. The area of the zero circle is equal to the true area of the small circle plus the strip between the small circle and the zero circle or to the true area of the large circle less the strip between the large circle and the zero circle. Hence, the area of the zero circle is equal to the true area of the small circle plus the recorded area of the small circle or to the true area of the larger circle less the recorded area of the large circle.

To obtain the best readings with any planimeter the zero circle should always be found and the area to be measured placed so that the circumference of zero circle runs through it, and the point of origin should be chosen on the zero circle so that a slight failure in returning to the origin will not be recorded by the wheel.

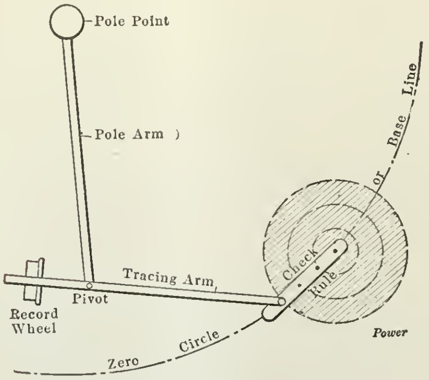

FIG. 3. TEST FOR ACCURACY OF LENGTH OF TRACING ARM

To test for the accuracy of the length of the tracing arm, trace by means of the check rule several circles of known areas a number of times, as shown in Fig. 3. Compare the mean of the results with the known area. If the results are too small by $\frac{1}{n}$th of the area the length of the tracing arm must be shortened $\frac{1}{n}$th of its length and *vice versa.*

To test for parallelism of the record-wheel axis use the check rule, as shown in Fig. 4, and make a series of tracings of a circle outside the zero circle and another series of a circle inside the zero circle, keeping the circles as far from the zero-circle circumference as possible. If the readings outside and inside are equal the axis is parallel, but if the area recorded outside is the larger the end of the axis nearest to the tracing point must be moved toward the outside and *vice versa.*

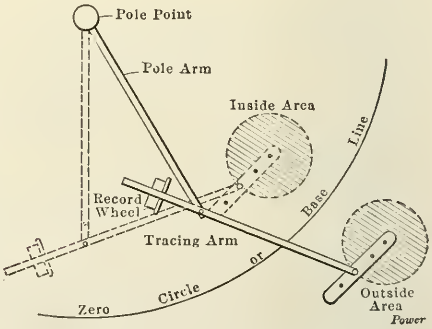

FIG. 4. TEST FOR PARALLELISM

According to the British Consul at Stockholm, the total water power in Sweden is estimated at approximately 10,000,000 horsepower, available during from six to nine months of the year, and 2,500,000 horsepower available during the time of low water. In the near future 600,000 horsepower will be in use, including 340,000 horsepower for the generation of electrical energy. At the present time the state possesses, or is part proprietor of, waterfalls of about 880,000 horsepower, of which 670,000 horsepower could be used without previous regulation of the respective waterfalls. Only 63,000 horsepower is now in use by the state, including some 40,000 horsepower at the power station at the Trollhattan falls. Work is now in progress at Trollhattan by which a further 40,000 horsepower may be used there. At the Porjus fall on the Lulea river in Lapland, a power station is to be erected with a capacity of 50,000 horsepower. In addition, the Swedish government proposes to set up works at Alfkarleby, in Upsala lan, Uppland, and other places.

Estimating Heating Requirements*

By Reginald P. Bolton

The usual method of estimating the amount of heating required for a given building, based on the areas of exterior surfaces and their relative heat transmitting values, with an additional allowance for cubical contents, is considered to be inexact because it does not take accurately into account loss due to infiltration. A method which does include this factor is presented and discussed.

The common method of relating the work or cost of heating to the cubic contents of a building, is one which is liable to be misleading, and may lead to over or under estimation from causes that will be evident. The actual work of heating is not varied by the fact that the interior of the building is of a certain size, but it is directly varied by the exposure of the exterior of the building to outside temperatures and to wind effects.

The usual methods of computation applied to new buildings are based upon the ascertainment of exterior surfaces and their relative heat-transmitting values, with an addition thereto of a certain fixed allowance for cubical contents, on the ground that these contents represent approximately the amount of air leaking into the building from the exterior in one hour—giving for one change of contents per hour:

$$\left(\frac{W}{4}+G+\frac{C}{55}\right)t = \textit{heat losses in units}$$

where,

$W =$ Wall surface exposed;
$G =$ Glass surface exposed;
$C =$ Cubic contents of interior space to be heated;
$t =$ Temperature difference between exterior and interior.

While the losses of heat from the exterior of a building are by this method fairly ascertainable, the loss by infiltration is empirical, and this item becomes of considerably increased importance as buildings are increased in hight. Methods of building construction, while they have been largely improved in a general way, have not relatively been improved in the matter of the construction and setting of window frames, and in high and unprotected buildings the loss of heat due to this cause is quite a substantial element in computations of this nature. The increase of wind velocity and corresponding pressure due to hight is shown in Fig. 1, from which it will be seen that any opening or leakage point situated at a considerable hight above the ground must be regarded as of greater extent or capacity for leakage than one which is below the average level of buildings in the immediate neighborhood.

Another element which contributes to this result is the funnel or chimney effect of tall buildings, which establishes, under certain conditions of wind, a strong draft through the corridors and elevator shafts, thereby drawing away heat from the lower portion of the building. On account of this effect it has been found necessary in certain tall buildings to add heating surface to the lower floors, in excess of computed requirements.

The following theoretical method, which has been followed by the writer, of ascertaining the heating work for any building has given very close approximations to actual results.

An examination of the recorded variations of temperature for thirty years past has afforded the means of plotting the average hourly variations for each month of the heating season, bringing out the fact that the night temperatures are generally lower than the day temperatures, and that the work required for heating is always relatively lower at the period of maximum day work.

The diagram in Fig. 2 is used for the ascertainment of these average hourly conditions, the temperature curves being inverted and becoming a relative scale of boiler horsepower, so that the average proportion of heating required at any particular part of the day in any month of the season can be rapidly ascertained. These curves are based upon the recorded conditions of the heating season of 1903-4, which were found to

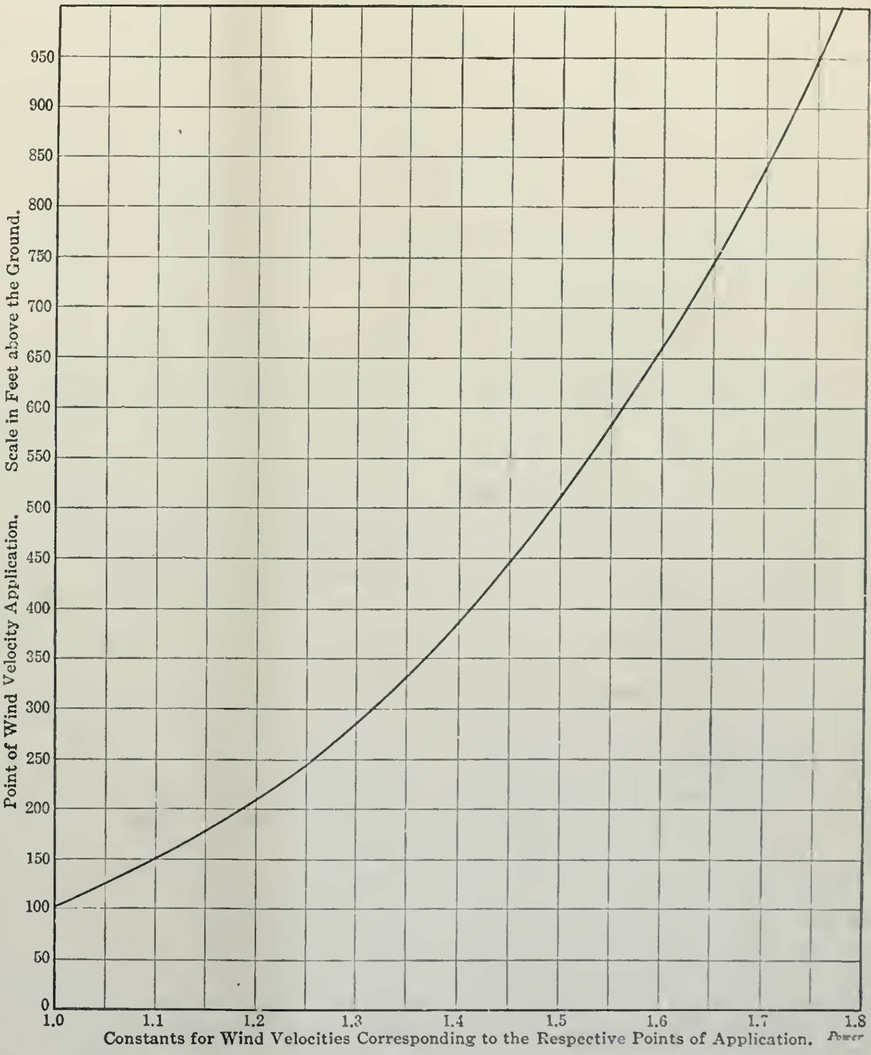

Fig. 1. Increase in Effect of Wind Velocity Due to Hight

be very close to the general average for thirty years prior to the year 1907.

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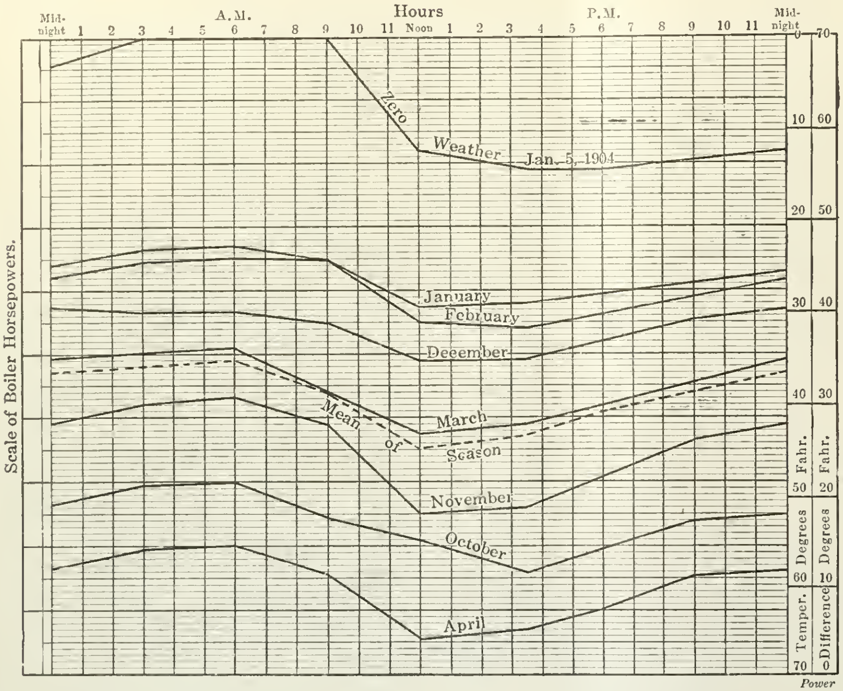
*Excerpt from "The Heating Problem in Its Relation to Central Station Lighting and Power Service," a paper read at the annual meeting of the Association of Edison Illuminating Companies."

It will be seen that the maximum observed heating work, which is that at the top of the diagram, on the rare occurrence of a zero temperature, falls off after the day hours, and, as has been already noted, on only about eight days of the heating season, or less than 4 per cent. of the heating period is there any temperature approaching the neighborhood of zero Fahrenheit.

In times gone by, a temperature of zero was more frequently recorded, and as it is frequently reached in the vicinity of New York City, it seems probable that the general temperature of the borough of Manhattan has been raised by the heat emitted from the numerous heating plants in the closely built-up center of the city.



Division of the Heating Season, New York City.

Month	Days	Holi-days	Sun-days	Work-days
October	16	—	2.20	13.78
November	30	2	4.20	23.70
December	31	1	4.40	25.57
January	31	1	4.40	25.57
February	28.25	2	4.00	22.25
March	31	—	4.40	26.57
April	15	—	2.14	12.80
Totals	182.25	6	25.92	150.30

EXPLANATION.

The curves are the averages of the hourly variations of temperature and represent conditions of the Season of 1903 - 1904 which was a representative Heating Season closely following the average of thirty years observations in New York City.

By ascertaining the maximum heating requirements, at Zero, of any building, the vertical scale at the left can be used for Boiler horsepowers, from which the average requirments during any month, and for any hour of day or night can be read off on the scale.

FIG. 2. HOURLY VARIATIONS OF HEATING WORK FOR NEW YORK CITY AND VICINITY

The effect of wind in increasing the results of low temperature has been made the subject of separate study. Close observation has been made of the variations of wind during the heating season, derived from the invaluable hourly records maintained for many years past by the Meteorological Bureau of the Department of Parks of the Borough of Manhattan, under the care of Dr. Daniel Draper. Combinations of high velocity and low temperature have been searched for over the records of many years and the most severe heating combination found has been that of a wind velocity of 42 feet per second accompanied by a temperature of 20 degrees above zero, which combination occurred on January 1, 1904, and would represent for an exposed and lofty building a condition somewhat in excess of zero temperature in a still atmosphere. The combination is, therefore, adopted of a zero temperature with this wind speed in order to arrive at the suitable amount of heating surface on exposed windowed sides of any building.

The average wind movement for a month may then be utilized with effect in a computation of heating work, notwithstanding the variations of movement and direction, and the system of proportioning heating surface to meet wind effect from any point of the compass eliminates dependence upon common methods of adding empirical amounts, such as 10 to 35 per cent., to heating surfaces on the north or other side of a building.

The effect of wind is mainly found to be in the leakage of cold air around the window framing, sashes and meeting rails, an effect increasing with hight from the surface or general level of other protecting structures. The use of close metallic weather stripping to reduce leakage around window sashes has been the subject of much discussion, and of some investigation, and in the recent experiments of H. W. Whitten, the writer had the interest of taking some part. By mounting a small room on a pivot upon the roof of the West Street building, the opening in which room was alternately provided with various forms of window construction, it was possible to test the amount of air passing through the sash at various speeds and pressures of wind. The general result of this and other investigations on the subject is to indicate that the excellence or poverty of construction of window frame and sashes may affect the total heating work to be done in a building by as much as 15 per cent. but that the natural ventilation of occupied spaces may be almost entirely precluded by close metallic weather stripping of window sashes.

From these observations the conclusion is derived that the work of heating buildings closely follows variations of temperature, modified by the variations in wind velocity, the effect of this latter being dependent upon the number of windows and their construction, but being in all cases in the direction of an increase of the heat to be provided. This is found to be the case not only in a single exposed building, but in a large number of buildings grouped together. Thus, the Crawfordsville, Ind., district (hot-water) heating plant, has an established practice of a reduction of the temperature of the hot-water supply of one degree per degree rise of temperature above zero, of an addition of two degrees per one degree fall below zero, but these modifications are varied by the addition of one degree for each mile per hour of wind velocity.

The district-heating system of the Columbus Railway and Light Company also follows the same general practice with success, varying the amount of hot water pumped through the mains, as well as the initial temperature, in order to meet rise and fall in temperature with modifications to meet the velocity of the wind.

The study of wind effect has indicated that the heat transmitted through the building materials, due to the action of convection, is not materially enhanced by the movement of the air, because the action of convection in still air is always to establish a vertical current of heated air rising around the building, and the action of wind merely translates this vertical motion into a horizontal movement, at the same time creating on the lee side or sides a negative condition, reducing the motion due to convection and stagnating its action.

The method of defining the heat losses of any building which combines the elements of temperature and wind infiltration is therefore, for zero Fahrenheit exterior and 70 degrees inside temperature:

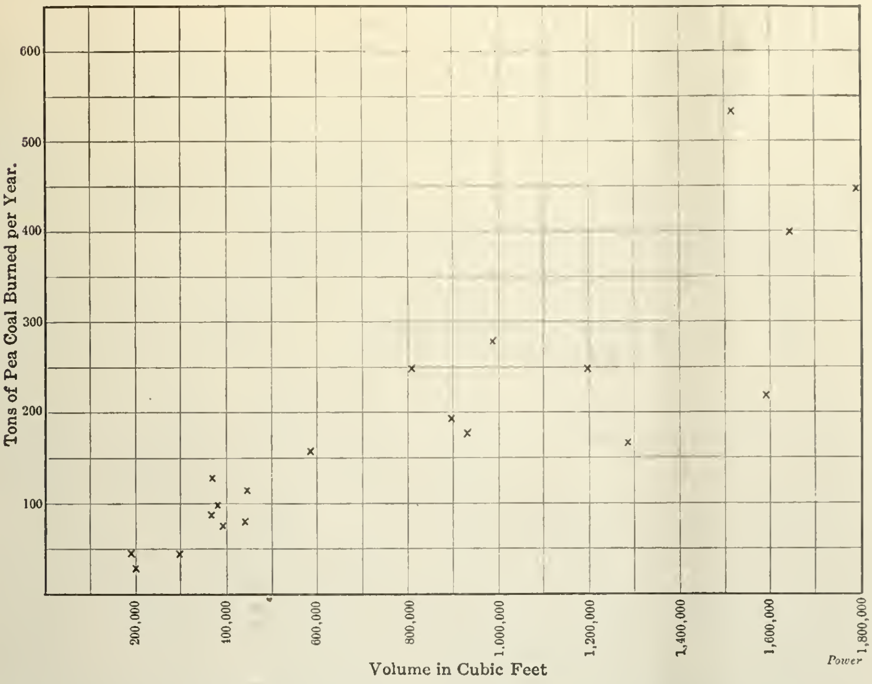


FIG. 3. RELATION BETWEEN COAL CONSUMPTION AND CUBICAL CONTENTS

(1) Transmission losses.

W = Exterior exposed wall surface in square feet × 21 = heat units per hour;

G = Area in square feet of window openings × 70 = heat units per hour.

(2) Leakage to be heated from all sides of building.

L = Perimeter of window openings in feet for ordinary construction × 72 = heat units per hour. Perimeter of window openings in feet for metallic stripped sashes × 31 = heat units per hour.

The sum of all in heat units, plus 15 per cent. for line and pipe losses, divided by 30,000 equals the boiler horsepower required per hour to meet the most extreme weather conditions.

This total capacity placed on top of the diagram in Fig. 2 gives a scale by means of which all monthly hourly averages and totals may be ascertained.

As the lineal feet in the perimeters of the window openings plus the meeting rail is, in standard types of windows, equal to the amount of the area of the opening in square feet, the computation of wind leakage may be simplified by treating L as G, reducing the computation to

$$G \times 70 + G \times 72.$$

The methods above outlined have been interestingly confirmed by observations conducted by E. F. Tweedy, of the New York Edison Company, the results of whose investigations are shown in Figs. 3 and 4.

In Fig. 3 are plotted the actual consumptions of coal of a number of buildings related to a basis of their cubical contents. It will be observed that the plottings show no definite relation to one another, and that the rate of coal consumption per season per 1000 cubic feet appears to be the same in the case of buildings widely different in total cubical contents.

The same cases show a harmonious relation when plotted as in Fig. 4 on a basis which takes into consideration the exposures of the buildings, with due allowance for air leakage.

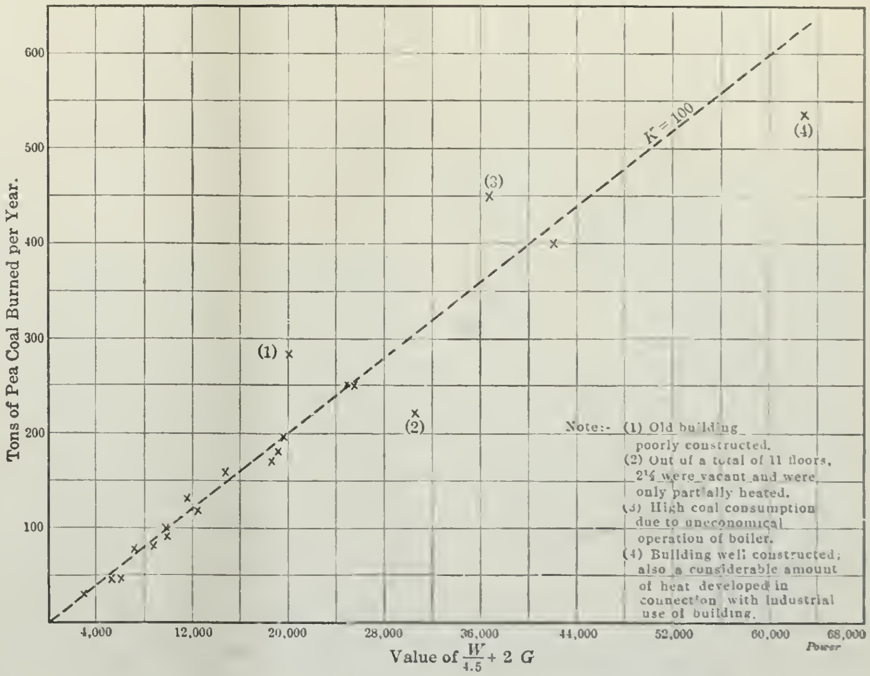


FIG. 4. RELATION BETWEEN COAL CONSUMPTION AND AREA OF EXPOSED SURFACES

The observations and deductions therefrom are, in part, thus described by their author:

At the outset, it was necessary to adopt an average ratio for the rate of heat transmission of glass to that of wall surface, and to obtain this it was, of course, necessary to adopt some thickness of wall as representing a fair average for loft and office-building construction in this city. Without entering into the details which led up to its adoption, I will say that a ratio of 4½ to 1 was finally adopted.

Realizing the uncertainty which arises in providing for the leakage loss on the basis of assuming an arbitrary number of air changes per hour, I decided to combine this loss with the transmission losses. Such a combination was deemed possible by the following line of reasoning: the leakage loss being proportional to the summation of the window perimeters (assuming the leakage space to be approximately constant in width) and the sizes of windows in similar types of buildings being fairly uniform, it becomes possible to consider the leakage loss as a function of the total glass surface without any very appreciable error.

Having made the foregoing assumptions, the following equation is obtained:

$$T = K\left(\frac{W}{4.5} + G\right) + K'G$$

where,

T = Tons of coal consumed per year;
W = Total exposed wall surface in square feet;
G = Total glass surface in square feet;
K and K' = Constants.

As it is not necessary to solve the above equation, K and K' need not be actually determined; for purposes of plotting, however, a relation between K and K' must be obtained. For $K = K'$ the equation evidently becomes

$$T = K\left(\frac{W}{4.5} + 2G\right)$$

The relation between K and K' can be varied considerably without affecting the results to any appreciable extent, other than to alter the value of K. For $K = K'$, $K = 100$, which is a very convenient quantity for practical use. Fig. 4 shows the data for 20 office and loft buildings plotted from the equation in the above form. It will be seen that the values conform very closely to the equation as given. It will be observed that pea coal was burned in all of the buildings cited. Data were secured, however, from a number of buildings where the buckwheat coals or various mixtures were being used. All such cases—as would naturally be expected, due to the lower calorific value of such fuels—fall above the pea-coal line, but in most cases to a considerably greater extent than is to be explained by the difference in the thermal value of the fuels. I think that this can be accounted for by the fact that these poorer coals are very rarely burned under proper conditions as regards type of grate and draft, thereby nullifying the saving which might otherwise result from the use of a cheaper fuel.

For apartment-house buildings, the same general relation has been found to hold good, the only difference being that the value of the constant K is less, being about 60 instead of 100. This can doubtless be explained by the generally poorer construction of such buildings and by the fact that heating is required for more hours during the day.

Control of Condensing Water*

By B. Viola

In condensers of the jet type the efficiency depends upon the degree of subdivision of the cooling water, the quantity supplied and the length of the time of contact between the incoming steam and gases and the cooling water. This paper illustrates a design of condenser in which the first and last factors are maximum and a device for automatically regulating the cooling water supply so that the temperature of the discharge remains very nearly constant.

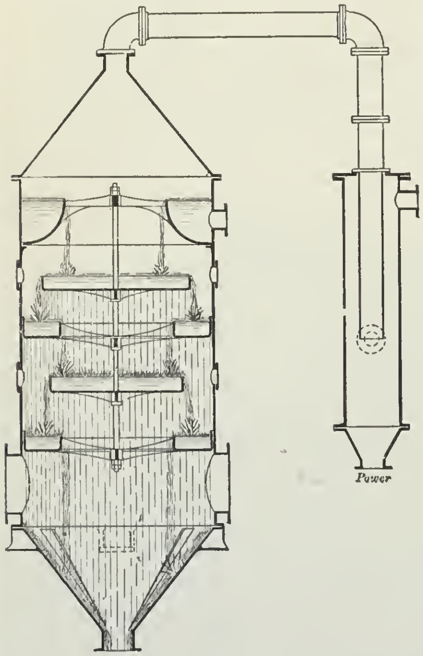


Fig. 1. Concentric Countercurrent Condenser

In the operation of the jet types of condenser it is often found that the injection of cooling water is unnecessarily increased at times by incorrect manipulation. Besides other visible indications of this mistake the temperature of the condensed water shows it most strikingly. When too much water is used the work of the air pump is made more difficult because of the air liberated from the water and if the pump is accurately proportioned for normal operation it is unable to accomplish the extra work. If the pump is large enough, the work is accomplished, but at the expense of additional steam. If a wet-air pump is employed it is called upon to remove an excess of both air and water. Moreover, the excessive injection of water causes a higher pressure in the condenser. Assuming the falling water to be a liquid cylinder, the outer surface of this cylinder upon entering the hot steam takes up the heat of the steam, while the interior parts remain cool. If, owing to the great velocity of the fall, the water column is unable to effect cooling by the interior portions of the column as well as by the exterior surface, the heat of the discharged water will be too low. But if the equalization of temperature can be brought about through the whole water cylinder during the fall the cooling power of the water can be utilized more completely.

The more finely the water can be divided in falling the more readily will the gases pass through it and the resistance will be diminished as the surface is increased. The division of the water brings about to a great extent the escape of the air contained in the cooling water. This division, however, is never fine enough; hence, in countercurrent condensers the real counterflow takes place only upon the surface of contact between the water and steam.

The proper utilization of the cooling surface can only be provided for by a proper arrangement of the time of contact.

Theoretically, the time increases as the square root of the hight of the fall, but in the condenser the resistance increases with this hight.

The velocity of the water is governed by the hight of the fall and it is advisable to restrict this hight in order to prevent frictional resistance, if for no other reason. The smallest value of frictional resistance is obtained when steam and gases flow over standing water. Plants of such construction, however, would be too large. Moreover, water and gases being poor heat conductors when in a state of rest, it is necessary to provide for a certain degree of motion for water in a condenser.

Further, the quantity of steam condensed per unit of time is directly proportional to the cooling surface and a division of the cooling water is therefore equivalent to an extension of the cooling surface.

Therefore, to heat the condensing water quickly to the necessary temperature by direct contact with the steam, it is necessary (*a*) that the surface of the cooling water be large, (*b*) that the cooling surface change quickly, and (*c*) that the time of contact between steam and water be as long as possible.

To fulfil these conditions a concentric countercurrent condenser such as that shown in Fig. 1 gives the best results. This condenser is designed for use in connection with two or more vacuum pans. By the addition of basins or trays the duration of the contact between water and steam is raised, resulting in an economy of water. Some water will accumulate in each of these trays temporarily and the temperature is therefore equalized at each stage. The fresh supply of water will sink to the bottom of the tray, forcing the lower stratum upward on account of the difference in temperature, and the warmer water will flow over from one tray to the next. The trays in this manner form zones with the desired small difference of temperature and constitute a graduated scale from the temperature of the steam admission to that of the gas outlet. The surface of condensation depends primarily on the size of the cooling-water cylinder when the depth of the trays is greater than zero. It may even be said that these moving water columns promote the cooling and air and gases will pass at the surface of the trays even though they do not pass at the bottom. During the time the water remains in the trays it has a chance to take up more heat, which is withdrawn from the steam. It is very important that enough water be permitted to flow to secure good results, but by providing the trays with spray holes, the cooling effect of the water is considerably increased.

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*Abstract of paper read before the American Society of Mechanical Engineers, New York, December 6-9.

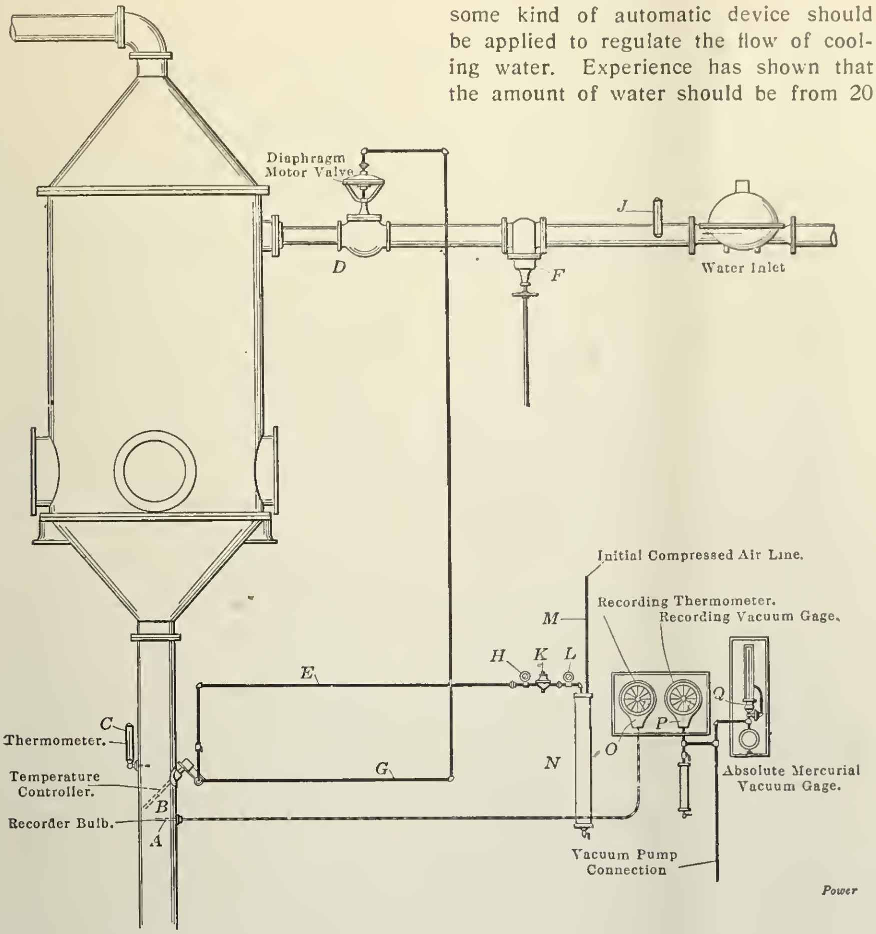


Fig. 2. Diagram of Automatic Condensing Water Control

The work of the condenser varies with the temperature, but in general it performs about 610 times the work of the air pump, volumetrically speaking. It is absolutely necessary to exhaust the air and gases to promote condensation, but the best air pump cannot do good work with a poor condenser, while a good condenser accomplishes at least fair results with an indifferent air pump.

Advantage of Automatic Regulation

It is very important that the right quantity of water be used to condense a certain quantity of steam. This matter is neglected in most condensing plants and some kind of automatic device should be applied to regulate the flow of cooling water. Experience has shown that the amount of water should be from 20 to 40 times that of the steam to be condensed. This is a very wide limit and in large condensing plants where immense quantities of cooling water are required some controlling device should be applied from an economical standpoint, so that water will not be wasted.

The first step is to decide at what temperature the water should be discharged from the condenser. This, of course, bears a certain relation to the steam to be condensed. I first applied a recording thermometer in the tail pipe of a concentric barometric condenser for the purpose of controlling the cooling water by hand operation of the inlet valve; but it was very difficult to convince the operator of the condenser that it was not necessary to let the cooling water run off cold, because it is generally believed the colder the water the higher the vacuum.

After experimenting for some time it was found very difficult to secure good results by hand regulation because of the continuous changes which must be made to meet conditions as the number of pans connected to the condenser is increased or decreased. This led to the application of an automatic temperature controller in the tail pipe connected to the compressed-air inlet valve, by which it was found possible to feed the condenser approximately in proportion to the steam or vapor which had to be condensed, resulting in a better average temperature of the condensed water and only slight variations.

By maintaining a nearly equal temperature of the condensed water, a very uniform vacuum was obtained except at periods when a new vacuum pan was charged.

Construction of Automatic Controler

The construction and operation of the automatic controlling device is shown in Fig. 2. The stem *B* expands and contracts in response to the temperature changes of the condensed water. The pipe *E* supplies compressed air of about 15 pounds pressure to the valve mechanism of the controller, from which it passes through pipe *G* to the water-inlet valve, where it actuates a regular globe valve *D* by means of a diaphragm motor which replaces the ordinary handwheel. The temperature of the condensed water affecting the stem of the controller permits all, a part or none of the compressed air to pass to the diaphragm-motor valve and consequently the latter is very nearly closed, partly open or wide open, according to the conditions prevailing, allowing the proper flow of water to the condenser.

Assume that the vapor pipe has been delivering an amount of vapor to the condenser which requires such an amount of water as will flow through the inlet valve when the latter is exactly half open. Should more vapor enter the condenser the temperature of the water passing through the tail pipe would, of course, start to rise, but as soon as the con-

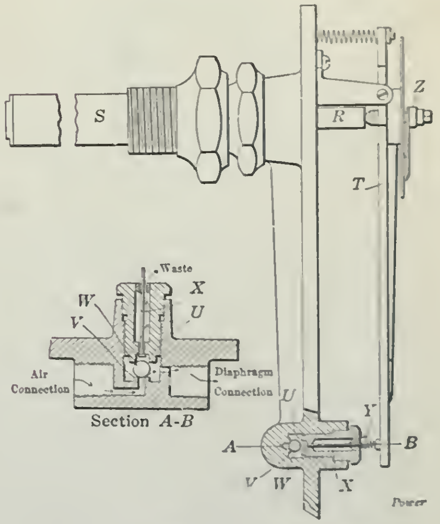


Fig. 3. Construction of Automatic Controller

troller stem was affected by this higher temperature it would expand and cause the valve mechanism of the controller to allow less air to flow to the water-inlet valve. Consequently, this valve would act to allow more water to pass through and thus maintain the proper proportion between the volumes of vapor and cooling water. The auxiliaries are; *J*, a thermometer to indicate the temperature of the cooling water; *F*, a hand valve for emergency use; *C*, a thermometer to indicate the temperature of the water passing through the tail pipe and serving as a check on the recording thermometer, which has its bulb at *A* and its dial at *O*, and *P*, a recording vacuum gage, checked by a mercurial absolute-pressure gage, *Q*. The supply of compressed air in *M*, which ultimately operates the diaphragm-motor valve on the water inlet, enters the trap *N*, where any moisture or sediment is separated and which also acts as a storage reservoir. The gage *L* shows the initial pressure of the compressed-air supply, which is reduced by passing through the reducing valve *K* to the pressure required for the controller, as shown by the gage *H*.

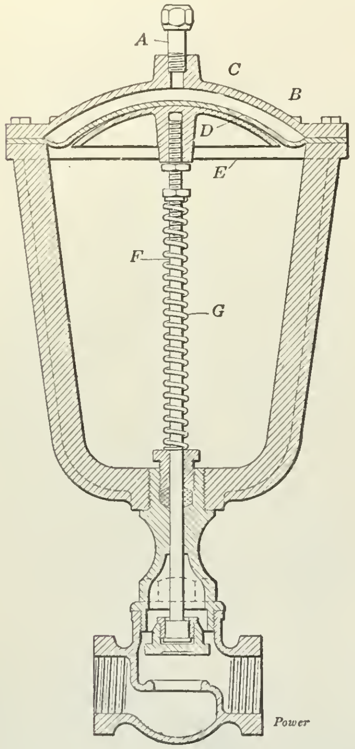


FIG. 4. SECTION THROUGH DIAPHRAGM MOTOR VALVE

Fig. 3 shows the very simple construction of the controller. The steam *S* is in two parts, the outer brass tube and the inner rod. The latter is composed of a material which has an extremely low ratio of expansion and therefore retains an almost constant length, regardless of temperature, while the outer tube, which is of brass, elongates or contracts in response to temperature changes. The tube and rod are so arranged that they are always in intimate though frictionless contact at the extreme end. At *R* is shown the extension of the inner nonexpanding member, which recedes when the outer tube elongates and advances when the outer tube contracts.

The movement thus obtained at *R* is transmitted by the lever *T* to the valve mechanism shown in cross-section. This is simply a three-port valve, shown more clearly in the section *A—B*, all ports of which are controlled by the ball *W*. The first port *V*, is the inlet for the compressed-air supply; another port, the second, connects with the line leading to the diaphragm-motor valve; the third port is the outlet for the air which has served its purpose by actuating the diaphragm motor. The position of the ball *W* depends upon the stem *X*, which is in contact with the lever *T* at *Y*. When the extension *R* moves outward due to the drop in temperature at *S*, the lever *T* is raised, and the stem *X* allows the ball *W* to leave the port *V* where the compressed air enters. The air can then flow through the second port to the diaphragm-motor valve, which starts to close. If a higher temperature acts upon *S*, expanding the outer tube and turning extension *R* inward, the lever *T* forces stem *X* against ball *W*. The latter then closes the port *V*, shuts off the air supply, and opens the third port so that the air can escape from the diaphragm motor. The diaphragm-motor valve consequently begins to open.

In actual practice, the diaphragm-motor valve is only completely closed or wide open if the conditions are such that the tendency is toward a sudden change in temperature. The controller is so sensitive that the diaphragm-motor valve is constantly throttling; that is, letting in just a little more or a little less water as may be required properly to effect the condensation.

If it is desired to change the setting of the controller, to maintain a higher or a lower temperature, it is necessary simply to apply a key to the part *Z* and by turning it one way or the other require extension *R* to travel more or less before the controller operates.

Fig. 4 shows the diaphragm-motor valve in section. This valve has a regular globe body, with a seat and disk arranged in the usual manner, but with a sliding stem that is operated by the diaphragm motor. The compressed air enters chamber *C* at opening *A*. This chamber is composed of a cast-iron top *B* and a rubber bottom *D*, called the diaphragm. The air pressure forces the diaphragm downward and the saucer *E* thus forces the stem *F* downward while compressing the spring *G*. When the air pressure is relieved the spring opens the valve again.

This installation has been the means of saving about $400 monthly for over a year in the charges for city water which was formerly wasted.

The question arises as to whether, with an equipment as described above, it is possible to attain a cooling-water rate near enough to that called for by theoretical formulas to be economical. Records of the water consumption as determined by meter readings show results very close to the theoretical and indicates great savings over the water required with former conditions of operation.

In sugar refineries, combined surface and spray condensers are usually employed. The vapor from the vacuum pan passes the surface condenser first for the reason that in case the vapor carries along some sugar particles, they will be condensed at that point and can be used to dissolve the raw sugar. In this way nothing is lost. The remainder of the vapor passes to the spray condenser, is condensed and flows down the tail pipe.

Condensers of this type require a considerable quantity of water and permit of great waste when no controlling device is employed.

Even where salt water is used waste should not be permitted because it increases materially the quantity of steam necessary for the operation of the pumps. In many cases where salt water is now used on account of the great quantity required, the use of a well constructed countercurrent condenser operated economically with a precise temperature controller may make possible the use of sweet water. This is a decided advantage where the condensed water can be used for boiler-feed purposes. I would not recommend its use in the same condition as when it flows from the tail pipe, in spite of the assertion that condensed water does not contain any impurities carried over from the source of the vapor. In steam-engine plants, for instance, it is well known that condensed steam contains oil, etc., and it is never pumped direct to the boilers. It has been determined from frequent analyses that condensed water from vacuum plants does not contain sugar or acid from the evaporation of acid liquor, but the so called untraceable losses in vacuum plants must necessarily be looked for in the condensed water. The quantity of water used is so great that the presence of these impurities is not noticeable, but practical experience has proved them to be present. However, when this condensed water at a high temperature (110 to 120 degrees) is passed through an efficient water-purifying plant, there is no doubt that it may safely be used for boiler feed.

When a gasolene blow-torch burner gets plugged up with dirt from dirty gasolene, it can be cleared by first removing the filling plug, then using a bicycle pump and forcing air into the burner, thus forcing the dirt back into the tank, from which it can be washed out.

Clearance and Compression

By Prof. R. C. Heck

> *A reply to Professor Dwelshauvers-Dery's criticism of the writer's original article appearing in the issue of September 13.*

Because of pressure of work along other lines, the words of Professor Dwelshauvers-Dery upon the subject of compression in the issue of September 27 have just come to my attention, nearly two months after their appearance. Such complete misunderstanding is shown of the purpose, and in lesser degree of the substance, of my contribution in the number for September 13 that it seems desirable to clear up some of the points involved.

My discussion was intended to cover, in a brief and very general way, the whole question of compression and its effects; not merely the particular body of information furnished by the experiments upon the engine at Liège. The view taken of the latter would have been more clearly set forth by the statement—which formed the closing sentence of my introductory paragraph, but which was unfortunately subjected to editorial excision—that "the harmful influences here shown in such full play exist in all engines, but their effects are generally of a much lower order of magnitude." Even without this distinct disclaimer, there is no shadow of the contention, as regards the phenomena of heat interchange between the wall and the steam, that for small engines the laws differ from those observed in large engines (page 1740, column 1). Differences among engines exist, not in the laws of the phenomena but in the amount of their net effect.

USE OF THE EQUILATERAL HYPERBOLA

Boyle's or Mariotte's law for the isothermal expansion of air, or of any practically perfect gas, is represented by the equation

$$P V = C;$$

but when this equilateral hyperbola is used as a steam curve it is not Boyle's law, and has absolutely no physical relation to that law or its conditions. In much of the traditional theory, the curve $P V = C$ is called the "theoretical curve of expansion" for the ordinary engine with saturated steam; this is a misnomer, because there is no rational theory of the action of the steam in the cylinder, whether when expanding or being compressed, and no curve can claim that title. Its appearance on page 1658, near the end of the last column, is another emendation.

The generally prevalent use of the hyperbola, as a standard of comparison for actual steam curves and in laying out preliminary diagrams, is fully justified and accounted for by the broad fact that in most engines the curves drawn by the indicator come close to this form. There are, of course, wide departures from constancy in the value of the product $P V$, under special conditions. As an extreme example, in the *Transactions* of the American Society of Mechanical Engineers, volume 10, page 751, is shown a diagram in which $P V$ increases by 75 per cent. from cutoff to release. It was taken from a 17x30-inch engine, unjacketed and noncondensing, with steam at 33 pounds absolute, cutoff at 0.16 of the stroke, and a speed of only nine revolutions per minute; but at normal speed the expansion curve came back into close agreement with the hyperbola. On the other hand, in the *Transactions* of the American Society of Mechanical Engineers, volume 25, page 273, are given diagrams from an engine using a high degree of superheat which show a decrease of 25 per cent. in the value of $P V$ from cutoff to release in the high-pressure cylinder; but here again, there was fair agreement with the hyperbola when the engine received saturated steam. The Dwelshauvers-Dery results constitute an extreme case for the compression curve, showing a shrinkage in $P V$ which is far in excess of what is usual, even in small engines—compare Nos. 16, 35, 41, 42 and 54 in Barrus' "Engine Tests."

THE FIRST ASSUMPTION

Actual steam curves of expansion and of compression represent the resultant effect of very complex thermal interactions, complex, especially from the point of view of a rational, quantitative and perhaps synthetic theory. Indeed, it is doubtful whether such a theory will ever be developed much beyond the present rudimentary and almost wholly qualitative stage. In face of this condition, the use of a simple, average result like $P V = C$, as a first approximation and as the basis for some easy deductions, is very convenient and entirely proper. Relations in terms of this assumption can readily be modified so as to take account of closer empirical information.

In my presentation of September 13, the purpose of Figs. 1 to 3 and the accompanying text was twofold: First, to set forth the common theory in simple form; second, to develop a simple graphical method of representing the effect of changes in compression upon efficiency, in the curve $K H$, Fig. 3. Incidentally, the well known but often insufficiently emphasized fact is clearly brought out, that even for the ideal case of similar curves in expansion and compression the economical degree of compression depends upon the degree of expansion.

Of Figs. 4 to 6 and the related discussion, nothing need be said, as they merely fill out the presentation under the assumption of curves of the form $P V = C$. That the clearance steam has a negative work cycle of some sort is indisputable. Only Figs. 7 and 8 are directly concerned with the Dwelshauvers-Dery experiments and their bearing upon the design and operation of the engine.

COMPRESSION—ACTUAL CURVES

The point really at issue is summed up in the difference between the forms of the curve $E F$ in these two figures. In a small engine at low speed the type of curve in Fig. 7 was obtained: the conditions favored a high relative amount of cylinder-wall effect, because small size means more metal surface per unit (weight) of steam handled, and low speed means plenty of time for heat transfer. In larger engines at ordinary speeds, or in small engines at high speed, while the action is of the same character as in this small and low-speed engine, it is relatively much smaller in amount.

Without absorption of heat from the steam by the cylinder walls, compression would be adiabatic; superheating would result and increase, and $P V$ would increase progressively. In many engines, the heat absorption is only enough to lower the curve about to the form $P V = C$; in others, it causes the moderate decrease shown in Fig. 8; only under exceptional conditions does the kind of curve in Fig. 7 appear.

Probably the largest published collection of data from engines of the large-clearance high-compression class is the report of the locomotive tests made in the Pennsylvania Railroad Company's testing plant at the St. Louis Exposition of 1904. Two simple and six compound engines were tested, the clearance ranging from 9 to 18 per cent. (considering only high-pressure cylinders), and the speed from 40 revolutions per minute upward. Almost one hundred sets of indicator diagrams are reproduced in the volume of reports. In no case is there a sign of the hooked curve, except when compression goes above the admission pressure and a loop is formed; then, of course, it is difficult to isolate the effect of the cylinder walls from that of early opening of the valve for admission. But in many instances the less intense action evinced by the straightening of the compression curve, as of EF toward F in Fig. 8, is strikingly apparent.

Suppose that the method shown in Fig. 3 is made complete by letting the ordinates represent the ratio of effective work to steam actually consumed, measured by test, the diagram being based on positions of the point E at which compression begins; the scheme will then be essentially the same as that of Figs. 3 and 4 (June 28 issue, page 1161), except that the ratios represented bear to each other the relation of reciprocals. Then curve KRH would have vertical ordinates, point K lying on the line AM, or corresponding to zero compression. According to the Dwelshauvers-Dery results, the curve would have its highest point at K and would descend steadily and with increasing speed toward H; in other words, the more the compression, the lower the efficiency. But in the absence of the tremendous condensation which causes the hook at the top of the compression curve, the saving of kinetic loss secured by moderate compression will certainly not be at once neutralized by increased cylinder condensation, and the curve KH will rise to a maximum before it begins to fall off.

To sum up, the simplified and incomplete analysis in Fig. 3 determines a certain degree of compression which appears to give the best economy. The successive introduction of actual and more closely empirical modifications brings R closer to K and lowers H, or shows that less compression is desirable and that the loss due to excessive compression will be greater. The essential contention that a steam action which makes even a small degree of compression cause net thermal loss is not normal and typical, but abnormal and extreme. This is all that was claimed anywhere in the article of September 13, and it does not seem to be seriously antagonistic to anything presented by Professor Dwelshauvers-Dery; even to express a slight doubt on the subject of leakage was not serious sacrilege.

Educational Program of N. A. S. E.

A list of questions relating to the topics to be taken up at the weekly meetings of association No. 37, of Ohio, during the present season. They are well worth looking over. See how many you can answer offhand.

An important feature of the program for the coming season, prepared by the educational committee of the National Association of Stationary Engineers No. 37 at Dayton, O., is a printed circular containing numerous questions bearing upon the topics to be taken up at the weekly meetings. These are distributed in advance among the members and afford an opportunity for them to come prepared to take part in the discussion. Following is a partial list of the questions:

ENGINES—TYPES

Classification—Types of modern engines; horizontal and vertical.

Name the different types of steam valves. Define the advantages of each.

For a 500-horsepower engine, which will be preferable, high or low speed?

Does the speed of an engine indicate any advantage in efficiency? Why?

Would there be any preference between the long-stroke or the short-stroke type?

What would be the maximum foot travel per minute of any piston?

Suggest the best points of cutoff for the highest efficiency, naming advantages.

Effect, if an engine were changed from one-third to two-thirds cutoff. Why?

Should there be any preference between a horizontal and a vertical engine?

What is meant by compound engines? Name advantages and economy.

Define difference and name advantages between cross- and tandem-compound types.

What should be an approximate ratio between high- and low-pressure cylinders?

Give the efficiency of the same engine on 100 pounds or on 150 pounds steam pressure.

Would a higher efficiency be secured by using condensers on an engine?

What gain in efficiency would be secured through a condenser? Why?

Is aid given to the steam piston by a vacuum? At 26 inches vacuum, how much benefit?

Name the best average vacuum obtainable through wet-vacuum pumps.

What is gained by covering steam cylinders with mineral wool and lagging?

How do we figure the horsepower and the steam consumption of an engine? Example.

STEAM—QUALITY AND PRESSURE

What is steam? Its chemical composition. Name the chief gases.

What gas predominates? Does dry steam carry water impurities with it?

Is low steam pressure as economical as high pressure? Why? Why not?

Would the same evaporation be required for steam at 100 pounds and 150 pounds?

Evaporating one cubic inch of water, name the volume of steam at 100 pounds and 150 pounds pressure with the steam piston 14 inches diameter; steam pressure 80 pounds; 550 feet piston travel, 100 brake horsepower.

Name the size of piston with steam at 150 pounds; 540 feet piston travel and 100 brake horsepower.

Can steam at 80 pounds be as dry as steam at 150 pounds? Why? Why not?

What percentage of moisture can steam carry without detriment to the engine?

What is superheated steam? Name process for superheating steam.

Does superheating the steam increase its pressure, or its expansive quality?

If expansive quality is secured, will it require as much steam as before?

What is meant by wire drawing the steam? Cause and effect.

What is the effect on steam if the reducing valve is placed at the throttle? Why?

What is the effect on a 14-inch piston if the pressure and expansive properties are increased?

CONDENSERS—TYPES

What is a condenser? Its uses and advantages. Name types fully.

Should a condenser be selected to conform to local conditions of water?

Square feet of cooling surface required for each engine horsepower.

Define the process with a condenser, for returning steam to water.

How would you determine the quality of cooling water for a condenser?

Would the same amount of cooling water be used if the temperature differed?

Given cooling water at 60 degrees, and water at 75 degrees, name the amount required.

Does the temperature of condensation indicate the degree of efficiency?

What is vacuum; how is it obtained, how maintained and the maximum obtainable?

What is the vacuum required for average efficiency? Name the advantages of a high vacuum.

Could a vacuum be maintained at the engine without the use of a condenser?

Are cooling tubes of large or small diameter preferable? Why?

In what way do barometric condensers differ from surface and jet types?

Define the different ways cooling water is used in different types.

Does the quality of cylinder oils have any effect on surface condensers?

Name the standard ways for measuring vacuum. The most reliable way.

Condenser Pumps

Name types of pumps required in condenser systems.

Define the types, construction, uses and advantages of different types.

Is there any difference between a water pump and a vacuum pump?

Define the difference between a wet-vacuum and a dry-vacuum pump.

Does a wet-vacuum pump give as high a vacuum as a dry-vacuum pump? Why?

Can a wet- and a dry-vacuum pump be attached successfully to condensers?

Are crank and flywheel vacuum pumps successful as wet-vacuum types?

Have direct-acting steam duplex vacuum pumps been successful? Why?

What is termed slippage in wet-vacuum pumps? Is it necessary? Why?

How would you determine the vacuum for a given condenser?

Name the remedy for a vacuum pump's inefficiency, when it is drawing steam.

Electric Equipment

Define the general laws governing electricity. Classify the basic laws.

Do we create electricity, or only furnish means for its transformation?

What would be the effect, were a generator placed in a vacuum and run?

Are generators and motors affected by barometric conditions?

What is a unit of electric force? Define a unit.

State preference, if any, between alternating and direct current.

Advantages in either high or low voltage. What is the term voltage?

Define the difference between electric and magnetic forces.

Define the principals in generator and motor construction.

Does the number of feet of wire, and the number of coils regulate the capacity?

Describe transformers, their use, construction, advantages and necessities.

Describe exciters and boosters, their uses, necessities and advantages.

Describe storage batteries, their uses, construction, care, safety and capacity.

Describe the wiring of generators and motors, also commutators and brushes.

Normal rating of motor; name the possible overload without injury to the motor.

Is there any difference in the shock at starting between alternating- and direct-current types?

What is meant by "phase," "cycle" and "alternations?"

Of the alternating- or direct-current types, which is the more economical?

Refrigeration

What is refrigeration? Classify the types.

Name the earliest tried systems. Name refrigerating materials.

What is anhydrous ammonia; its source of supply and its boiling point?

Ammonia: its latent heat, its sensible heat, units of refrigeration.

Is the temperature of ammonia at 150 pounds pressure the same as steam at 150 pounds?

Describe the difference between the compression and the absorption systems.

What is aqua ammonia, and its use in refrigeration systems?

How are refrigerating plants rated? What is meant by a one-ton capacity?

Describe plate and can ice; their advantages, difference and which is the more economical?

Can both plate and can ice be made by the same system? If not, why not?

Name the functions of cooling coils. Are cooling coils and condensers the same?

Define the uses and effect of brine in refrigeration.

What is circulation, its necessity, how is it accomplished and give the strength of brine?

What are skimming and reboiling tanks, their uses and effects?

For each ton of refrigeration, how many pounds of ammonia are necessary?

What is the lowest possible temperature obtained through refrigeration?

Under ordinary care, can ammonia come in contact with the forming ice?

Define the cause of soft white or red core in ice cakes, and name the preventative.

Lubrication

Classify lubricants. Name the lubricants used before petroleum oils.

Would the steam-cylinder lubricants of 50 years ago answer now?

Have animal fats or oils been improved upon by petroleum products?

Define differences in crude oils, as found in different localities.

Can all crude oils be converted into cylinder oils? Why not? Why?

Define the difference between "dry and wet" distillation of crude oils.

Of the two systems, which is better?

Define the difference between pure mineral oils and compounded oils.

Where can pure mineral oil be used, and are there any pure oils used?

Do the temperature of steam and its quality affect cylinder oils?

What is known as "saponification" of cylinder oils? Name its cause and effect.

Give the meaning of the following terms: Specific gravity; viscosity, fire test.

From what part of petroleum do engine oils come?

What are the most desirable qualities in engine oils?

Are all engine oils filterable? Are filterable oils preferable? Why?

Are not more lubricating oils wasted than are really used or necessary?

Do you keep records of the oils used? Do you test the different oils you use?

Should oils be compounded from an analysis of conditions?

What causes a large part of so-called mineral oils to become rancid?

Suggest the best method for feeding cylinder and lubricating oils.

"Rules is Rules" on the Railroad

By E. Dixon

The man on the locomotive is guided by the signal and extremely rigid rules are enforced on the big trunk lines in regard to running by signals, even when they are out of order and being repaired. Defective signals cause train delays, which in the case of the fastest runs have to be very thoroughly explained, particularly if that special train is the pet of the grand high "panjandrum." In order to keep tab on the observance of the signal rules the "surprise test" is given at intervals. Some signal on the division is selected and, if in the day time, the blade is removed, while at night the lamp is extinguished or hooded. Then a note is made of all trains which fail to stop and investigate. Some moons ago the "president's pet" was a few minutes late and the moon shone bright as day; the light was out, but the paddle showed clear, so Fitzgibbon ran the signal and threw off a report at the next tower. It happened to be a surprise test and Fitz, like the others, got his little thirty days. The mere fact that he had brought his train in on time was not even an extenuating circumstance, nor the fact that it was not his regular run. The high mogul told Fitz, "Rules is rules." "All right," says Fitz, "you just can remember you gave the instructions."

Fitz's regular run was the one ahead of the flier when they pulled off the next test. The light was out and Fitz stopped; the inspector gave him a manual sign to go on but Fitz was obdurate; there was no light lit. Finally the inspector came down and was told Fitz would stay there till the light was lit. In the meantime the flier had crawled up on him and was telling him to get a move on in a most emphatic fashion. Cooney can pretty near swear with his whistle and he outdid himself then. The inspector climbed the pole, lit the light and cleared Fitz, who gave a derisive pull to his whistle as he started up. Fitz had the rules in back of him this time. The inspector could not claim the lamp was out of order, while the rule clearly said that the engineman must not pass the signal unless the light was lit. Of course, there were requests for the reason why the flier was laid out, but all they could get out of Fitz was, "Rules is rules."

Gas Power Department

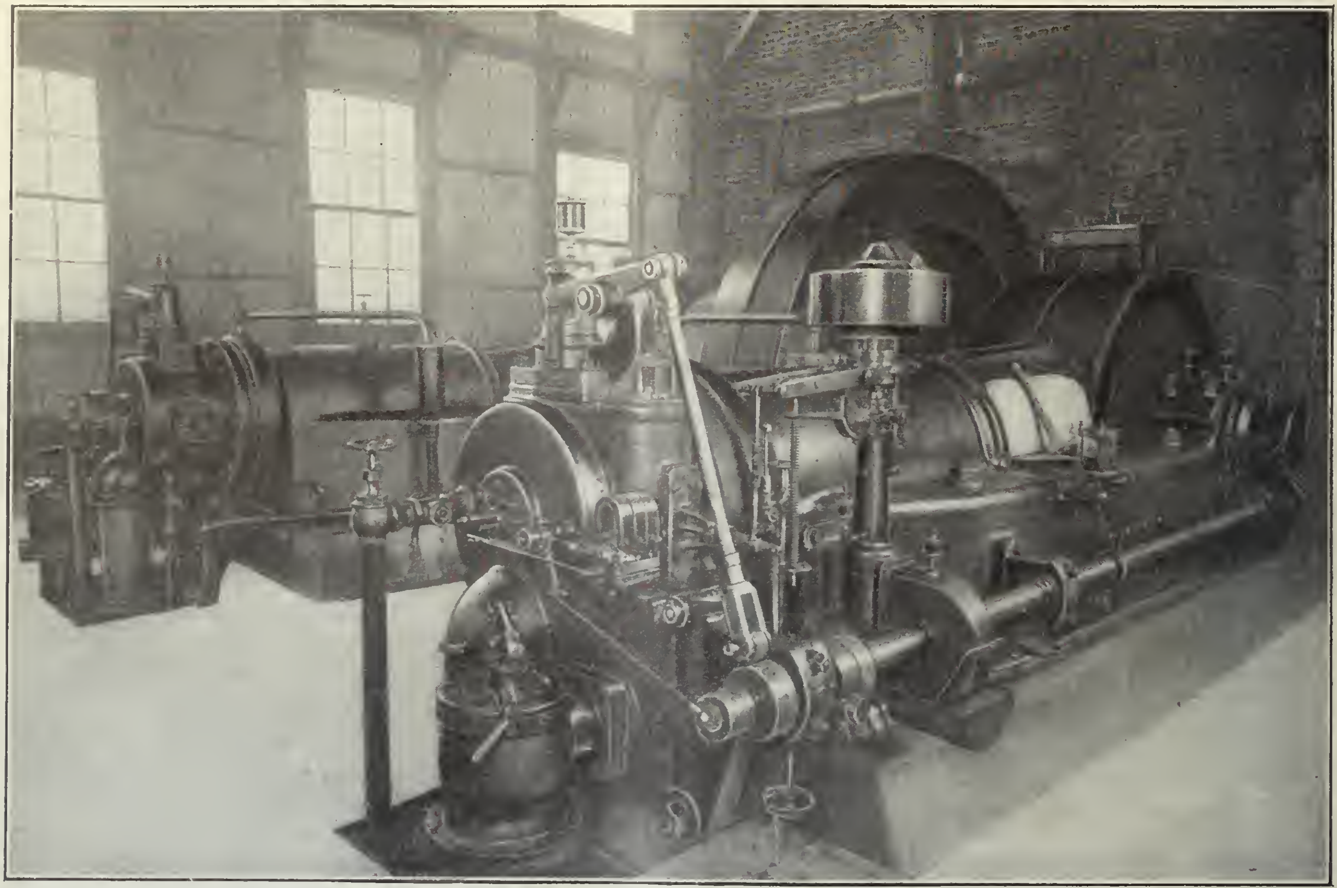
Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Performance of a Gas Power Plant Using Lignite

By A. M. Levin

The gas-generating apparatus of the power plant of the Olympia Brewing Company, Tumwater, Wash., consists of an up-draft suction producer of somewhat special form, a tower scrubber and a centrifugal gas washer, all designed and built by Peter G. Schmidt, mechanical superintendent of the Olympia company. The producer is specially adapted for operation on noncoking bituminous coals and lignite, which are mined in the State of Washington and are obtainable there at low prices. The generator lining is 72 inches in diameter inside, and the hight of the fuel bed ordinarily carried approximates 6 feet. The producer has been in regular service since April, 1909, with entire satisfaction, and for the past six months its fire has not been drawn.

The gas engine is a 20¾x32-inch single-acting twin-cylinder machine, rated at 250 brake horsepower when running 180 revolutions per minute, built by the Minneapolis Steel and Machinery Company. It is belted to a Triumph refrigerating ammonia compressor, with a cylinder 17x30 inches, and also drives, by belt, a 250-volt direct-current generator. Fig. 1 gives a view in the engine room of the installation.



Twin Single-acting Engine Running on Lignite Gas in Washington

A few months ago a test was made to ascertain the efficiency of the plant as a whole under normal working conditions, and, as far as the facilities at hand would allow, the efficiencies of the engine and the producer separately. As the plant was operated with a very small load during the time of the test, as is generally done in the fall season, it was deemed advisable to add a water rheostat to the electrical load during a part of the test in order to obtain a total load somewhat more normal and better suited for the capacity of the installation.

A preliminary 12-hour test was made on August 15, 1910, and a final 24-hour test on August 23 and 24. Table 1 gives the average figures from the log sheets of the tests for each six hours' run, and the totals for each test. The second column of the table contains the averages of the electrical load in kilowatt-hours. From these figures and the efficiency of the engine and dynamo were computed the electric horsepower in column 3 and the corresponding brake horsepower at the engine, in column 4. The efficiency of the electric generator was estimated at 92 per cent. and the belt loss between the engine and the generator at 8 per cent. of the power transmitted; the reduction factor for the translation of electrical to brake horsepower, therefore, was

$$\frac{1}{0.92 \times 0.92} = \frac{1}{0.846}$$

The justification of this factor will appear later.

The mean effective pressures corresponding to the work of compression in column 8 were computed by means of the formula:

$$M.E.P. = 0.92 \times 3.5\, P_i \left[\left(\frac{P_f}{P_i} \right)^{0.286} - 1 \right],$$

in which P_i represents the absolute inlet pressure and P_f the absolute discharge pressure. The average mean effective pressures of the indicator diagrams from the compressor, which were taken every fifteen minutes during the test of August 23 and 24, are given in column 7 and it will be noticed that these average values and the average values obtained by the formula agree very closely. The mean effective pressures of column 8 of the test of August 15 can, therefore, be considered practically correct, although not corroborated by indicator diagrams. The figures for the indicated horsepower of the compressor in column 9 were computed for a speed of 49 turns per minute, which was approximately the average speed throughout the day's run. The brake horsepower in column 11 is that at the engine shaft corresponding to the indicated horsepower of the compressor, an allowance of 15 per cent. being made for friction in the compressor and 6 per cent. for loss of power in the belt transmission between the engine and the compressor. The reduction factor for converting the indicated horsepower of the compressor into brake horsepower at the engine was therefore

$$\frac{1}{0.85 \times 0.94} = \frac{1}{0.80}$$

TABLE 1. RESULTS OF TESTS.

Time.	Electric Load.			Compressor Load.						Total B.H.P. at Engine Col. 4 + Col. 10	Total Coal Consumed, Pounds.	Coal per Hour. Pounds.	Coal per Hour per B.H.P., Pounds.	Coal per Hour per I.H.P., Pounds.
	Kilowatts.	Electric H.P. 1.34 × K.W.	B. H. P. at Engine $\frac{E.H.P.}{0.846}$	Inlet Pressure, Pounds Gage.	Discharge Pressure, Pounds Gage.	M. E. P		I.H.P. of Compressor at 49 R.P.M. Col. 8 × 1.65	B.H.P. at Engine $\frac{I.H.P.}{0.80}$					
						From Cards.	Computed by Formula							
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Test of Aug. 15.														
8:30 A.M. to 2:30 P.M. ...	55.079	73.8	87	13.875	155.25	......	61.0	101	126	213				
2:30 P.M. to 8:30 P.M. ...	32.911	44.1	52	9.1	152.4	......	57.5	95	120	172				
										385 192.5	3428	360.6	1.873	1.611
Test of Aug. 23 and 24.														
8:00 A.M. to 2:00 P.M ...	63.1	84.5	100	16.67	155.1	62.5	62.46	103	128	228				
2:00 P.M. to 8:00 P.M. ...	67.303	90.0	105	17.92	157.7	64.08	63.6	105	131	236				
8:00 P.M. to 2:00 A.M. ..	72.35	96.0	112	12.6	157.3	61.71	61.1	101	126	238				
2:00 A.M. to 8:00 A.M. ...	70.629	95.0	111	12.04	158.2	59.0	61.12	101	126	238				
										940 235	8730	364.0	1.55	1.333

As the basis for estimating the losses in the electric generator, the compressor and the two belt transmissions, a set of indicator diagrams taken from the gas engine during a two-hour run when the load was heavy and tolerably steady was used. The diagrams were taken simultaneously from the right-hand and left-hand cylinders and at reasonably even intervals. Observations of the electrical output as well as of the compressor load were made while the diagrams were being taken. The average figures for this heavy-load period were 84.41 kilowatts delivered by the dynamo, 95 indicated horsepower at the compressor, and 290 indicated horsepower in the two gas-engine cylinders. Apportionment of the gas-engine's work between the generator and the air compressor was determined by the individual figures from which the averages here stated were obtained.

Of course, nothing is proved with regard to the mechanical efficiency of the gas engine, but this factor is amply well known by experience with engines of the same build as the one tested. Because of the excellent condition of the cylinders and very thorough lubrication, it is reasonably certain that the mechanical efficiency of the engine was not lower than the 86 per cent. allowed in translating indicated horsepower into brake horsepower.

In order further to verify the correctness of the estimated losses of power between the gas engine and the driven machines for lighter loads, a set of indicator diagrams was taken during the period from 4.15 to 5 p.m. on August 15. The average indicated horsepower was 226, from which the average brake horsepower was calculated to be 194. On the basis of figuring losses just described, the brake horsepower required by the dynamo was 79 and that for the compressor 120, making the total 199 as compared with 194 actual. The discrepancy is not material.

Because of a misunderstanding with regard to the proper depth of ashes to be maintained on the grate of the producer during the test of August 15, the fuel consumption became unusually high, and an abnormally high percentage of unconsumed coal remained in the ashes. This test, therefore, is not of value as an independent efficiency test, but it is of importance as a preliminary test to that made on August 23 and 24.

The total coal consumption during the 12-hour test of August 15 was 4328 pounds or 360.6 pounds per hour for 192.5 brake horsepower, giving 1⅞ pounds per brake horsepower-hour and 1.61 pounds per indicated horsepower-hour. The average calorific value of the fuel as fired during this test was 7409 B.t.u., and hence the economy of the plant as a whole was 11,928 B.t.u. per indicated horsepower-hour.

The total fuel consumption during the 24-hour test of August 23 and 24 was 8730 pounds, or 364 pounds per hour for 235 brake horsepower, giving 1.55 pounds per brake horsepower-hour and 1.333 pounds per indicated horsepower-hour. The average calorific value of the fuel, as fired, was 7560 B.t.u. per pound, according to analyses, so that the economy of the plant as a whole was 10,077 B.t.u. per indicated horsepower-hour.

The engine jacket water was kept at approximately constant temperature during the tests, observations made at regular intervals showing the temperature of the discharge water to vary from 123 degrees at light loads to 129 degrees at heavy loads. The heat dissipated in the jacket water can, therefore, be considered to vary approximately in proportion with the load carried by the engines.

Careful measurements of the jacket water showed that the right-hand cylinder jacket discharged water at the rate of 5507 pounds per hour, at an average temperature of 128 degrees Fahrenheit and the left-hand cylinder jacket discharged water at the rate of 5850 pounds per hour, at an average temperature of 126 degrees. The total rate of flow was therefore 11,357 pounds per hour, or 6 gallons per brake horsepower-hour. The inlet temperature of the water was 68 degrees, hence the total number of heat units carried away per hour was 669,700. The average horsepower developed by the engine during the time the jacket-water measurements were made was 239 brake horsepower, or 277 indicated horsepower. Hence, the heat loss in the jacket water was at the rate of 2418 B.t.u. per indicated horsepower, or approximately 95 per cent. of the heat converted into indicated work.

Facilities were lacking for determining the heat dissipated in the exhaust gases, but for engines of the same type as the one tested, with the valve gear adjusted properly, this item, together with the radiated heat, has been found to average practically 1.28 times the heat actually converted into indicated power. Assuming this ratio to be substantially correct, the heat distribution would be as follows:

DISTRIBUTION OF THE HEAT VALUE OF ONE POUND OF COAL.

	B.t.u.	Per Cent.
Total heat value of 1 pound of coal.	7560	....
Heat converted into brake horsepower	1642	21.7
Heat converted into indicated horsepower	1910	25.3
Heat dissipated in the jacket water	1814	24.0
Heat carried off in the exhaust and radiated	2445	32.3
Heat loss in the gas producer	1391	18.4

The thermal efficiency of the engine, accordingly, was 30.9 per cent. on the basis of indicated horsepower or 26.6 per cent. on the basis of brake horsepower.

The apparent unusually high efficiency of the producer, 81.6 per cent. on a fuel high in volatiles, is mainly due to the fact that the entire volume of the generated gas must pass through the zone of highest incandescence, and partly perhaps to the circumstance that the composition of the volatiles is such that they readily become changed into fixed gases in the apparatus, leaving the proportion of tar extracted from the gas less than 1 per cent. of the weight of the fuel.

During the test of August 15 the gas was sampled at intervals and chemical analyses made. Continuous calorimeter tests of the gas were also made, from 11 a.m. to 8:30 p.m. A fair average composition for the gas is obtained by taking the mean of the following two analyses:

ANALYSES BY VOLUME OF GAS SAMPLES TAKEN AUGUST 15.

	At 9 a.m.	At 3 p.m.	Averages
CO	17.2	15.2	16.2
CH_4	4.7	5.2	4.95
H	24.7	25.9	25.3
O	0.6	1.9	1.25
CO_2	11.2	2.7	6.95
N	41.6	49.1	45.35
High heat value, B.t.u.	184	186	185
Low heat value, B.t.u.	166	167	166.5

CALORIMETER GAS TEST MADE AUGUST 15.

	At 11 a.m.	At 3 p.m.	Averages
High heat value, B.t.u.	184	187	185½
Low heat value, B.t.u.	166	169	167½

The average heat values of the gas obtained by calorimeter tests during the period from 11 a.m. to 8:30 p.m. on August 15 were 185 B.t.u. high value and 167 B.t.u. low value. The condensation collected in the calorimeter was 0.016 of a pound per cubic foot of gas. The hydrogen and methane content shown by the gas analysis being unusually high for producer gas, it will be of interest to check these items by the weight of the condensation collected in the calorimeter. Each cubic foot of gas contained, on the average, $0.0495 \times 0.0422 = 0.0021$ pound of methane and $0.253 \times 0.00526 = 0.0013$ pound of hydrogen. The weight of water formed at the combustion of these gases should be $2.25 \times 0.0021 = 0.004725$ pound from CH_4 and $9 \times 0.0013 = 0.0117$ pound from H_2, making the total weight of water per cubic foot of gas 0.016425 pound. This computed weight of water checks very closely with the amount, 0.016 pound, actually obtained in the calorimeter, showing that the hydrogen and methane proportions given by the analyses are substantially correct.

The analyses of the gas showed it to be entirely free from any tar-forming heavy hydrocarbons, and this tar-free condition of the gas as delivered to the engine was also indicated by the nonluminous flame with which it burned upon being lighted at a 5-inch vent pipe near the engine.

A purity test of the gas was made on August 15 by passing 40 cubic feet of gas through a filter paper which before the test weighed 1.32 grams and after the test, 1.352 grams. The impurities abstracted therefore weighed 0.0320 gram, making the weight of impurities per cubic foot of gas 0.0123 grain.

The analyses of the lignite fuel used during the tests were as follows:

ANALYSIS ON AUGUST 13.

	Per Cent.
Moisture	17.6
Volatile matter	28.2
Fixed carbon and sulphur	31.9
Ash	22.3
Heat value, dry coal, B.t.u.	8992
Heat value, as fired, B.t.u.	7409
Color of ash, yellowish gray.	

ANALYSIS ON AUGUST 24.

	Per Cent.
Moisture	16.2
Volatile matter	28.9
Fixed carbon and sulphur	32.1
Ash	22.8
Heat value, dry coal, B.t.u.	9021
Heat value, as fired, B. t. u.	7560

The quantity of water used in the tar extractor for the cooling and cleaning of the gas amounted to about 4 gallons per hour per brake horsepower.

The question of using lignite as fuel for gas-power purposes being comparatively new and discussed infrequently, it may not be out of place in conclusion to emphasize the great ease and efficiency with which the fuel can be utilized in the producer. It cannot be claimed, of course, that the test I have described will prove that equally good results can be obtained from all lignites, because the composition of different fuels varies extensively, but as the situation is at present, and as a general proposition, it may be said that, for equal numbers of heat units utilized for power purposes in a suitably constructed producer, the efficiency of a lignite fuel compares favorably with that of any ordinary anthracite, and it will most likely prove to be superior to that of bituminous coal; and this in spite of the fact that its percentage of moisture will probably be 10 to 15 per cent. greater than that of the other fuels. What economy may under these conditions be derived from the use of lignite depends, of course, on the price per unit of heat value at which it can be laid down at the place that the lignite is to be used.

Considerable interest has been aroused by what is now frequently referred to as the "growth of cast iron after repeated heatings." Some difference of opinion as to the cause of this effect was at one time manifested, but in a paper on the subject read recently before the Sheffield branch of the British Foundrymen's Association, by Prof. H. Carpenter, this growth was definitely ascribed to the chemical reaction that took place between flame gases and the constituents in the iron, particularly the silicon constituent which has changed into silica. It was pointed out that another cause at work consisted of the gases which were dissolved in the iron itself originally, which also reacted with the flame gases. As the result of this enlargement, the iron to a very large extent lost its metallic properties, and became practically useless. In order to insure that iron should not grow in this fashion, it was explained that the silicon must be reduced to a very low figure, and there must not be any free carbon. The carbon must be chemically combined with the iron.

To prevent much of the pitting and corrosion in the dome of a pressure filter, or high portion of pipes containing warm or hot water, drill and tap at the highest point and put on a ¼-inch valve; no free oxygen or other gases can remain to attack the metal if this valve is allowed to leak a little water continually.

Electrical Department

Single Phase Transformer Connections

By Norman G. Meade

Alternating current is now used almost universally for electric light and power systems where the territory served covers a considerable area, because it has the advantage of being easily changed from a high voltage to a low voltage or *vice versa* by a simple and efficient apparatus, the transformer. This makes it practical to operate the transmission lines at high voltages and thereby secure maximum economy in the cost of the wires and at the same time avoid excessive losses in the transmission lines.

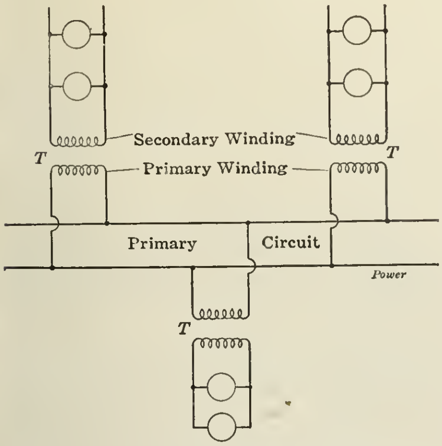


Fig. 1. Single-phase Distribution

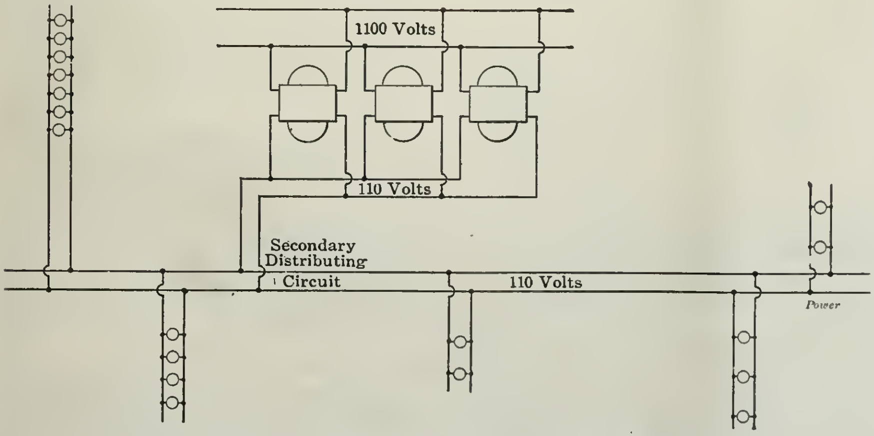


Fig. 2. Transformers "Banked" on Secondary Circuit

The single-phase system is generally applied where the current is used mostly for lighting purposes. When, however, there is a large motor load the two-phase or three-phase system is preferable. A typical single-phase circuit for furnishing current to lamps and motors is shown diagrammatically in Fig. 1. The primary windings of the transformers *T* are connected in parallel to the high-voltage primary circuit and the current-receiving devices are connected in parallel across the low-voltage distribution circuits connected to the secondary windings of the transformers. This system automatically maintains almost constant electromotive force in the secondary circuits and permits the greatest flexibility of distribution from those circuits.

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

When a large load is concentrated at one place, the transformers may be "banked"; that is, the primary windings of several transformers are connected in parallel across the high-voltage circuit and the secondary windings are also connected in parallel, dividing the load between the several transformers. Fig. 2 shows three transformers connected in this way.

Transformer leads are generally connected to the windings so that the instantaneous direction of flow of the current in certain selected leads is the same in all transformers of the same type. For example, referring to Fig. 3, the transformer shown is connected inside so that when the current flows inward in the primary lead *A* it flows outward in the secondary lead *C*. Some such system is advisable in order that transformers may be readily connected in parallel; by this means it is only necessary to connect similar primary and secondary leads on different transformers for parallel operation.

To make sure that a transformer is identical in this respect with other transformers of a different make, a polarity test is necessary. This is advisable to avoid the possibility of connecting the transformers in such a way as to short-circuit the windings. The polarity may be tested in the following manner: In Fig. 3, the primary lead *A* should be of the same instantaneous polarity as the secondary lead *D*. The primary lead *B* is connected to the secondary lead *D*, and an electromotive force of approximately 100 volts is applied to the primary leads *A*, *B*. The voltage measured from *A* to *C* should be greater than the applied voltage if the transformer is correct in polarity; if it is less, then the leads *A* and *D* are not of the same polarity, and either the primary or the secondary leads should be "crossed" if the transformer is connected in parallel with others which have the standard relative polarities.

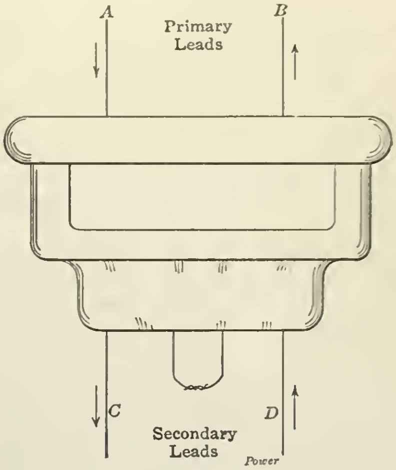


Fig. 3. Momentary Polarities

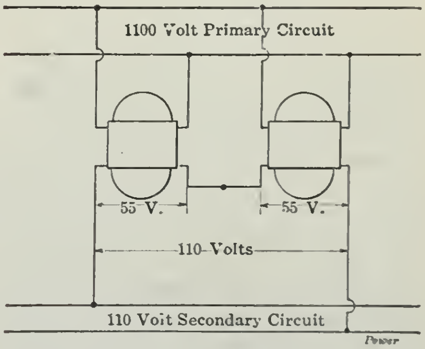


Fig. 4. Secondary Windings in Series

Sometimes it is desirable to connect two transformers with their secondary windings in series across the distribution circuit, as indicated in Fig. 4. By this arrangement of connections, twice the secondary voltage is secured that would be obtained if the secondary windings were connected in parallel as usual. In connecting the secondary windings in series, those leads must be connected to each other which are of *opposite* polarity at any given instant. Otherwise the two transformers will oppose each other and no secondary current will be delivered.

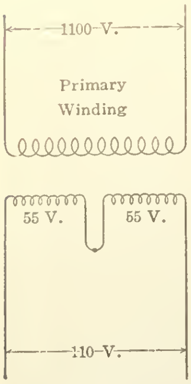


FIG. 5.

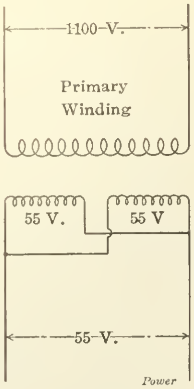


FIG. 6.

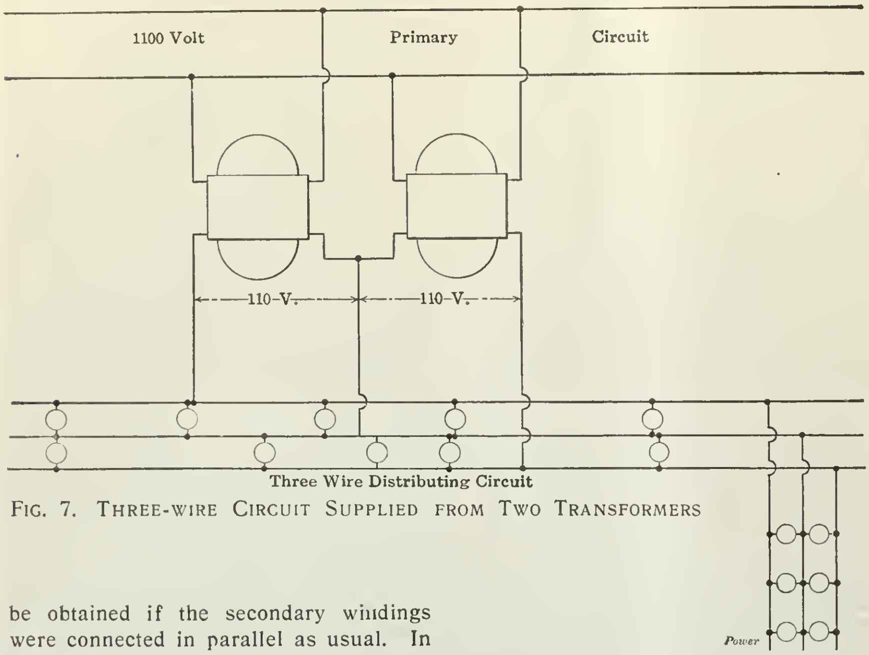


FIG. 7. THREE-WIRE CIRCUIT SUPPLIED FROM TWO TRANSFORMERS

Transformers are usually built with the secondary winding divided into two equal sections, and some have both primary and secondary windings divided. With a transformer having a divided secondary winding it is possible to obtain either one or two secondary voltages without banking, because the sections of the secondary winding can be connected either in series or in parallel. Fig. 5 is a diagram of a transformer with two secondary windings connected in series, which gives a secondary voltage twice as high as that obtained when the secondary windings are connected in parallel, as represented in Fig. 6. The same precaution against short-circuiting the windings has to be observed as when separate transformers are banked.

The object of providing a divided primary winding is to allow the transformer to be used on either of two voltages, one twice the other. For example, if each half of the primary winding is designed for 1100 volts, with the two halves connected in parallel the transformer can be used on an 1100-volt primary line, and by connecting the primary windings in series the transformer may be used on a 2200-volt line.

THREE-WIRE DISTRIBUTION

In Fig. 7 two separate transformers are represented, each wound for a voltage ratio of 10 to 1; that is, 1100 volts to 110 volts, and connected at the secondary leads so as to supply a three-wire system with a voltage of 220 between the outside conductors. The secondary windings are therefore connected in series, giving 220 volts between the outside leads. A lead is carried from their point of juncture to the neutral conductor of the three-wire system, in order to obtain the secondary voltage (110) of each transformer between the neutral and each of the main wires. This method of connection gives a system which is almost the exact equivalent of the regular direct-current three-wire system. The secondary windings of the transformers correspond to the armatures of the direct-current dynamos. The only point in which similarity fails is in the voltage regulation. Both of the direct-current dynamos can be regulated, and independently of each other, but the transformer voltage cannot be regulated at all except by the use of very special, complex arrangements.

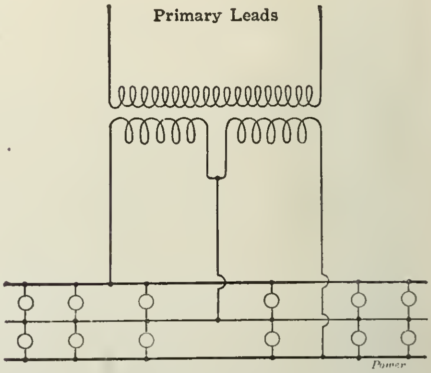


FIG. 8. THREE-WIRE CIRCUIT SUPPLIED FROM ONE TRANSFORMER

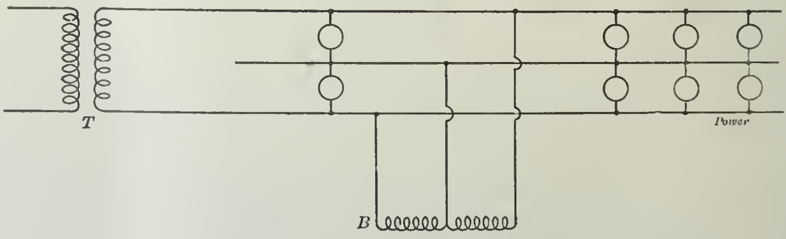


FIG. 9. THREE-WIRE SYSTEM WITH BALANCING TRANSFORMER

Fig. 9 shows still another method of connecting transformers for supplying a three-wire secondary circuit. The main transformer *T* is connected to the outside wires of the three-wire system and a balancing transformer *B* is connected to all three wires near the center of distribution. The current capacity of the balancing transformer need be only enough to carry one-half of the greatest difference in load between the two divisions of the system. A balancing transformer for this purpose is merely an autotransformer with an intermediate tap midway between the terminals of its single winding; this tap is connected to the neutral wire, as shown. If one-half of the three-wire system is loaded more than the other half, there will be a corresponding increase in voltage drop on the heavily loaded side. The half of the balancing transformer connected to the lightly loaded side of the system will aid the half connected to the heavily loaded side and tend to raise the voltage to normal. Its operation corresponds to that of the balancing set used on a three-wire direct-current system with one dynamo.

LETTERS

Circuit Breaker Alarms

Referring to Mr. Blomberg's letter on this subject, published in the issue of December 6, it may be well to point out the fact that it is not at all necessary to attach a live connection to the circuit-breaker handle, as my diagram showed it. The arrangement can easily be modified so that when the circuit-breaker opens the handle will strike an insulated lever and close the switch in the alarm circuit. My diagram merely represented the system as it was actually in use.

In view of the fact that the system is in use and has been for a number of years, Mr. Blomberg's assertion that the bell will not ring with the three lamps in series is rather hasty. However, it is only fair to him to explain that 50-volt lamps were used because the filaments are thicker and therefore more substantial than those of standard lamps. His argument that the failure of a lamp will disable the system is, of course, true, but it applies also to his own system. If one of the three lamps in parallel should go out, the current will be reduced to about 70 per cent. of the normal, which would not ring the bell properly unless the normal current were excessive.

Mr. Blomberg's system is open to the objection that the alarm will be operated by any failure of the main current or by the breakage of one of the connections between the relay and the main circuit, whereas the system I described can be operated only by the opening of the circuit-breaker—the event which the system is used solely to indicate. Moreover, the Blomberg system cannot be applied to several circuit-breakers without using as many relays as there are circuit-breakers, which would be expensive and entail a considerable clutter of small mechanisms.

GEO. W. MALCOLM.

Brooklyn, N. Y.

Governor Adjustments for Engines Driving Alternators in Parallel

When operating alternators in parallel I have found that if the governors are adjusted to give 3 or 4 per cent. drop in speed from no load to full load, the operation of the machines will be more satisfactory than if the speed regulation is closer. Some engineers favor close regulation, and this is all right for direct-current work, but for alternating-current machines 3 or 4 per cent. is better. It is a hard matter to set the governors so they will act exactly alike, and if they are adjusted to regulate very closely between no load to full load, any discrepancy in responsiveness will be a much larger percentage of the total variation than if the latter were greater; consequently, a small difference in the action of the governors will produce a larger difference in the load division than it would with less sensitive regulation.

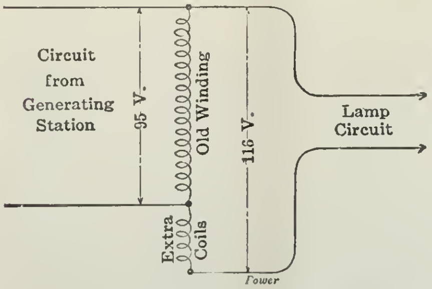
The engine governors should be so adjusted that there is no tendency to cause a transfer or surging of the load from one engine to another. This action may be caused by a sudden variation of the load, or by the angular variation of velocity in different parts of the revolution which is common to most reciprocating engines. If the governor is equipped with a dashpot it should be kept full of oil thick enough to prevent the governor from being sensitive enough to respond to such disturbances, but still not thick enough to prevent the governor from acting under the influence of a reasonable and definite change in the speed of the engine.

R. L. MOSSMAN.

Tampa, Fla.

An Improvised Alternating Current Booster

It frequently happens that some means of raising the voltage on a feeder or a single distribution circuit, without altering the busbar voltage of the system, is very desirable. Where direct current is used, the only way to raise the voltage of one part of the system alone is to insert a booster dynamo, which is expensive. In an alternating-current system, however, the voltage can be boosted or lowered by means of a transformer or an autotransformer; when the amount of boosting is small, an autotransformer is preferable.



BOOSTER CONNECTIONS

On one occasion in our plant a "lame" circuit was boosted from 95 to 116 volts by using the secondary winding and core of a 50-watt instrument transformer, the primary winding of which had been burned badly. The remains of the primary winding were removed from the core and the secondary winding was increased by 25 per cent. more turns and connected to the weak circuit as shown by the accompanying diagram, making a simple autotransformer of it. The secondary winding originally contained 32 turns of No. 22 wire, and we added 8 turns of No. 18, that size being used because the full-load current had to pass through that part of the winding. The additional wire reduced the heating and gave us a point to connect in the 95-volt tap without disturbing the insulation.

The potential at the extreme terminals of the "booster" was 118 volts at no load, but the drop brought it down when the load was on, and the load was either all on or all off, so that the question of regulation was not important. This little improvised booster has been supplying six incandescent lamps satisfactorily for several months.

WELDY S. YEAGER.

Colgate, Cal.

Mr. Gorilla's Induction Motor and Flickering Lamps

The trouble described by Mr. Gorilla in POWER for November 22 is probably due to slippage. When overloaded, the motor momentarily increases its slip and takes a heavy current from the mains, and then runs slightly above speed. This effect is called "hunting" and causes the lights on the circuit to flicker, due to the current in the motor being leading or lagging. A leading current causes the lights to burn brightly while a lagging current decreases their brilliancy. The trouble spoken of would be expected from a synchronous motor more than from an induction motor.

THOMAS H. WATSON.

Chicago, Ill.

[An induction motor cannot "hunt" under any conditions; on the contrary, it acts as a damper when the system shows a tendency to "hunt." If Mr. Gorilla's voltage fluctuates, the motor will follow the fluctuations but never lead them; neither will it take a leading current, unless driven as a generator considerably above synchronous speed. The fluctuations in voltage may be due to a sticky engine governor or to a load on the motor involving heavy reciprocating masses which during a part of their travel drag the motor above synchronous speed; but not to hunting inherent in the motor. —EDITOR.]

Bill Grimes et sum dried apples fer lunch tother day an' then crawled inter his biler t' titen up a loose brace. He got sorter het up, an' ast th' fireman t' hand 'im in a drink uv water. Well, when th' water an' them dried apples got t'gether thar wuz sumthin' doin' in Bill's interior department. He jist swelled up like a pizened pup an' when he tried t' crawl out he wuz jist 'bout four sizes too big fer th' hole. He laid in th' biler all th' arternoon with his hed stickin' out an' cussin' a blue streak while he wuz waitin' fer th' apples t' digest. Th' boss told 'im thet arter this he'd better eat a chunk uv alum fer his lunch 'fore he went inter th' biler so he'd sorter pucker up 'stid uv swellin'.

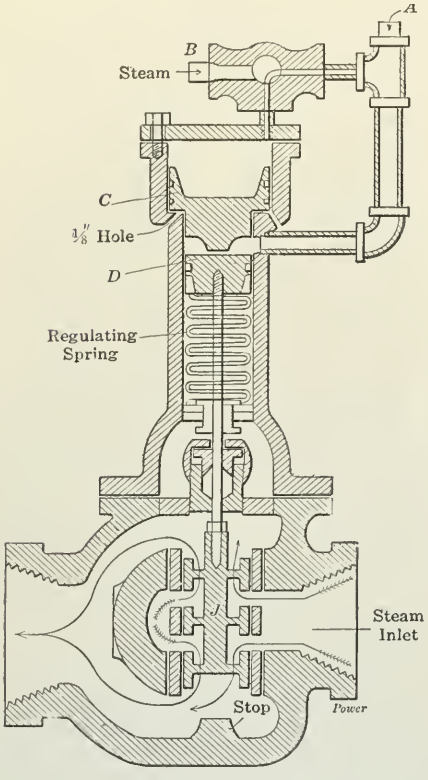
Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Safety Stop for Pressure Regulating Valve

When a pump is controlled by a pressure-regulating valve there is danger of injuring the pump, should the discharge pipe or any of its connections break, as the regulating valve would instantly open wide and turn on a full head of steam and probably wreck the pump, as has occurred in a number of cases which have come under my observation.

Because of this, I set about to devise some means to overcome this danger, and I believe I have effectually accomplished this by means of the device illustrated herewith.



Sectional View of Automatic Safety Stop

The inlet water pressure from the discharge main enters through the connection *A* and the three-way cock *B*, which is normally open from the connection *A* to the top of the piston *C*, which is open to steam only when just starting, as there is then no water pressure on the pump to hold the stop piston *C* in place.

A leather cup piston *D* actuates the balance valve *J* due to the pressure from *A*. The stop is to prevent a too far downward movement of the balance valve *J*. Should the discharge main break, or the pressure be suddenly relieved from the pipe *A*, the spring would force the pistons *C* and *D* to the top of the cylinder. When in this position steam would be shut off by the balance valve, thereby stopping the pump.

B. U. Potter.

Holyoke, Mass.

Troubles of a Refrigerating Engineer

In some of the mechanical ice machines that were built about ten years ago, it was the custom to install a small pump, driven from the engine shaft by a belt, to supply the oil for the stuffing box through which the piston rod works.

The oil for this pump is stored in a recess under the rod and is pumped into the gland, and in most cases considerable oil is carried through into the compressor. This was the case with the machine that I had charge off. On the discharge side of the compressor is placed an oil collector, also a rectifier, but the quantity of oil that was carried over into the compressor was so great that it was impossible to remove the oil from the rectifier as fast as it was carried over and, as a result, considerable oil was carried into the expansion coils in one room of the warehouse. The system f cooling was composed of coils of pipes suspended from the ceiling and fed by two ½-inch pipes with a regulating valve attached at the point of entrance to the expansion coils.

During the hottest part of the season the temperature began to rise in this room and I noticed that only one-half of the pipes were frosted over. During my search to locate the trouble, I found a pocket in the liquid pipe that fed this series of coil. On taking this lead pipe down, I found it clogged at the drop or pocket with a mixture that, when analyzed, proved to be oil and wearings of the cotton packing I was using in the piston-rod stuffing box of the compressor.

To remedy this trouble I devised a sight-feed lubricator and placed it on the machine and piped the pump to the lubricator. After installing the lubricator, the rectifier was kept going, and before I had caught up with the system nearly two barrels of oil had been recovered that had been hidden away in the recesses of the refrigerator coils.

At another time it was reported to me that there was a smell of ammonia in one of the cellars, and it was found that ammonia was escaping from the stem of the regulating valve. When attempting to take up on the packing a little more, I broke the bonnet. Warning the men who were working in the cellar to get out, I made for the engine room and shut off the supply of liquid and, closing off the rest of the suction valves, pumped on that set of coils until a vacuum had been secured. Then, supplied with a new valve, I went to the cellar to make the repair, after securing a large towel, saturated with water, over my mouth and nose, and a rope secured around my waist, leaving the end at the door in the hands of my assistant to pull me out in case I was overcome with ammonia, the new valve was put in place. This experience put me wise to the fact that cast-iron valves on a refrigerating system were not the best, and I took the first opportunity of changing them in my plant for those of steel construction.

Another experience made me do some thinking before I located the trouble. We have three suction valves on the machine I had, also three regulating valves conveniently placed in the engine room. One side of the machine is used for the direct-expansion system and the other is used in conjunction with brine tanks, centrifugal pumps, fans, etc.

It had been my custom on going off watch at night to shut off the direct-expansion side of the machine, as it was not necessary to run it continuously to secure the required temperature. I closed it off one night and instructed my night engineer as usual, but was called to the plant about 11 p.m.

I found the compressor shut down and all the information I could get was that the machine ran hot on both suction and delivery sides and had done considerable groaning in starting up. After securing a vacuum on the suction side and a low pressure on the condenser, it was plain that the machine was not getting any ammonia, and upon looking at the regulating valve that led to the direct-expansion side, I found the valve open about one-half turn. This explained the trouble, for the liquid ammonia had all been forced into the system that was closed off. Opening the suction valve by degrees and running the machine slowly for an hour or so, I got things back into normal conditions. The night engineer admitted that he had opened the valve as the pressure was a little too high on the condenser, but had closed it when the pressure had been reduced on the condenser.

W. G. WALTERS.

Stratford, Can.

Washing Boilers Externally

I would like to read the opinion of POWER readers on washing boilers externally. In one plant, containing water-tube boilers, each boiler is cut out of service by order of the superintendent about every four weeks and, as soon as it is cool enough to permit, a man is sent in on top of the tubes with a hose and all the ashes and soot washed down from the drums and tubes. The boiler is also washed out inside and is then inspected by the superintendent. This method of cleaning the tubes is used on account of the boilers being set so close together there is not room enough to get a blower of the proper length in between the tubes to properly blow the ashes from the tubes; they were blown to the opposite side and sometimes the space between the tubes was found packed solid in places, thereby decreasing the efficiency of the boiler.

There are several points for and against this practice. The boiler can be more easily and thoroughly inspected. Tube renewals (which are at short intervals here) can be more easily made as well as any other repairs to the boiler and setting, and under these circumstances, inspection and repairing are not dreaded as much as they are otherwise, and as the boiler is quite cool before the water is put on, the walls are not cracked by sudden contraction. The firebox lining usually lasts about one year with the boilers working at an overload one-half the time and seldom less than three-quarters load.

The objections against this practice are that it is liable to cause the setting to settle and crack the walls, and it takes more coal to get the boiler setting up to the working temperature again. No serious results have occurred in this plant to my knowledge after two years of this practice.

Several of the engineers' books say that "water must not touch the outside of the boilers, as it will cause corrosion." This is caused by the sulphuric acid in the soot combining with the water and attacking the iron, but in this case the soot is all washed off and, as the walls are warm and the draft is left open, the moisture remaining on the walls and tubes is soon evaporated, and after standing idle a week, the rust or corrosion on the outside of the tubes and drums can be easily wiped off by hand.

J. CASE.

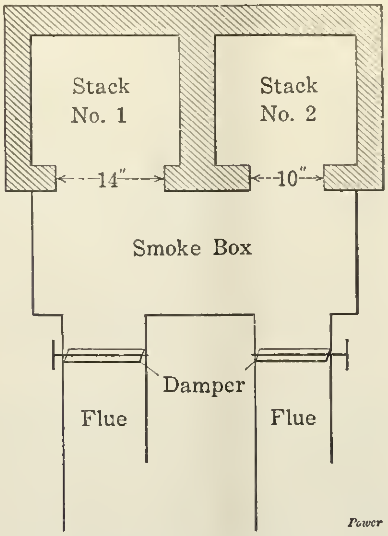
Hyattsville, Md.

Chimney Problem

With two smokestacks of equal hights and cross-sections as shown in the figure, will the hot gases pass up the stack with the larger opening and draw cold air down the stack with the smaller opening and hamper the draft?

The stacks serve two boilers of the locomotive type of 50-horsepower rating each. The stacks are 22 inches square and 40 feet high.

Would it be advantageous to divide the smoke box and so have one stack for each boiler and thus interrupt a circuit of air from one stack to the other? Or, would one stack be sufficient to serve both boilers, thus allowing the other to be cut out?



ARRANGEMENT OF CHIMNEYS

L. G. WATRY.

Pueblo, Colo.

Why the Engine Stopped

While working for a threshing-machine company last summer I was sent out to find and eliminate the cause of a peculiar binding in an engine that would run smoothly for some time and then suddenly begin to bind and after a few more revolutions stop. Then the threshing crew would try to turn the band wheel to throw the crank over the dead center, but could not move it.

Then the engineer would go over the engine, feel of the bearings, perhaps make a few adjustments, as he had seen the old engineer do, pump some cylinder oil into the steam chest, then try again to move the band wheel as one man could turn it easily, but the trouble would return.

They concluded the oil pump was not doing its work, so purchased a new one, but in vain.

An expert was called and after two days' hard work making several adjustments confessed he could not fix the engine, and returned home.

It was then that I was called to try my luck.

The water used in the boiler came from various sources, some from a flowing well of mineral water, some from a surface well of alkali water and some from a flowing stream impregnated with considerable vegetable matter.

The boiler foamed badly and with a decrease of pressure or increase of load the engine would pick up a volume of water and the piston would begin to groan.

They were also using a loose hemp rope of a very poor grade for piston packing, and when the engine primed the water would be forced out around the piston rod and wet the hemp, which would swell and grip the rod; but after standing a few minutes the heat from the rod would dry out the packing and cause it to release its grip upon the rod.

I advised blowing off frequently, washing down regularly three times a week or else use one kind of water and wash once a week, keeping the steam at a high pressure and about one gage of water, and getting some good grade of piston packing.

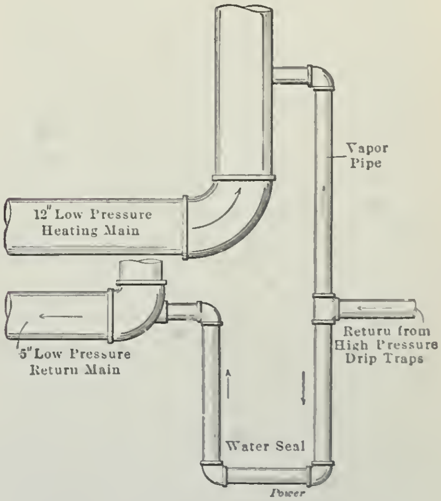
WILLIAM E. PIPER.

Farmington, Utah.

Connecting High Pressure Drips to Heating Mains

The plant at which I am employed had about fifteen high-pressure traps discharging to the sewers at various points about the main and adjoining buildings.

When our supervising engineer tried to get an appropriation to put in a return line recently it was refused, as the lawns and asphalt would have to be dug up to put in a pipe and conduit to the boiler house 500 feet distant.


Vapor Pipe
12" Low Pressure Heating Main
5" Low Pressure Return Main
Return from High Pressure Drip Traps
Water Seal
Power

CONSTRUCTION OF WATER SEAL

While discussing the problem our boys hit on a plan that worked out successfully.

A return line varying from ¾ inch to 2 inches in diameter was run through the cellars to the point where the main return left the main building for the power house. This line was connected to a water seal about 4 feet deep, from the top of the seal on the inlet side. A 2-inch vapor or equalizing pipe was run to the low-pressure heating main to prevent any steam or pressure blowing the seal out into the return to cause water hammer.

Now it is possible to heat with ½ to 2 pounds pressure (Paul system), and the vapor from this high-pressure drip now helps to heat the buildings instead of the sewer. We saved digging up the lawns and driveways and the cost of 500 feet of pipe and the labor of putting it in.

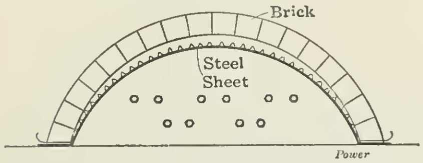
W. T. Meinzer.

Brooklyn, N. Y.

Sheet Steel Brick Support

While looking over a couple of old boiler settings recently, I came across a method of covering a boiler that I think is worth passing on.

The usual practice when bricking in the top of a return-tubular boiler is to lay the bricks directly on the boiler shell. But the boilers referred to, which had been in service for 25 years, had an arch formed by sheets of steel sprung over the top of the boiler and supported on the side walls as shown in the accompanying illustration. This arch carried the weight of the brick covering and provided an air space between the boiler and brickwork. This opening was closed up at each end to prevent air from circulating between the boiler and the steel casing. This arrangement also served to prevent any water that might drop from overhead valves and piping from soaking through the brickwork to the boiler shell.



Bricks Supported Over Boiler

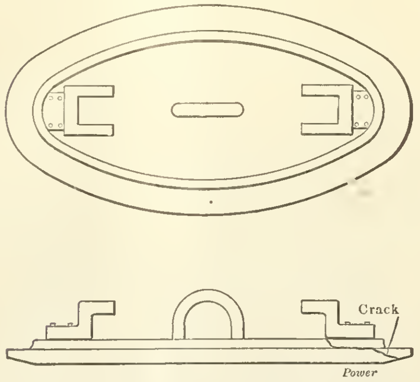
Charles Fenwick.

Qu Appelle, Can.

Cracked Manhole Plate

A new 66-inch by 18-foot tubular boiler was put in a plant three years ago and I had more or less trouble with the bottom manhole in the front head, which leaked, and was due, I thought, to the rough surface, the countersunk rivet heads or a blow hole in the cast-iron plate.

A few weeks ago I washed the boiler and a few drops of water leaked out around the gasket when the plate was in place. On a closer examination I discovered that the leak came from inside the flange that was riveted to the manhole plate. When the boiler was again empty, I took a sledge and broke the manhole plate and found a crack on the outside of the manhole plate half way through the flange of cast iron, as shown in the illustration.



Manhole Plate Cracked

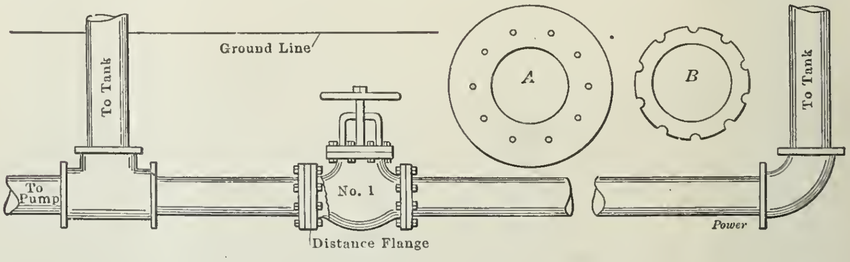
G. A. Gustafson.

Chicago, Ill.

Using a Distance Flange

At the plant where I am employed there is a duplex pump which discharges through a 5-inch line, over the tops of two tanks, each 30 feet high, one being 75 feet from the pump and the other 200 feet.

Last winter the ground settled and caused a 5-inch gate valve to break, as shown in the accompanying illustration. The old valve was taken out and another ordered, but in connecting up it was found that the new valve was ½ inch short.



Showing Distance Flange in Place

As the pressure in this line never exceeds 30 pounds it was decided that the ½-inch space could be taken up by a ½-inch pine board, made in the form of a distance washer as shown at *A*. Everything went fine till the next afternoon, when the joint sprunk a leak. Another piece was put in which held for about the same length of time.

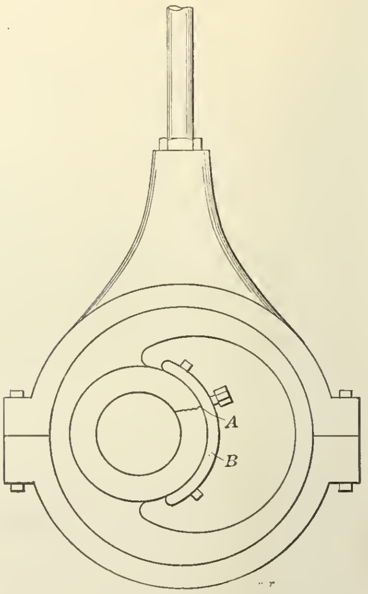
The third trial resulted in a disk made of wood and formed as shown at *B*. The flanges were pulled squarely up, making a comparatively flexible joint, and the job is still holding.

B. F. Hartley.

Tipton, Cal.

Repairing Broken Eccentric

The accompanying sketch shows how a repair was made to an eccentric of a wet-air pump, which cracked at *A*, through the hole for the adjusting screw. A steel plate *B*, ⅝ inch thick, and as wide as the eccentric body, was bent to conform to the curvature of the eccentric, and four ½-inch holes were drilled in it, corresponding holes being tapped in the eccentric to receive set screws which held the plate firmly in place. The hole in the plate for the adjusting screw was tapped, the old thread through the break being drilled out. This secured the eccentric against any possibility of shifting on the shaft. After the valve was set, the shaft was cupped out about ⅛ inch under the adjusting screw.



Repaired Eccentric

Henry W. Randall.

Saginaw, Mich.

If short a reducer on threaded cold-water pipe, one can easily be made by first stuffing the fitting with oakum and pouring in babbitt metal where the bushing should be; when the babbitt has cooled the oakum should be removed and the newly made bushing and pipe should be made up with a pipe wrench, to prevent leakage from shrinkage.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Effects of Cold Air

In reading POWER I often see articles relative to firing methods in which the boiler plates are subjected to chilling drafts. Of course, this is rightly condemned, as there certainly is no benefit derived from admitting cold air to a furnace through the fire doors.

It is well known that a vessel containing boiling water does not get heated to a very high temperature relative to that of the fire.

Now, in the case of a boiler, how could cold air from an open fire door damage the plates? In my opinion, before the air could strike the plates, it would be heated to even a higher temperature than that of the plates over the grates. And, certainly, before it could reach the tubes, it would be hot enough so as not to do any harm.

H. C. FISKE.

North Yakima, Wash.

Thermometers versus Gages

C. H. Peabody's article in a recent issue on "Thermometers versus Gages" has suggested to me the desirability of using the thermometer to serve as a check for the gage.

When the fire is started under a boiler which has been out for cleaning or repairs, the steam gage is depended upon to indicate when the boiler should be cut into the line. If the gage fails to indicate the pressure correctly, the safety valve calls attention to the trouble. However, should the safety valve stick, which sometimes happens, there is a good chance for an explosion.

Perhaps there are not many instances when the steam gage fails to indicate the pressure correctly, but I have had this occur five or six times when steam was being raised.

After the first occurrence the gage pipes were blown out twice a week. But this did not stop the trouble. The boilers are of the water-tube type. The gage pipe is ¼ inch in size and tapped into the steam drum on top and brought down over the front. When the pipe was blown out the water was thick and black and seemed to choke the small pipe.

I am going to increase the size of the pipe to ⅜ inch and use an angle valve on the nipple which screws into the boiler. With this arrangement the pipe can be disconnected and cleaned with a wire.

A thermometer to show the temperature of the steam in the boiler would serve as a guide in case the gage pipe should become clogged.

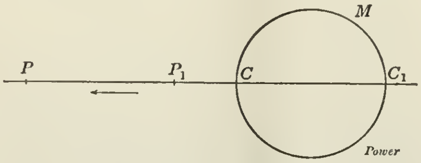
P. L. WERNER.

McKeesport, Penn.

Does the Crosshead Stop?

In the November 22 issue, Mr. Fryant asks, "If the crosshead does not stand still, how can it stop?" and refers to questions and answers relative to this in "Stevens' Mechanical Catechism."

A more satisfactory geometric proof that the crosshead does not stop will be made apparent by observing the accompanying figure.



CROSSHEAD DIAGRAM

Let P and P_1 represent the positions of the crosshead center when the crank-pin center is at C and C_1 respectively. When the crank pin is approaching the point C, the velocity of the crosshead is gradually decreasing, until the point C is passed, when the crosshead begins its return stroke. If C had a dimension, length, the motion of P might be altered, but as C is a point without dimension, formed by the intersection of two lines, P cannot theoretically stand still, as the progress of C around the circle M is not stopped.

The only time when P could stand still would be when C stood still or moved in an arc whose radius was PC, described about P as a center.

CHARLES M. ROGERS.

Detroit, Mich.

In the issue of November 22 appears a question regarding the motion of the crosshead. A quotation from a catechism is given to the effect that the crosshead does not stand still at dead-center points; and the writer of the letter asks how the change of direction can be made without a stop.

The answer in the catechism appears to me to have this fallacy in its reasoning: the crank pin is always moving and therefore the crosshead which is connected to it must move also. In the accompanying diagram $C—D$ represents the connection between pin and crosshead. The pin is moving in the direction $A—B$, which for a very small distance is at right angles to $C—D$; therefore, at this instant, it is impossible for the pin to produce any motion at right angles to itself through the connecting rod. The crosshead cannot be moving in either the direction H or K at this particular instant, and, hence, must be standing still.

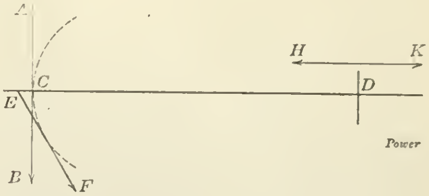


DIAGRAM OF CRANK-PIN TRAVEL

When the pin has passed on, its momentary direction will be along $E—F$, which, by shortening the distance between the pin and D, causes motion of the crosshead in the direction K.

In actual time the stoppage is very short, but it exists nevertheless, while the motion of the pin through C neither shortens nor lengthens the horizontal distance $C—D$.

H. S. BURROUGHS.

New York City.

This question can be settled in a very simple manner if it is assumed that the motion of the crank pin is uniform. With this as a basis, a velocity curve can be plotted for the crosshead, motion in the direction of the crank being plotted above the base line and motion in the opposite direction below the base line. The length of the base line represents the distance which the crosshead moves while its velocity at any point will be represented by the distance from the base line to the curve. The shape of this curve will show that the crosshead velocity decreases to zero and immediately commences to increase, and that this change from positive to negative direction occurs just as smoothly as do numerous changes in velocity at other points in the stroke.

This crosshead question is very similar to the older one about the barrel of wine. Half of the wine being consumed, the barrel was refilled with water. This sequence of operations was repeated. The problem is: How long was it before the barrel contained nothing but water? Should the crosshead actually stop when the engine was at top speed, things would commence to happen immediately.

Why is a cow?

E. DIXON.

Cleveland, O.

I note in the issue of November 22 the letter by H. J. Fryant regarding that old question, "Does the Crosshead Stop?" I fail to find any good clear argument in favor of its not coming to a dead stop, for when rotary motion is converted into rectilinear motion there must certainly be a cessation of the latter motion every time the rotary part, viz., the crank, reaches the end of its throw. This interval of cessation may be longer or shorter, dependent, of course, upon the radius and speed of the rotating part and the length of the connecting rod. It seems to me to be a very simple thing to find the solution of this problem as it is utterly impossible for an object advancing in a straight line to reverse its motion and return over the same line without first ceasing to move in the original direction. The crank-pin center is an established fact notwithstanding all arguments to the contrary. Or, to be more explicit, there is a dead point at each end of the stroke when the crosshead ceases to move while the crank pin is moving across the line of crosshead travel.

Of course, the time consumed by this operation or the length of the cessation of motion may be 1/10 of a second or 100 seconds, dependent, as before stated, on the speed, etc. But, the fact remains that the crosshead is at rest for a certain period of time. As an aid in understanding this, let any engineer turn an engine over slowly until the crosshead ceases to move. Make a mark on the flywheel as when using a tram to find the center when setting the valves, and then turn the wheel until the crosshead commences to move in the opposite direction.

Charles H. Taylor.

Bridgeport, Conn.

In the issue of November 22, under the heading, "Does the Crosshead Stop?" Mr. Fryant wants to know how the crosshead of an engine ends one stroke and begins the next, going back over the same line on which it came out, without coming to rest for a short period of time while the crank is passing the dead center.

Mr. Fryant has brought up a question concerning which there is a conflict between the practical and the theoretical answers. Speaking in a practical sense, the crosshead does come to a stop at the end of every stroke. But, from a theoretical point of view the following explanation will, I think, "put him straight" and be of interest to other readers who have not before given this problem much attention.

If the crosshead should come to rest, assuming that the connecting rod fits the pins accurately at both ends, the only motion that could be given to the free end of the rod would be that of an arc with the crosshead pin as a center. From this it is evident that the crosshead cannot stand still while the crank end of the rod is passing along any portion of the crank circle. The crosshead does, however, reverse its direction and return along the guides. But, if it cannot come to rest in changing its direction, the change must be accomplished by the crosshead describing a loop. The exact shape of the loop depends on the type of engine and the valve setting. The width of the loop is equal to the amount of clearance between the guides and this, of course, is very small in actual cases.

Every operating engineer knows of the pounding that will take place if this clearance is of any appreciable amount and every good engineer calls it his business to see that there is only enough clearance barely to avoid heating. So, for engines that are properly adjusted, the amount of clearance is negligible. The loop is then so narrow that it may be disregarded altogether and the travel of the crosshead may be taken to be back and forth along a straight line with a full stop for an infinitely short period of time at each end of the stroke.

E. O. Smith.

Newport News, Va.

Pumping Problem

I was much interested by the pumping problem presented by J. Ellethorn in the November 15 issue. If the 2½-inch well pipes are connected to the bottom of the 5-inch suction, when the pump is started a vacuum will be created and the following will occur:

When the vacuum reaches 8.8 inches the water from the 10-foot well will reach the 5-inch suction pipe. When the vacuum reaches 10.6 inches the water in the 12-foot well will reach the 5-inch suction line and the 10-foot well will be delivering nearly 100 gallons per minute. If the pump is speeded up until the pipe in the 22-foot well is full, and the water in it neither flowing up nor down, the other three wells will be delivering far in excess of 235 gallons.

If the vacuum is any less than 19.4 inches or that represented by a 22-foot head of water, there will be a flow of water from the 5-inch pipe into the 22-foot well and if the vacuum becomes less than 15.8 inches there will be a flow into the 18-foot well.

It will be seen, then, that this would be a very poor system. The down pipes must be connected through goose necks to the *top* of the 5-inch suction.

Then, when the vacuum reaches 12.4 inches or 8.7 pounds per square inch absolute pressure, the water in the 18- and 22-foot wells will rise in the 2½-inch pipes 14 feet above the surface of the water in their respective wells and the other two wells will be delivering about 235 gallons per minute, 140 gallons from the 10-foot well and 95 gallons from the 12-foot well.

By feeding the 5-inch suction from the top, the pump will receive all of the water discharged from the wells.

A. Langman.

Aurora, Ill.

Removing Clinkers

W. N. Wing, in the issue of November 22, speaks of the use of oyster shells in removing clinkers from a boiler furnace. The theorist with high ideals for the fire room does not consider clinkers as really necessary attributes of a boiler or gas producer; but, like the poor, they are always with us. The little ones do not cause trouble but the big ones do. Coal ash is largely composed of silica and alumina, the same elements which make firebrick refractory, and in this fact lies one of the reasons why clinkers have such a firm attachment for the sides of the firebox. The fusing point of the clinker being only slightly lower than the fusing point of the firebrick, the clinker when plastic readily unites with the fire-softened surface of the brick.

The "why" of the efficacy of oyster and clam shells in assisting in the removal of the clinkers lies in their being composed almost entirely of lime, which causes them to act as a flux for the clinker, reducing its fusing point. A too liberal use will cause the clinker to get lime bound and infusible at furnace heats. Oyster and clam shells, however, are not to be found in the vicinity of most boiler rooms in any quantity, since the present high costs of living have largely eliminated the succulent "raw" from the fireman's bill of fare. This is no reason for despair, however, as a very efficient substitute for such shells, limestone, can be obtained at the nearest iron foundry. This will do the work of fluxing the cinder as thoroughly as it acts as a flux in the cupola.

E. Dixon.

Cleveland, O.

Power Plant Design

In regard to the letter by Waldo Weaver in the November 29 issue, I think that it would indeed be an ideal power-plant layout that could not be improved upon, and I believe that it is true in many instances that faulty arrangements are made as the results of the notions of some doctor, lawyer or other person having no mechanical inclinations whatever who happens to be the fond owner or an influential director.

I was once called upon to operate a new plant in connection with a woodworking shop which had been laid out under the direction of one of the directors of the concern, who was a lawyer by profession and who thought that he knew quite a good deal mechanically. A consulting engineer with no practical experience but having a college degree was employed to work out the plans because, I suppose, being a well educated, cultured professional man who knew how to charge for his services, his personality appealed more strongly to the lawyer than did that of a grimy, greasy operating engineer.

The plant contained an old-style Wetherill-Corliss engine of about 150 horsepower capacity, a second-hand engine but in good order. The flywheel was 12 feet in diameter and carried a 24-inch leather belt driving a 5-foot pulley on a jack shaft. The center of the jack shaft was only 17 feet distant from the center of the engine shaft, and there was, consequently, but little belt contact on the 5-foot pulley. The engine was set up on a subbase of concrete 16 inches above the floor level, and a short step-ladder was required to enable one to reach the throttle.

The boiler grates were placed level with the floor of the boiler room, with a pit 2 feet deep and 4 feet wide, covered with iron slabs which could be removed when it was necessary to clean the fires or the ashpits. These pits extended to the outside of the building and comprised the air ducts to the furnaces.

The plant was in operation when I was engaged to take charge as operating engineer. But, although the machinery in the shops was not more than half installed, the engine would not handle the load required of it at the time. The main belt gave considerable trouble by slipping and it was hard to keep up the steam pressure.

After my first inspection of the plant I told the manager that I was going right back to my old job, that I had never seen so ridiculous an arrangement before. He wanted to know if I could make some changes in the arrangement so that the plant could be operated. He said that he had expected it to be a modern, uptodate plant, that an expert mechanical engineer had been retained at considerable expense in anticipation of this, but upon giving the plant a trial he was sadly disappointed in the engineer's work.

I told him that I would stay if they would remodel the plant, and after a consultation with the owners, I was told to go ahead with my changes.

I had the concrete subbase under the engine removed and the engine set 4½ feet further away from the jack shaft, making a total distance of 21½ feet between centers. A new 6-foot pulley was substituted for the 5-foot one and the engine speeded up from 60 to 72 revolutions per minute. I had the floor of the boiler room lowered 20 inches and a truck made with an iron spout at one end from which the fuel could be fed to the fires very handily.

After these changes had been made, the plant was operated with greater satisfaction, and when the full equipment of machinery was installed, there was enough power available for all requirements.

Consulting engineers say the operating engineer is not competent to plan, because he is prejudiced, narrow-minded and lazy. He is apt to want things too handy in order to make his routine work easier, and he does not take into consideration the cost of things. But, however that may be, I have had experience enough to know that the more handily and more conveniently a plant is arranged, the more efficient will the human element be.

Of course, due consideration should be taken to the costs, and it is not impossible for the operating engineer to equip himself with the ability of estimating costs and the profit or loss to be incurred by the installation of this or that machine or appliance.

The operating engineer is not, perhaps, in a position to acquire so wide a knowledge of all kinds of power plants as the consulting engineer, for the reason that he does not change jobs so often. I believe, however, that it is advisable for operating engineers to specialize in some particular line as, for instance, factory work, electrical-power work, hotel work or whatever may be his choice. Then, let him study it faithfully, every phase of it, until he becomes a master of his specialty. Having done so, I believe that no man on earth could be more competent to lay out plans in his line of business than he.

C. F. Clark.

Hartwick, N. Y.

Fusing Temperature of Ash

The article on "The Fusing Temperatures of Ash," by Messrs. Bailey and Calkins in the issue of November 8, deals with a subject which is for many plants a vital one in the selection of a fuel. Many coals which would otherwise be entirely satisfactory both from the point of heat value and from the point of price, give results which are an absolute failure from an operating point of view when they form clinkers to such an extent that it is entirely impractical to continue their use.

I have seen many tests made with coals of this nature. While these coals would give the lowest cost for evaporation, on account of the complaints of the firemen it was much more agreeable to have a little higher cost and save the excessive hard work. I have never found that a clinkering coal can be burned successfully on the flat grates which we use, which are of the shaker type. As they are of large area (80 square feet), they are difficult to clean and costly as well, for there is always a considerable drop in steam pressure when the doors are opened at time of cleaning.

Apparently, when it becomes necessary to burn a clinkering coal, it is necessary to turn to some of the mechanical stokers for relief. A chain-grate stoker will afford this relief very readily, for all clinkers and ash are continually carried to the pit and dumped and there is no clinker remaining to impede the air flow when fresh fuel is applied. The gravity-feed stokers usually clog more readily and if the clinker is of low fusing point it is apt to run down on the bars and stick. It is probable that stokers will be the salvation of many plants which are at the present time hand fired, when the availability of coal with good nonclinkering qualities is lessened, due to the exhaustion of many of the present seams.

At times when we have had in coal for test that clinkered badly, we have attempted to avoid this by mixing this coal with other coal and while it does lessen the trouble in many cases to just that extent to which the bad coal is diluted with good coal, yet we have noticed cases where adding a coal which only formed medium clinkers to one which formed no clinkers would effect a combination making very bad clinkers. This seems to be due to the fact that the nonclinkering coal had a very high fusing point of ash and the ash as fused into clinkers only formed into small nubs which could easily be shaken through the grates. On adding the coal which clinkered slightly, an element was introduced which had a low fusing point, the result being that the low fusing point ash flowed around the small lumps of the infusible clinker and ran it all together into one large mass. It seems that mixing a coal will never prove to be a remedy and, although this subject has been taken up with a number of different authorities, their opinions have always been to the effect that it is impossible to eliminate clinkers by mixing the coal. If lime, silica or some other ingredient could be added to a clinkering coal and entirely eliminate this trouble it would be a godsend to the engineer suffering from such a trouble, but nothing of this kind seems ever to have been discovered, although we have no doubt but that some New York cobbler may yet discover an alum wash or some other fake which he will sell as suitable for this purpose.

Referring to the point, which is so well expressed, that the percentage of sulphur was no indication of the clinkering tendency of a coal, we have also found this to be a fact, although it seems strange that it should be so, for the greater per cent. of sulphur in coal is combined in the form of pyrites, and the iron in pyrites would be just the material to form a hard clinker. Notwithstanding this point, it seems very inconsistent to claim that there is any intimate connection between the presence of sulphur and clinkering properties.

Within two years, tests have been conducted on over fifty coals and from these we have selected a number of samples, showing the analysis and giving the extreme range in sulphur contents and clinkering properties. Unfortunately, these tests were not conducted in such a thorough manner that a record was kept of the actual percentage of clinkers; the only record kept being whether or not they did clinker when fired in actual use.

In the table there is one item in regard to clinkers that must be explained. Here, when we express the fact that there are no clinkers, we mean that there are none which will not pass through rocking grates, which open to 1½-inch clearance when shaken, and this means that the clinker can be removed without slicing the fire.

COAL CHARACTERISTICS.

Coal.	Coal Field.	Heat Value.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.	Character of Ash.
Pocahontas, nut	W. Va.	14,700	19.63	74.06	6.31	0.63	Clinkering iron ash.
Steam, R.O.M.	W. Va.	12,400	32.88	50.58	16.58	0.60	Medium quantity of clinkers.
Cabin Creek, N. P. & S.	W. Va.	13,406	33.67	56.01	10.32	0.82	Clinkers very badly.
Queen, N. P. & S.	W. Va.	12,440	32.77	52.47	14.76	1.09	No clinkers.
Pocahontas, R.O.M.	W. Va.	14,198	22.34	69.71	7.95	1.22	Ash heavy and some clinkers.
Jackson, N. & S.	Ohio.	12,149	27.83	56.75	15.42	1.62	Abundant ash, no clinkers.
Lookout, N. P. & S.	Ky.	12,526	33.69	54.73	11.58	1.55	Light-weight ash, no clinkers.
Blue Gem, N. P. & S.	W. Va.	11,282	30.73	46.81	22.46	1.47	Heavy, hard clinkers.
Franklin County, N. P. & S.	Ill.	12,065	33.66	51.51	11.84	2.08	Ash heavy, clinkers very badly.
Glen Alum, N. P. & S.	W. Va.	13,340	35.66	54.68	9.66	2.09	Light red ash, no clinkers.
Gas coal, R.O.M.	W. Va.	15,200	34.54	55.18	10.28	2.68	No clinkers.
Iroquois, N. P. & S.	W. Va.	14,100	34.95	57.46	7.65	2.90	No clinkers.
Sullivan County, N. P. & S.	Ind.	11,162	39.53	44.75	17.72	3.01	Heavy ash, and few hard clinkers.
Nut, P. & S.	W. Va.	11,527	33.65	57.09	9.26	3.44	Light white ash, no clinkers.
Staunton, washed nut	Ill.	12,600	41.72	48.06	10.22	3.16	Heavy ash and no clinkers.
Gaylord, R.O.M.	Ohio.	13,100	38.19	48.54	13.27	3.82	No clinkers.
South Illinois, N. P. & S.	Ill.	11,440	35.22	49.38	15.40	3.88	Ash, slag and clinkers very heavy.

The examples mentioned in this table have been selected from a long list of test coals and it has been done with the intention of showing how little connection there is between the sulphur contents and the clinkering tendency. It will be noted that with a very low sulphur content and a low percentage of ash, clinkers were obtained and also that with a very heavy ash and high percentage of sulphur, no clinkers were obtained. It, therefore, appears that the only possible way to determine the percentage of clinkers in an ash is by obtaining its fusing temperature and knowing the temperature of the furnace; but as few are in a position to obtain data of this kind, it is reasonable to suppose that sole reliance in matters of this kind must be based upon tests actually conducted with the boiler.

J. V. Hunter.

Fort Wayne, Ind.

Uncle Pegleg on Velocity

Uncle Pegleg's explanation of the problem of the pitched ball, printed in the December 20 issue, is excellent—mathematically. It is evident that all the pitcher really does is to impart *relative* motion and energy to the ball. Probably all the readers of Power realized the natural fact promptly.

Now, Uncle Pegleg has reconciled his equation based upon this relative motion between pitcher and ball with the same equation applied to the rotative motion of the earth. But, is the same equation applicable in both cases?

When the pitcher throws the ball he imparts a *forward* motion to the ball and —assuming the earth to be previously at rest—a *backward* motion to the earth. It is because the pitcher, with his muscular force, gets in *between* the ball and the earth, pushing one in one direction with his hand and the other in the opposite direction with his feet, that he develops energy. This energy is embodied in the relative motion between ball and earth.

Now comes Uncle Pegleg and says, following the tenets of orthodox mathematical mechanics, that since the earth is also imbued with a motion of its own, twenty-five times as great as that of the ball relatively to it and in the line of the ball's motion, the ball must receive and the earth must impart

$$2 \times 25 = 50$$

times the pitcher's energy. The question is: Is this true to nature? Is this mathematical computation a natural parallel with the other, although the same equation was used?

When we ascribe a velocity of 1500 feet per second to the earth's surface, what do we mean? What is this velocity measured from? From a purely imaginary, massless, geometric line drawn from the center of the earth to—to—to what? It cannot be the center of the sun, for the sun itself moves, enough so that we should lose in a year some 85,000 times 1500 feet. If the "fixed" stars are to be relied upon, which one shall we chose? For they, too, are all in motion.

Of course, in practical astronomy an average of these stellar motions is struck which is amply accurate for all purposes. The reason for raising this objection is to make clear that the earth's rotative motion is one relative to something quite intangible, massless and indefinite—something which the pitcher could not push against hard enough to develop a particle of energy. It is a motion of the very greatest importance in the kinematics of astronomy. It is one which has no part whatever in problems of mechanics and energetics.

If the force developed by the pitcher had acted or reacted upon any mass which was stationary relative to the heavens, then it would have been proper to compute energy by using this force and the earth's motion. Should some comet happen along and wipe off a portion of the earth's crust, as one did in Jules Verne's "Meridiana," then the rotative velocity of the earth, not relative to the sun, or to the fixed stars, but to the comet, would be a very real thing, and positive or negative energies computed upon it would have a very real existence. But, as it is, the introduction into a problem of an imaginary, mathematical fund of energy, based upon a massless geometric motion, is wrong and misleading. The mere fact that careful mathematics will bring in an equal fund of imaginary, geometric energy on the other side of the equation, making the problem "come out" all right, as Uncle Pegleg has skilfully shown, makes it only the more misleading, because it looks so accurate.

The fundamental rule in mechanics which is sought to be impressed is that no *force*, no *velocity* and no *energy* may ever be introduced into a problem in energetics, unless they exist *between two masses*. In studies of pure motion, such as problems of gear teeth, valve gears or astronomical orbits, it is proper to mention velocities without their massive sources. In studies of pure force, as in parallelograms of force, it is proper to mention forces without their massive sources. But while this is being done the student should be frequently reminded that neither force nor motion can exist except *between masses;* for the present lax methods have led to a very widespread idea, prevailing sometimes with very high authorities, that force and motion can and do exist otherwise than *between* two or more masses—which is a fundamental error. In the study of energy neither force, motion nor energy should ever be mentioned as divorced from mass at both ends.

S. A. Reeve.

New York City.

Boiler Explosion at Weatherly, Penn.

> *A lap seam of a return tubular boiler failed due to an old crack that extended about two thirds the length of the sheet and over half way through the plate. The explosion killed two men.*

The silk mill of the Reed & Lovatt Manufacturing Company, Weatherly, Penn., was the scene of a disastrous boiler explosion on December 12 shortly after the plant had been shut down for the day's run, when one of a nest of five boilers exploded, instantly killing two men.

According to the clock in the engine room, which was stopped, the explosion occurred at 6:08 p.m., after most of the five hundred operators in the mill had left for the day, and this undoubtedly reduced the death toll, as many of them just before the closing hour had been in the habit of congregating directly back of the boiler room, which was partitioned from the factory by an 18-inch brick wall. Had the boiler exploded ten minutes earier, many of the workmen would have undoubtedly been killed.

The five boilers in this section of the steam plant were made by Samuel Smith & Son Company, and the boiler that exploded was located in the center of the nest. It is stated on good authority that this is the first boiler that the makers have lost due to an explosion.

The force of the explosion entirely wrecked the boiler house, and knocked down a section of the 18-inch brick wall between the boilers and the mill. The side of the engine room next to the boiler was demolished, and the engine room was partly filled with brick and wreckage. From the violence of the explosion it is evident that the boiler was full of water at the time, although there is the usual talk of low water as the probable cause.

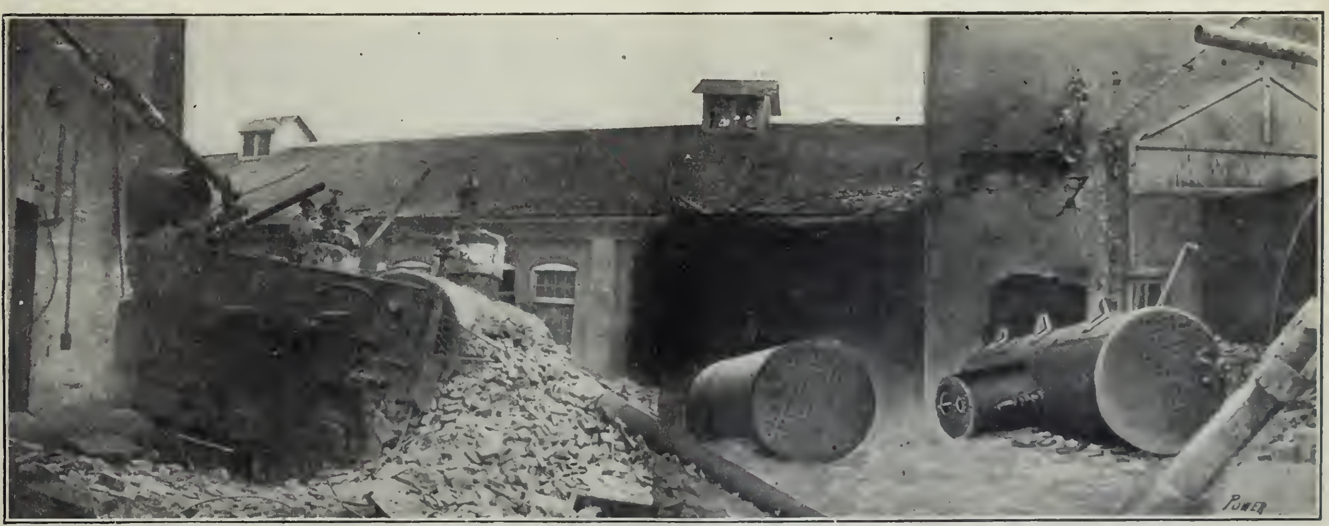


Fig. 1. View of the Wrecked Boiler House. Back of the Hole in the Wall is where the Operatives Were Assembled Ten Minutes before the Boiler Exploded

The day fireman had left after being relieved by the night fireman, Richard Beers, who was killed, as was also Michael Moony, the chief engineer. Beers was found dead under a pile of brick at the rear of the boilers, close to the feed pump. Moony was found dead under a heap of brick and timbers at the door leading from the engine room into the boiler room. He was dressed ready for going home and was evidently only waiting for the return of Beers to announce that his end was shut down and bid him good night. Both men were badly scalded and bruished.

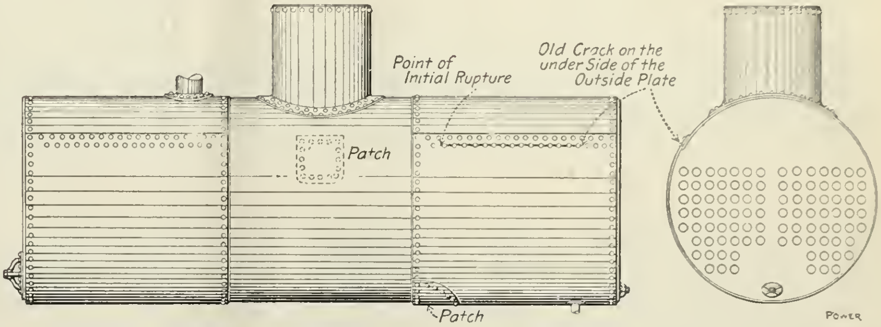


Fig. 2. Showing Location of Crack and Patches

Two of the boiler settings were entirely demolished, and two boilers were left on badly wrecked settings. A general view of the wrecked boiler house is shown in Fig. 1. The hole in the wall is directly in front of the place where the mill operatives congregated waiting for the whistle. The force of the explosion was up and sidewise, toward the two boilers next to the engine room.

Details concerning the boiler that exploded are as follows: It was put in place in 1888 and has been in service 22 years. It was a longitudinal return-tubular boiler 16 feet long and 66 inches in diameter. The plates were of ⅜-inch steel, but the tensile strength is not known as no stamp could be found. Both heads were made of 9/16-inch stock, and the rear head was braced by twelve ⅞-inch round crow-foot braces above the tubes and four braces of the same type and size below the tubes. The front head was strengthened by twelve braces above the tubes of the same size and spacing as those used on the rear head, but there was no bracing below the tubes in the front end of the boiler, as the manhole was reinforced. The other four boilers were all braced in the same manner and were of the same construction.

A 36-inch dome was attached to the boiler. It had a cast-iron head 1$\frac{15}{16}$ inches thick that was reinforced by a rib 3 inches deep and 1⅛ inches wide, cast on the inner side. The flat surface of the head was not braced, although the dome was stayed to the boiler sheet by six [illegible] braces. All of the five return-tubular boilers in this section of the plant were equipped with a dome of similar construction.

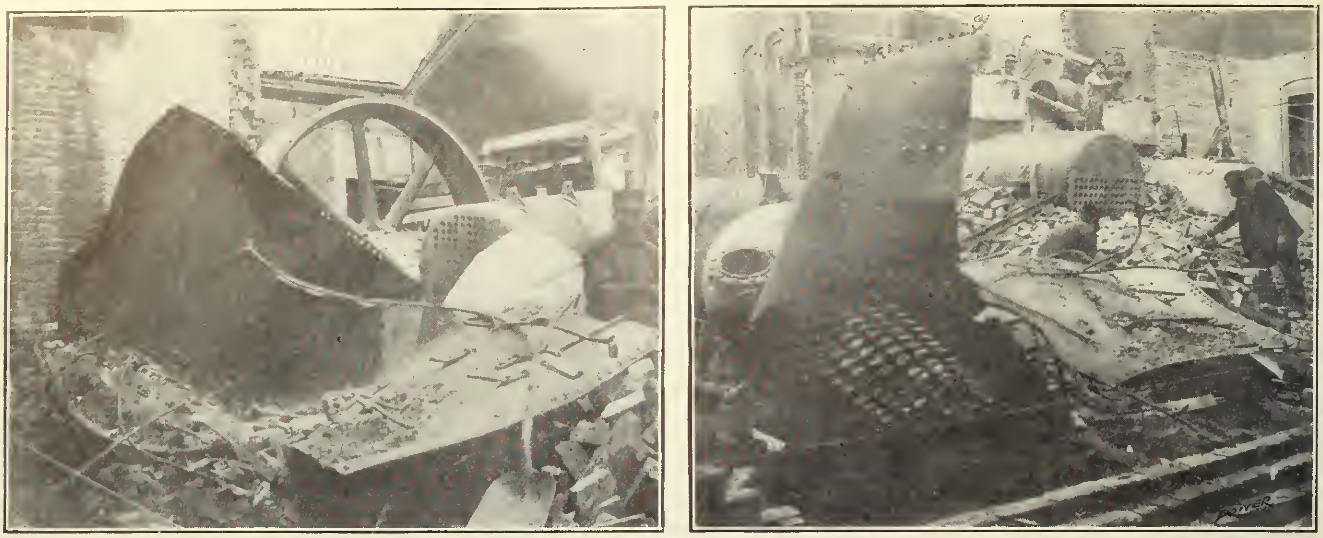


Fig. 3. Two Views of the Rear Head and Fractured Sheet

With two exceptions the braces between the heads and the sheets held, the rivets in all the others being sheared where the braces parted from the plate.

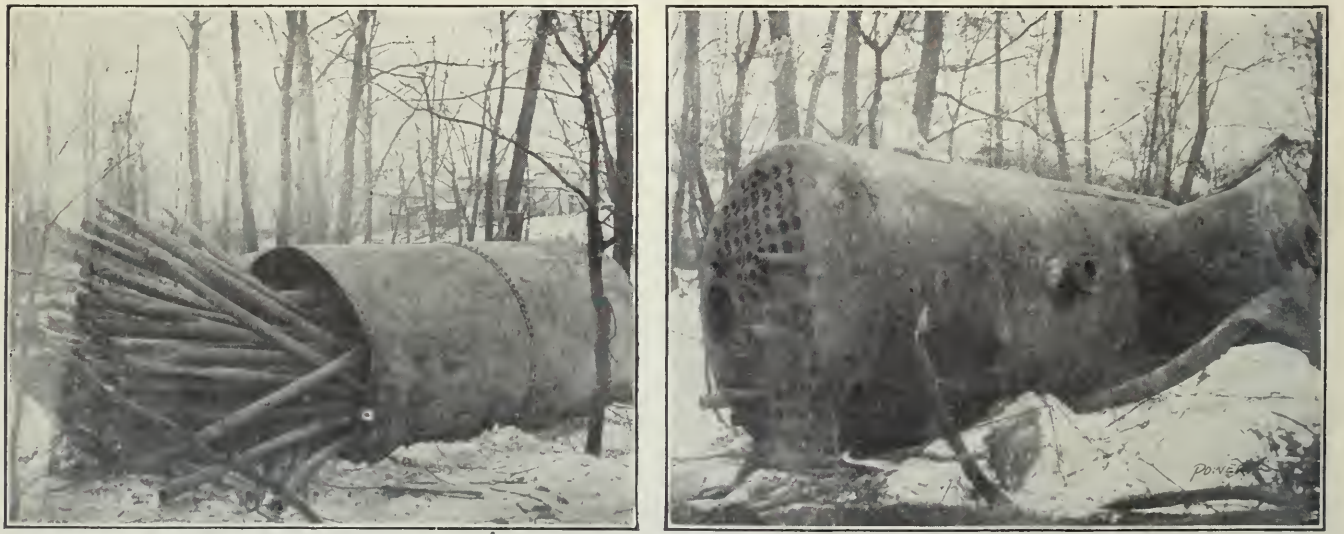


Fig. 4. Views of Main Sections of Exploded Boiler

The explosion was directly due to an old crack that extended from one-half to two-thirds through the plate in line with the second longitudinal row of rivets of the rear sheet. This crack was on the under side of the outer plate and was of such a character as to escape attention by the boiler inspector who examined the boiler on May 30 of this year. It extended from a point about twelve inches from the girth seam nearly to the rear head. This plate was considerably thinner at the inner end of the longitudinal joint seam than at the outer end and it was at this point that the initial rupture occurred, as indicated in Fig. 2. This sheet also had a patch 15 inches long by 24 inches wide located on the bottom of the plate. There was also a dry patch on the second sheet of the same boiler, as shown by the dotted line in Fig. 2.

After tearing along the lap-joint seam through the rivet holes, none of the rivets giving way at this joint, the sheet opened up one-half of the diameter of the shell, as shown in Fig. 3, the rivets shearing in practically every instance at both the head and girth seams. Many of the rivets indicated that the holes had not been in line when the boiler was constructed, but whether this is so or the ogee appearance of the rivet was the result of the plates being stripped from the others, is difficult to determine. In places the plate showed slight signs of brittleness and several cracks were visible around the rivet holes, which were of 2½-inch pitch. There was no scale to speak of in the boiler, either on the shell or tubes.

The force of the explosion blew the rear head and attached plate a few feet back from its original setting, every tube pulling out of this tube sheet. The other two sheets, front head and tubes were blown several hundred feet into the woods back of the plant, mowing down the trees that happened to be in their path.

Although several bad dents appear in the sheets and head of this section of the boiler, due to striking rocks in their flight, the plates do not show signs of fracture.

Many of the ninety-six 3-inch tubes were worn very thin, especially at the end that pulled out of the rear head, and a number of the lower tubes were badly pitted. The pressure carried on the boiler was usually between 85 and 90 pounds per square inch. A pop-safety valve that has not been found at the present writing was fitted to the boiler. All of the boilers were under steam at the time of the explosion and were connected to the same steam header. The boiler was insured by the Hartford Steam Boiler Inspection and Insurance Company.

The foregoing data and the accompanying photographs were obtained through the courtesy of Raymond Thorp, agent of the mill.

Notes on Power Station Economy*

By H. P. Wood

In discussing the economical operation of a 30,000-kilowatt turbine power plant the author shows the importance of apparently small items which in the aggregate make for either great economy or great waste.

It is the purpose of this paper to give members of our section an idea of the various things which enter into the economical operation of the Gold street power station.

FUEL

The present cost of fuel and its continual rise in price give a constant struggle with the problem of fuel economy. This item in a properly designed and conducted plant should amount to three-quarters of the total cost per kilowatt. The Gold street plant contains no complications or intricate devices and is so arranged that by careful operation very economical results can be obtained.

Starting with the boiler room, there are a number of factors which must be closely studied. First one among them is the burning of the fuel, consisting of a mixture of four parts No. 3 buckwheat and one of bituminous coal, which experience has demonstrated is the best suited to the conditions. This mixture is burned at times at the rate of 26.6 pounds per square foot of grate per hour, giving a furnace temperature around 2700 degrees which, though satisfactory as to evaporation, is hard on the furnace walls and arches.

UPKEEP

Another important point to be considered is the keeping of the boiler clean, both the inside and outside of the tubes and drums. The sulphur from the coal burning on the outside of the tubes and drums requires that they be cleaned daily by blowing with a steam jet. Though expensive, the cost of this blowing is small compared with the expense necessary to keep them clean on the inside. The boilers are blown down daily, and once in three weeks are emptied and filled with water from the condensers. If scale is found 1/32 inch or more in thickness, a cleaner is used to remove it. The seven lower rows of tubes are usually cleaned every six months and the others once a year.

FEED WATER

The feed water contains 1.8 grains of scale-forming matter per gallon, which means that the eleven and one-half million gallons of water evaporated deposit 1.3 tons of scale per year in each boiler which must be taken care of if any degree of economy is desired.

It is necessary to keep down the amount of air which passes through the fire. If any holes are allowed in the fire, or air leaks in around the brickwork, it is quite possible to lose 25 per cent. This is one of the easy things to keep track of, a CO_2 gas recorder constantly keeping a record.

Then comes the flue temperature, which should run from 500 to 600 degrees. A temperature above or below this means that one of several things has gone wrong. In case it has gone up, probably the tube brick in the baffle walls has come down; if it goes down, it usually means that the setting is leaking, or the fireman is not keeping his fires properly covered.

And next comes the question of superheat. There is approximately a gain of 1 per cent. for each 12 degrees that the steam is superheated.

STANDBY LOSSES

In the commercial operation of a plant under varying load conditions, such as are met with in our system, it is necessary to maintain steam pressure on boilers which would otherwise be idle, in order to provide sufficient reserve capacity. This represents 6.6 per cent. of the fuel burned. This requires fuel, though not adding in any degree to the output.

It seems astonishing that twenty boilers are used during the peak and only ten the rest of the time; that eleven boilers would give the same output if the load was constant.

The proportion of standby loss depends upon the load factor, duration of the peak, number of units, design, construction and methods of operation.

The temperature of the air going to the fires has a marked influence on the operating costs, a reduction of 2.7 per cent. in the amount of coal being made by a change in the location of the blowers by which warmer air was secured than formerly.

FEED WATER

The next important item to be looked after is the feed-water system. There was originally installed a closed-heater system. With this we were unable to maintain a temperature of more than 170 to 180 degrees, with the loss of approximately 12 per cent. of the total returned auxiliary steam. It was decided to install the low-pressure or open-type heater. The hotwell water from the condensers and the necessary makeup water are supplied directly to the open heater, from which it is pumped into the feed-water mains by four-stage centrifugal pumps. The saving of the exhaust steam was one of the factors which decided the installation of turbine-driven auxiliaries.

COAL AND ASH HANDLING

In the coal-handling system there is little to be saved. But by the equipment of one of the two towers with a one-man control system it is expected to reduce the cost of coal handling about 14.5 per cent.

Experiment has proved that two laborers can keep a large section of the ash cellar clean, which leaves practically little chance to make any marked saving, as the installation of an elaborate system of ash-handling apparatus would increase the upkeep more than the cost of labor saved.

TURBINE EQUIPMENT

There are in the plant four turbines, totaling a capacity of 35,000 kilowatts, based on the highest temperature at which they will operate continuously.

The day load runs from 12,000 to 14,000 kilowatts, which divides up very nicely. The night load is much less. If our units were divided so as to give a full load for two small or one large one, we would be able to save 5.7 per cent. of the present operation cost. This cannot be determined exactly, owing to the annual increase in the load. While the present units do not divide nearly so well as they might, in one or two years from now they will be right. Then, again, in one or two years more we will be back to the same conditions which face us at the present time.

With the present load factor, it requires two units for the ordinary day load, and three for the peak. After careful tests the most economical point of operation has been determined, and when the load will allow it, the turbines are run at that point, except during the peak. When two turbines are operating at approximately full load, the third turbine is kept hot and the auxiliaries running so that it may be brought up to speed and put on the line. By this method we save 5.6 per cent. over the former method of operation, which was to run three turbines when there was a load for two.

VACUUM

Another important item is vacuum. This required constant vigilance and the average vacuum is not what we hope to obtain. The loss of 1 inch of vacuum on one of the larger units means $14,700 per year. This item is of such importance that it calls for a great deal of time and consideration.

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*Abstract of a paper presented at the December meeting of the Brooklyn section of the National Electric Light Association.

Condenser Tubes

There has been a great deal of annoyance and expense resulting from the failure of condenser tubes. If all of the tubes could be kept tight so that all of the condensation could be returned to the boilers, $28,000 per year would be saved. In only one condenser have the tubes been tight for any reasonable length of time. In this one they have given satisfactory service for four years, only 12 tubes having to be replaced.

It is not possible to understand why they should stand in one condenser and fail in all of the others.

There are two losses which occur in handling the circulating water: one, the amount of steam required to pump an unnecessary amount of water; the other, the lowering of the hotwell temperature.

From experiments it has been found that where the temperature should be 100 degrees at 28 inches of vacuum, it is often found to be 85. This represents 15 B.t.u. lost in each pound of water, or for a year, 1630 tons of coal.

Discussion

Mr. Pope, referring to a curve showing the total kilowatt-hours, said that makes 1131 electrical horsepower output for each boiler, where the rating is 730.

This looks like perpetual motion to me, and I would like to have it explained—1131 horsepower output from a rating of 730 horsepower.

Mr. Buxton: We are greatly indebted to our operating engineer for the clear and concise exposition he has given us. It covers the ground, instructs, entertains and is of permanent value. It is worthy of wide distribution.

Mr. Helwig: It is only within the last few years that operating engineers have come to the conclusion that if there was any economy to be obtained in the operation of a station, it is in the boiler room. Not so many years ago the losses of the boiler room amounted to 30 to 35 per cent., while the losses now reach about 25 per cent. at the most. The boiler efficiency has increased from about 55 per cent. to 65 per cent. in regular station operation.

The losses were decreased by keeping tab on the flue gases both temperature and per cent. of CO_2, and sampling ashes for unconsumed coal. The loss up the flue, due to temperature, varies directly as the temperature of the gases, and may be caused by poor bridgewalls, defective baffles, tubes dirty internally and externally, or bad setting and bad firing, high flue temperature indicating bad baffles and bridgewalls, and low temperature, leaky settings and bad firing. The percentage of CO_2 is an indication of the efficiency of the furnace. About 4 per cent. CO_2 will mean a loss of 42 per cent. up the stack, and 14 per cent. CO_2 means 12.5 per cent. loss up the stack. The average per cent. of CO_2 for daily operation is between 10 and 12 per cent. at the present. Years ago 7 to 9 per cent. was considered as a fair result. The maximum CO_2 obtainable for perfect combustion is 20.8 per cent., which means using all the oxygen in the air. The percentage of CO_2, where the boiler is in good shape, depends entirely upon the fireman. If he permits his fire to get full of holes he reduces the CO_2 and the initial temperature. As high as 17 per cent. of CO_2 has been obtained by choking off the flue damper and air supply and firing light, but in doing this the capacity of the boiler was cut down. Good, high furnace temperature was obtained, but at a loss of capacity. The best results are obtained with an excess of air of about 100 to 120 per cent. Fuel losses in refuse, which used to be about 14 per cent., are now reduced to from 5 to 8 per cent., due to more careful firing and closer study of the grate problem.

Another point that may be brought out is that the ratio of the heating surface to grate area has been reduced from about 90 square feet of heating surface to 1 square foot of grate to about 45 square feet of heating surface, and in some cases as low as 40 feet, in some of the stations around New York.

Mr. McGee: It is stated that a furnace temperature of about 2700 degrees Fahrenheit was obtained. At what part in the furnace was this? Is it possible to get a higher furnace temperature with the present methods of firing? If it is possible to get a higher temperature, will the air circulate fast enough to prevent burning or blistering the tubes? Can furnace walls and arches be built that will not melt at the higher temperature? The present temperature destroys the furnace brickwork very rapidly.

Will the circulation in the superheater be rapid enough to maintain 100 degrees superheat without raising the pressure?

Professor Maxwell: I could give you some few personal experiences which bear immediately upon the questions brought out in this paper. About five years ago I was called upon to investigate the reason why four large tubular boilers would not supply sufficient steam to operate a plant successfully. In the boilers soft coal was piled into the furnace right up underneath the tubes, making a thickness of fire of from 20 to 30 inches. Experience, years before, had taught me the value of fires as thin as can be carried. I cut the fires down to about 10 or 12 inches. The result was, during a 24-hour test, they had all the steam they wanted.

Upon one occasion I performed a series of tests for the Navy department to demonstrate the loss of economy due to forcing a boiler. Under natural-draft conditions for every pound of coal burned nine pounds of water were evaporated. Under forced draft 60 pounds of coal were burned per square foot of grate, and the evaporation dropped to 6½ pounds of water per pound of coal.

The temperature of the flue should be between 500 and 600 degrees.

Two years ago I was called to test four boilers, guaranteed 70 per cent. efficiency. I ran preliminary tests before the final. I found the temperature of the flue gases was 850 degrees. On the second preliminary test 65 per cent. efficiency was obtained, and the temperature of the flue gases dropped to a trifle under 600 degrees.

The guarantee required 10½ pounds of water to be evaporated per pound of coal from and at 212 degrees. I determined to use 0.28-inch draft and put a differential draft gage in the ashpit of each of those boilers, marked the furnaces with chalk and gave instructions that no fireman should open a door or fire without orders. The draft gages were watched very carefully, and the instant that one gage dropped one-hundredth of an inch, it would signify that the fire had burned a hole in the coal bed. The fireman would open the door and would see a black spot in the fire, on which he would put one or two shovels of coal, just enough to cover the spot. In that way the minimum amount of coal that the grates would burn to give the evaporation was found. That was an unusual condition but made an efficiency of 85 per cent. in an eight-hour test.

In the turbine station at Gold street, I think the pressure is 180 pounds, and the temperature corresponding to that pressure would be about 380 degrees. Well, for 100 degrees superheat, it would be 480 degrees.

Mr. Johnson: There is one point in Mr. Wood's paper that fits in with my own experience so well that I could not help thinking about it all the time, and that is the matter of condenser-tube corrosion. Mr. Wood had one condenser run four years without giving any trouble, and others in which the tubes were constantly failing. My experience was just exactly the other way. In seven condensers I had only one that gave trouble; the others never bothered.

At the time we were having the most trouble, there was a great deal of talk about electrolysis, and the electrical department was blamed. We said that there were grounds or stray currents, or something that came into that condenser, or rather the tubes, with the water, and went away with it, and when the current left it took some of the tube with it.

They said that there were no grounds inside of the station, and several other things. The assistant engineer came to me one day and said, "I want to put up a job on the 'super.'" I said, "What is it?" He replied, "Well, I want something to be the matter with No. 5 for a week." "All right," I said. So he reported that we were waiting for some little detail; that the engine could be run, if it was absolutely necessary, but wanted to have the other units in the plant used if possible. For an entire week No. 5 auxiliaries were run, and then the condenser was opened, tested and found to be in perfect condition. The imaginary parts had not arrived; and at the end of another week everything was apparently all right. The imaginary repairs were completed, and the engine was reported to be in good order. It was started in. In four days there were two tubes down. They were put in, and more imaginary repairs were put on the engine. For two weeks the pumps were run, and there was absolutely no sign of any decay anywhere. The load was put on the engine again, and in less than a week we had to put in more tubes, and that thing continued for two or three years; and I want to say to Mr. Wood that I believe he should accuse the electricians and put the blame for the deterioration on them.

Mr. Kemble: I have not checked Mr. Pope's figures but if I may draw from my own experience for a moment, I will say that in Boston we had two pair of 500-horsepower boilers, a little old style—in fact, considerably older than those at the Gold street station. Five hundred horsepower requires the evaporation of 15,000 pounds of water per hour. There having been a little bit of discussion between the master mechanic and me as to what those boilers were actually doing, we tried out some tests, and found that regularly during the peak loads those boilers, which were supposed to evaporate 15,000 pounds of water per hour, were evaporating from 22,000 to 24,000 pounds, over 50 per cent. overload. I am a little inclined to think that the discrepancy between 730 boiler horsepower and 1130 electrical horsepower is simply due to the elasticity of the boiler rating.

As regards what Mr. Buxton said, I know that Mr. Wood, in writing it, attempted particularly to make it interesting to the nontechnical or semi-technical members of the section.

In regard to the remarks of Mr. Helwig, it is possible to increase the CO_2 as he mentions by shutting down on your furnace until you get a maximum percentage of CO_2 but have no capacity. It is also possible to increase your CO_2 remarkably, and a trick practised sometimes by firemen seeking a record is to close down on your air supply, raising your CO_2 percentage, but sending a large proportion of unburned gases up the stack. So the CO_2 percentage cannot be taken as an arbitrary value of the boiler, unless checked up by some other method.

Mr. McGee brings up the question of higher temperatures. That depends somewhat on the point of location of combustion. An extreme case would be if you had the combustion all on one point. No material would stand the temperature. Now, by a proper proportioning of furnace, combustion can be spread over the whole area without abnormally high temperature at any one point.

As to Mr. McGee's question about the circulation, I would say that boilers similar in type to those in the Gold street station have been forced up to 40 pounds of coal per square foot of grate, which means an increased output, without trouble from the formation of steam pockets or blistering of the tubes. As regards the superheat, 100 degrees, or even as experimentally tried, 180 degrees, does not seem to produce undue deterioration of the superheater, providing there is not too much sulphur or other impurities in the coal.

Regarding Professor Maxwell's remarks as to evaporation, it may be of interest to know that some of the new boilers which are being furnished for new warships are furnished on a guarantee of 11½ pounds of water evaporation per pound of combustible, with 2 inches of air pressure under the grates.

I was very much interested in his account of the beating of the specifications. It reminds me of that little story about a fellow who wanted to get into the navy, wanted to get into Annapolis, and was about three-sixteenths of an inch short; and so he went out and got one of his friends to swat him on the head with a club, and the bump passed him.

Perhaps in illustration of the method of figuring out very closely the results which can be obtained in practice I might mention the old navy trial-trip method of reporting the horsepower developed to four or five places of decimals, and the speed obtained by the ship to six places of decimals, when the probability was that if they came within 75 or 100 horsepower they were doing very close work. I could not help thinking, when he was speaking of the 85 per cent. efficiency, whether if the contractors had been offered a little bonus, he could not have done better than that.

As to his question of numbering the furnace doors, the practice was started in England, if I remember rightly, in 1889, the practice of time firing in the trials, and it was first put into practice in this country in the trials of two Russian warships, which were built by Cramps, and tried in 1899 and 1900. I had the fun of training the firemen for these trials. The engineer's department decided how much coal had to be burned under each boiler to produce the volume of steam which would produce a given speed of the ship, and clocks were set in each fire room, with speed buttons placed around the circumference of the clock similar to those clocks in a billiard room—if any of you gentlemen go to such places—where they could move them around and get an electric contact every so many minutes to ring a bell. On receipt of orders from the engine room that such a speed was to be made, the clocks were set so as to ring a bell every so many seconds or minutes, as the case was, depending on whether the speed was high or low. When the engine-room artificer ordered certain doors—they were opened, and so many shovelfuls of coal were spread on the fire. In the preliminary trials, while checking up the engineering department's calculations it was only found necessary to change one rating, and the speeds were carried through by that time firing system; and not only that, but the standardization trials, which consist of a series of runs at different speeds, were all carried through on that prearranged coal consumption; so, when a certain speed was wanted, the engineer did not open the throttle and yell for more steam; he simply notified the boiler rooms, and the boiler room provided the steam.

In regard to the remarks made by Mr. Johnson, I have no doubt—and I could not help wondering all the time if he had a piece of wood in his pocket which he touched when he spoke of having only one condenser trouble out of seven. Certainly the scheme of indicting the other fellow for petit larceny when you lose a tube is worthy of imitation, provided you can convict him. I presume Mr. Wood will be glad to use the method of charging it on to electrolysis, because we don't have any street railway connected on to the line system, and electric-light current, as everybody knows, never does any harm.

It occurred to me that possibly some of these troubles might be due to the presence of air. A ship running to South America could not keep condenser tubes in a trip of about six weeks; it was necessary to retube a portion of the condensers at each end of the trip. The discharge from the air pump was located high in the hotwell, so that a great deal of air was taken in; the water was churned, charged with air, and pumped into the boilers. An elbow with standpipe was put in so that the water was delivered at the bottom of the feed-water tank, so that it did not produce the churning and drawing of air, and the corrosion practically stopped. There are two cases that have come to my attention where trouble has been had with condensers in stationary plants, where the stopping of the leaks in the air circulation by arranging a sheet-piling shield around the intake stopped or materially reduced the corrosion. Possibly the presence of air and its effect in corrosion have not been given sufficient attention.

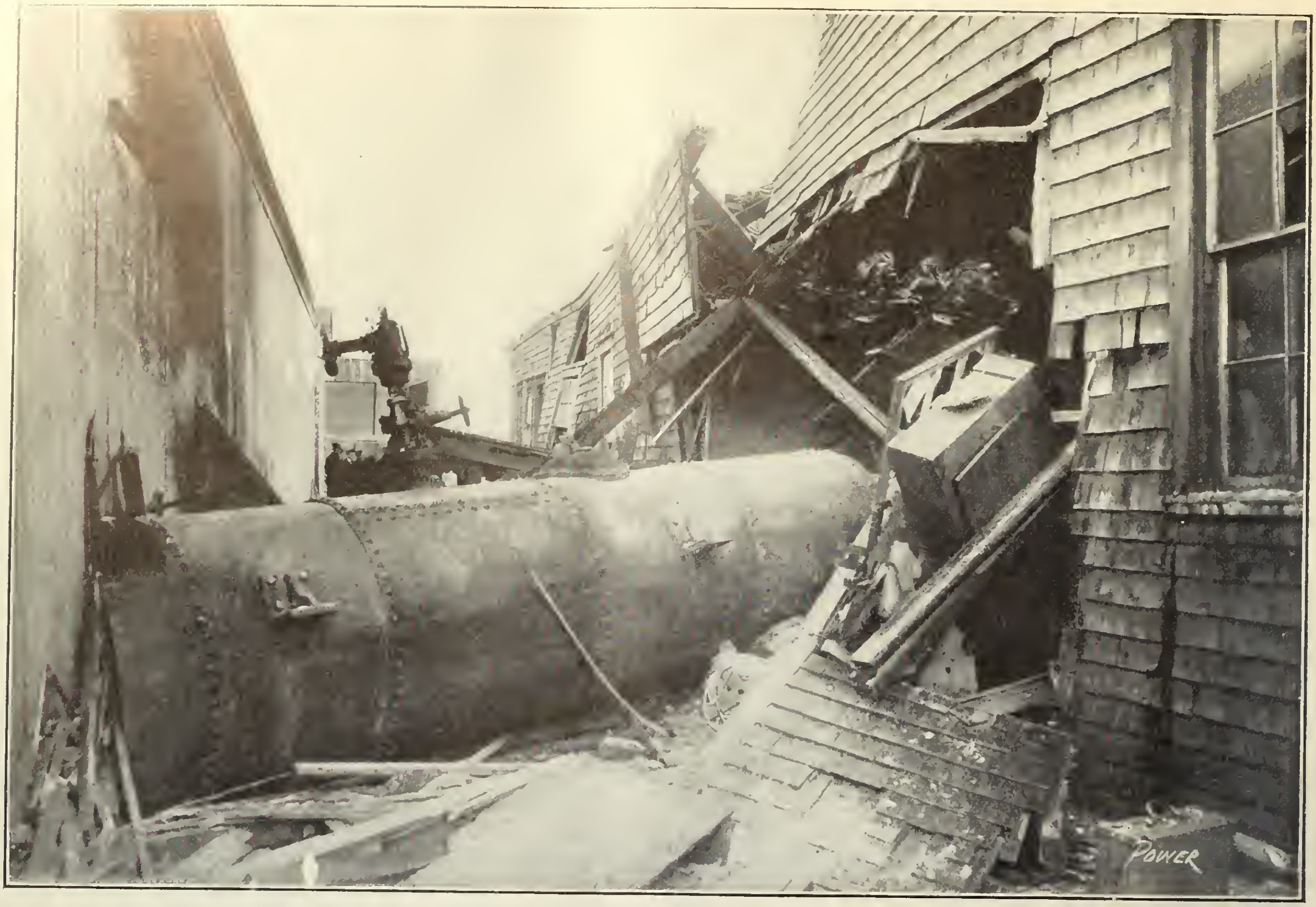

Fig. 1. Boiler Lying in Alley Between Mill and Water-works Shed Just as It Landed after Explosion

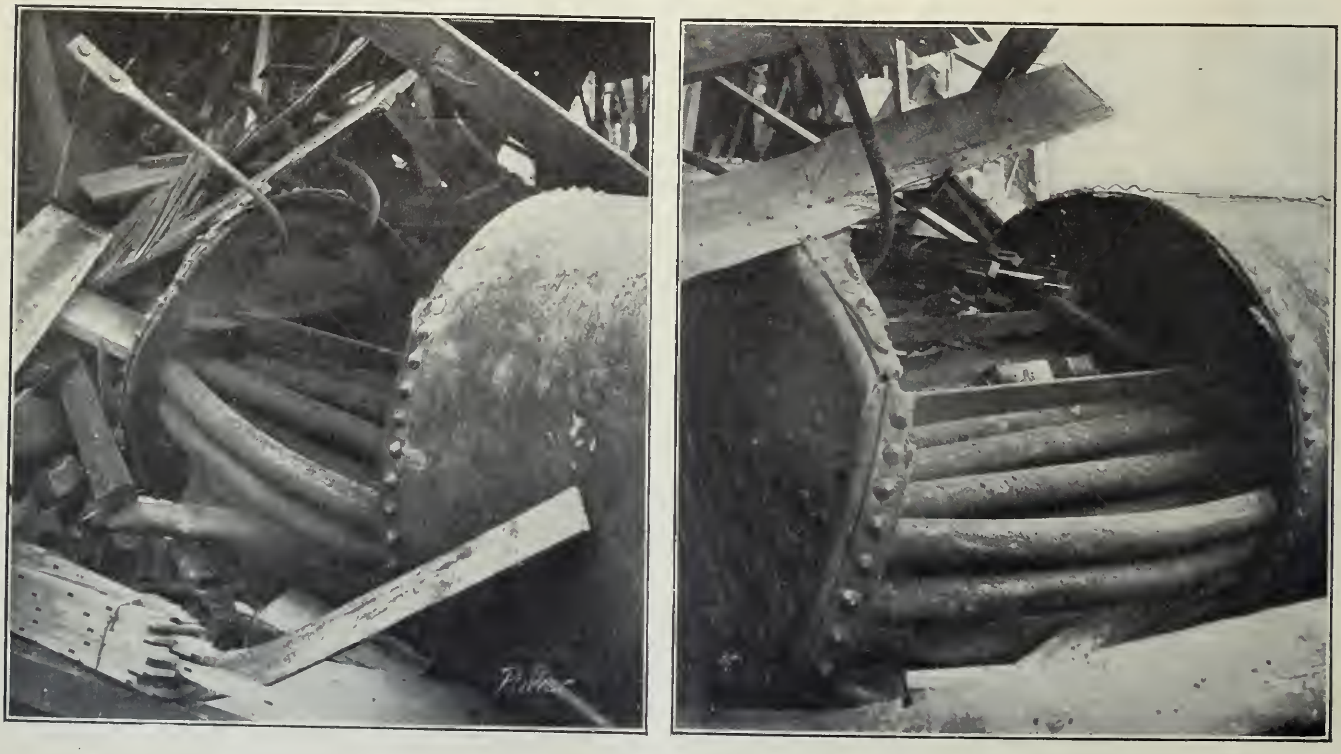

Fig. 2. Two Views of the Ruptured End, Showing How the Sheet was Torn through Solid Metal

Boiler Explosion in Massachusetts

Another case of lap seam boiler failure. The fifth disastrous explosion in Massachusetts within recent years and second in New Bedford, the others occurring at Fall River, Brockton and Lynn.

Though it is often the unexpected which happens, that which may be reasonably looked for also often occurs. This is exemplified in the fifth disastrous explosion in Massachusetts during recent years of a boiler of lap-seam construction.

At a few minutes past 11 o'clock Friday forenoon, December 9, the 42-inch by 16-foot return-tubular boiler in the planing mill of Arthur S. Allen & Co., North Water street, New Bedford, Mass., exploded, wrecking the building and dangerously, if not fatally, injuring Mr. Allen, and painfully scalding, bruising and cutting six of the ten employed in the mill at the time.

The boiler, which was installed in 1885, was of the common double-riveted lap-seam type, built in five courses with three supporting lugs on each side one in the middle of the alternate courses, the first, third and fifth course.

The sheets were 5/16 inch thick with 11/16-inch rivets with a pitch in the longitudinal seam, which with a rivet strength equal to the proportion of the plate uncut by rivet holes would give with a factor of safety of six a working pressure of 75 pounds per square inch. The tubes, 38 in number, were 3 inches in diameter and in good condition, clean and free from laminations or other defects.

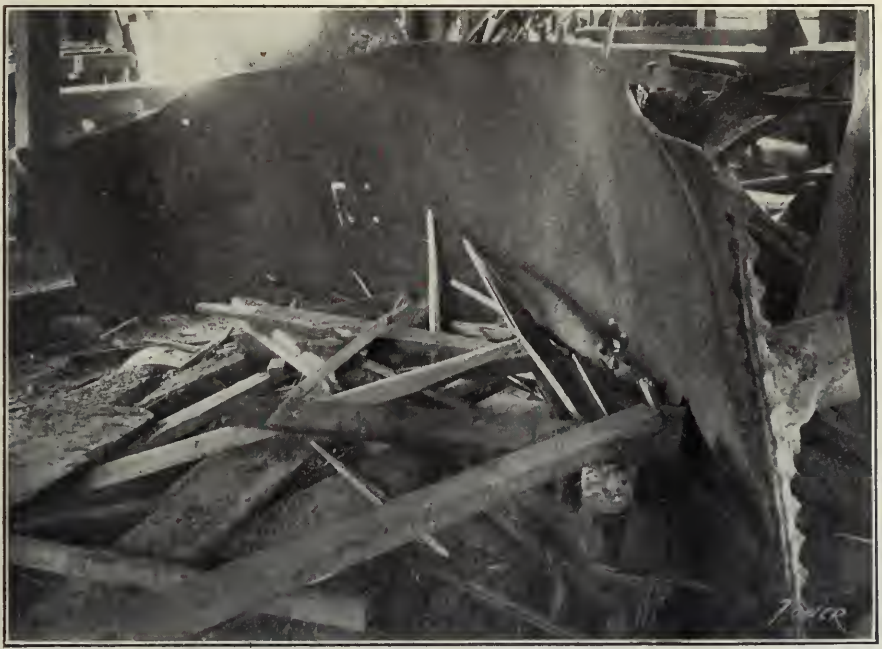

Fig. 3. Ruptured Sheet as It Landed

But one sheet, that making the rear course, was torn; not a rivet was sheared and but one pulled through. This was where the rear head cracked as the sheet was torn like wet paper away from the head and the course in front. It is plain that the initial rupture occurred at the longitudinal seam, the sheet tearing through solid metal away from the break and tearing downward enough in advance to allow the reaction of the steam and water rushing out of the rent to lift the boiler with a whirl and throw it through the side of the building, while the sheet went in a diagonal direction into the mill and landed among the machinery.

From the nature of its construction the lap seam hides the crack which inevitably must start sooner or later on the under side of the outside lap, close to the row of rivet heads.

There is no evidence that the boiler had anything but good and intelligent care. Two or three minutes before the explosion the fireman noticed that the steam gage registered between 60 and 65 pounds pressure and was probably correct, as the boiler had been inspected about three weeks previously. No investigation by the proper authorities has yet been made, and, so far as can be learned, no steps have been taken to fix the responsibility for the accident. There is a newspaper story to the effect that Mr. Allen had been notified by the inspector of the company by which the boiler was insured that it was unsafe and that a new one should be installed or the working pressure reduced. Mr. Allen's reply to this warning was the transfer of his insurance to another company.

It is unfortunate, if this story be true, that the specific section of Acts of 1908 of the General Court of the Commonwealth of Massachusetts, referring to this particular action on the part of a boiler owner, did not become a part of the law in the beginning.

Central Station Insurance

At the St. Louis meeting of the National Electric Light Association, statistics were presented by W. H. Blood, Jr., insurance expert of the association, tending to show that fire-insurance premiums paid by the electric-lighting companies of the country are fully twice as high as the low ratio of loss would warrant. Resolutions were adopted by the association asking for an investigation of the subject by the fire-insurance companies and directing that copies of the resolution be furnished to member operating companies so that they might take up the matter with their brokers. A thorough campaign along these lines has recently been organized as fitting at this season of the year, and large numbers of copies of the resolution have been placed in the hands of the operating companies, requesting them to bring the matter to the attention of their insurance brokers. Letters have also been written from association headquarters to some 250 insurance companies, calling their attention to the matter and inviting them to investigate the subject with a view to a reduction in rates commensurate with the premiums received and the risks involved. An extremely interesting report and analysis of the subject generally was presented at St. Louis by Mr. Blood and has been reprinted in the annual proceedings of the association.

Undoubtedly, states the *Scientific American*, the most troublesome parts of electric generating machinery are the armatures and rotors. Insurance records for 1909 placed the armature and rotor difficulties at 40 per cent., and commutator and slip-ring troubles at 29 per cent. In direct-current motors 39 per cent. of the failures were due to the armatures and 31 per cent. to the commutators, while in alternating-current motors rotor defects amounted to 23 per cent., and slip-ring and commutator troubles to 6 per cent., and 49 per cent. of the defects originated in the stator coils.

Inquiries of General Interest

Area and Mean Effective Pressure

How is the mean effective pressure in the cylinder found from an indicator diagram after the area has been found with a planimeter?

A. M. P.

If the area of the diagram is divided by its length, the quotient will be its average hight, and this multiplied by the scale of the spring will be the mean effective pressure.

Water for Condensing Engine

A condensing engine requires 800 cubic feet of water to be fed to the boilers in a day. What will be the quantity of water needed in a jet condenser to condense the steam?

W. C. E.

It will depend on the temperature of the exhaust steam and of the injection water. Seventeen pounds of water for each pound of steam condensed will be sufficient under average conditions, but it should be possible to furnish thirty in an emergency.

Weight of Bituminous Coal

What is the weight of a cubic foot of soft coal and how many cubic feet in a ton?

W. B. C.

Its weight varies from 50 to 55 pounds per cubic foot. Forty cubic feet of bunker space is usually allowed per ton.

Safe Receiver Pressure

With a triple-expansion engine, how is the safe receiver pressure determined?

S. R. P.

The receivers of most, if not all, engines where the cutoff is subject to hand regulation are provided with safety valves set by the builder at a point of release which will prevent a dangerous pressure.

Reducing Diameter of Eccentric

If ¼ inch is turned off an eccentric, would it alter the travel of the valve?

R. D. E.

It would not.

Rollins Engine Valves

What kind of valves has the Rollins engine?

R. E. V.

The steam valves are multi-ported grid and the exhaust are double-ported valves of the Corliss type.

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Water Disappears from Gage Glass

With the gage glass half full of water, the blowoff cock and all other connections to the boiler tight, the fire was banked so heavily at night that it went out. In the morning there was no water to be seen in the glass. What became of it?

W. D. G.

Such an occurrence is highly improbable. If it did occur it is likely that the heat of the setting evaporated enough water to lower it out of sight and this evaporation was condensed in the steam pipes which the question does not say were shut off from the boiler. This assumes that there was no leak at seams or elsewhere.

Formulas for Pulley Diameters

What are the rules for calculating the revolutions and diameters of pulleys?

F. P. D.

The calculations are simple proportions. In deriving formulas for the various conditions let

D = Diameter of driving pulley;
R = Revolutions per minute of driving pulley;
d = Diameter of driven pulley;
r = Revolutions per minute of driven pulley.

Then

$$D : d :: r : R$$

from which with any three of the factors being given the fourth may be found. To find the size of the driven pulley,

$$\frac{D \times R}{r} = d$$

To find the size of driving pulley,

$$\frac{d \times r}{R} = D$$

To find the revolutions per minute of the driven pulley,

$$\frac{D \times R}{d} = r$$

To find the revolutions per minute of the driving pulley,

$$\frac{d \times r}{D} = R$$

Positive Cutoff Engine

What is a positive cutoff engine?

P. C. E.

It is an engine in which the steam is cut off at the same point in the stroke and the speed is controlled by throttling.

Riding Cutoff

If a riding cutoff valve is set to cut off the steam at one-half stroke on the outward stroke, will it cut off at the same point on the return stroke? If not, what can be done to make it?

R. C. O.

If the valve stem and eccentric rod are of the right length, the cutoff will occur at the same point in both strokes. Corrections may be made by making the rod length right and then turning the eccentric until the cutoff takes place at the right point.

Inside Lead

If a valve is given inside lead, what effect will it have?

I. V. L.

The exhaust will open earlier and close later in the stroke.

Change of Engine Speed

How can the speed of an engine be increased by the governor?

C. E. S.

Slight increase in speed may be obtained by weighting the governor. But if any great change is desired the pulley on the governor shaft must be increased in diameter.

Diameter of Steam and Exhaust Pipes

What proportion should the diameters of the steam and exhaust pipes bear to the cylinder diameter?

D. S. P.

For a piston speed in the neighborhood of 600 feet per minute the diameters should be 0.3 and 0.4 that of the cylinder respectively.

Average and Mean Effective Pressure

What is the difference between the average pressure and the mean effective pressure on a piston?

C. N. D.

The mean effective pressure is the average pressure urging the piston forward minus the back pressure or that tending to hold it back.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

Of this issue 34,000 copies are printed.

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Please do not ask us to start subscriptions before the December 6 issue. All previous issues are exhausted.

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Power
and The Engineer

Editorial

Limit Lap Seam to Ten Years

Reports of the explosion of lap-seam boilers with the attendant casualties, the killing and crippling of men and the destruction of property, follow one another so rapidly that the statement by someone that the barbarities of peace are greater than those of war seems justified. In the suffering caused by the explosion of boilers of lap-seam construction this barbarism is exemplified to a degree unknown in any other line or country other than the United States. It is a matter of common as well as shameful knowledge that license in the design, construction and operation of admittedly dangerous boilers obtains in this country that is tolerated in no other part of the civilized world.

Lap-seam boiler construction is particularly barbarous from the mechanical point of view, which shows that steel resents the treatment necessary to force it into the shapes demanded by this construction.

It is barbarous from the financial viewpoint because every boilermaker knows and every boiler buyer should know that for boilers of equal size and strength the lap-seam type costs more than those with the butt joint.

It is barbarous from the common-sense standpoint to make, install and operate any piece of apparatus, the disastrous failure of which is as inevitable as death.

In fact, it is barbarous from any point of view that may be taken—business, sentiment, history, patriotism and ethics.

The strength of the lap seam is ample for the purpose when new, but that it rapidly weakens with use is too well known to [illegible] discussion, and where boilers with this seam are in use, the miscalled factor of safety should be increased annually to such a degree that will make such a boiler useless for power purposes before it has reached the age of ten years.

Few lap-seam boilers have failed under ten years, though the first one to fail in the Acushnet mill, New Bedford, Mass., in 1897, was but seven years old and the Harney boiler, which exploded in Lynn, Mass., in 1906, had been in use just ten years.

It is probable that eighty per cent. of the boilers in use in this country are of the lap-seam type, the greater proportion of which are well on the road to certain destruction and should be retired by the enactment and enforcing of laws as drastic as are needed for the purpose. Legislation along the lines tending to public safety in the matter of steam-boiler construction and operation is farther advanced in Massachusetts than in any other State, but here the hands of the boiler-inspection department are tied by the stultifying regulation which permits the operation of any boiler on which any insurance company will carry a policy.

In the rules formulated by the Board of Boiler Rules of Massachusetts will be found the factors of safety to be used in calculating the working pressure which may be carried on lap-seam boilers, and it is notorious that this factor is but twenty per cent. higher on a boiler which may have been in use half a century than on one which was installed the day before the boiler rules became operative.

If this board has the power to determine the factor which shall be used, why has it not the power to use common engineering discretion and consider the probable condition of a lap seam which has been distorted thousands of times daily and fix the end of service in Massachusetts at least at not over ten years?

The Right Idea

The recent action of the educational committee of the National Association of Stationary Engineers No. 37, of Ohio, an account of which appears elsewhere in this issue, may be regarded as an advance step toward fulfilling one of the principal objects of such organizations—that of educating its members into becoming more proficient in their present occupations and fitting them for advancement. Heretofore it has been the custom for the State committees to arrange a program of lectures to be given during the season under the auspices of the various local branches; but, as a rule, the initiative has seldom been taken by the local associations themselves.

One great drawback to the custom of merely making advance announcement of the lecture topics has been that the discussions which usually follow such lectures are confined to a comparatively few members, the large majority of those present feeling that they are not sufficiently prepared to enter into a discussion of the topic under consideration, and in this way many valuable suggestions are never brought out. To get the full benefit from any discussion, one should enter into it. Hence, if only a few take part the rest lose interest and the main object is defeated. In the present instance, however, a list of questions has been compiled and circulated in connection with the announced topics, the questions being of such a nature as to open up new lines of thought and awaken the interest of all members. With such an arrangement many, who would otherwise be backward in expressing their views, are afforded an opportunity to prepare themselves in advance and as a result contribute materially to the value of the discussion. The Dayton local association has the right idea and it is to be hoped that others will follow its example.

Handling Men

Much time and space are taken in discussing the management of equipment, and properly so; but the management of the men who operate the equipment is a matter of no small importance. A crew of dissatisfied, badly managed men can cause severe losses in the best equipped plant, while a well managed crew, through watchfulness and cheerful painstaking, will make the log sheet show up results.

Nearly everyone can point to a plant that the best firemen and engineers shun because of some shortcoming in the management. And who does not know of a plant where, although the work is hard, a position is held desirable because "the old man is all right." The continual changing of help is not conducive to successful operation. Proper management will hold the best men, but how shall this be secured?

First of all, the work of the plant should be definitely divided among the men. Trouble is sure to follow if two or three men are allowed to take a hand at keying up. Let one man attend to that, and hold him responsible. Go over the whole plant and make a list of the routine work: the packing of engines and pumps, the cleaning of the generators, the windows, the brasswork, the care of filters, the CO_2 machine—everything that there is to be done—no matter how trivial. It will be a list whose size will surprise you. Then, go over it and see how many of the tasks are assigned—and how many are not. More surprises are now due. Then, see that every item is definitely assigned to someone—and hold that one responsible. Is it necessary to say that in this way work can be more evenly and justly divided between the men—and that greater efficiency is sure to follow?

There may be times during the day when a steam pressure of 120 pounds is best, and other times when there ought to be 140 or 160. But, to leave this matter to the discretion (or convenience) of the various firemen (as is actually done in some plants) is a palpable folly. Let a standard practice be determined in such matters and hold every man to the line. Do not have one shift oiling the valve gear every hour, another every three hours, and the other—sometimes. Have a standard. There is a receiver pressure (varying, of course, with the load) at which the engines will show best economy. Do not leave each man to his own devices. Have a standard. Of course, these standards need not be too rigid. An intelligent, interested operative should be encouraged to test new ideas or make experiments; but variations from standard practice should be for that purpose and not to suit the whim or indolence of the operative, and tests or experiments should always be followed by records and reports, else their value is lost. So, have standard-practice instructions for the boiler room and engine room, but do not be a slave to the standards.

When you have thus systematized the work, you may be surprised to find how many of the difficulties in the management of men have been eliminated. Petty jealousies caused by one man encroaching on another's "prerogative" will disappear. There will be no more "shouldering" of work upon another. Tale bearing and the disgusting sequels which always result will have no more reason for existence. Each man has his work to do. If it is not done, the fact is self-evident and no one needs to be told.

It seems unnecessary to urge that the working conditions in boiler and engine room be made as pleasant as possible. Yet there are chiefs who need this advice. So long as the office chair is soft and the spittoon handy, the ventilation or lighting of the boiler room troubles such men not at all. Sometimes one is found who seems to imagine that the less comfort the fireman has the better he will attend to business. Experience teaches most of us a different lesson.

Make few rules and have them simple and explicit. Have them typewritten and posted for all to read. See that they are obeyed. How often does a rule become a dead letter almost as soon as it is made. Such proceedings are not merely useless, they are positively harmful. The indifference engendered communicates itself to other matters. If you have not the time or inclination to see that a rule is obeyed, better not make it.

If a man does a thing especially well, tell him so. Occasionally one needs to use discretion in this matter; but because it might spoil a "swelled head" do not withhold praise from the other fellow.

Frown on the tale bearer—or better, sit on him. The "sucker" is usually a poor workman and takes this means to "make good." Remember this, the same fellow who will bring stories to you, will carry stories of you to others. A man crooked enough to be a tale bearer is too crooked to stick to the truth.

Handling men and station management are subjects so vital to the efficient operation of a plant that, although some discussion has appeared in these columns, we would be glad to hear from others. Two heads are better than one and a dozen better still.

Everything from the Central

Sebastian Ziani de Ferranti showed so much foresight and appreciation of the possibilities of electricity in anticipating the use of high voltages and of large direct-connected units that anything which he may say upon that subject is bound to receive a respectful hearing and more than ordinary consideration, however radical and improbable it may appear. In an address delivered as president of the Institution of Electrical Engineers, Mr. Ferranti has just made the proposition that the whole of the coal which we use for heat and power should be converted into electricity. The electricity is to be distributed by mains and utilized for all industrial and domestic purposes in which heat, light or power is required. The smelting of metals in blast furnaces is even to be more economically effected by electric furnaces. The use of the nitrates obtained in the byproduct from the numerous generating stations will result in the intensification of agriculture and contribute largely to an extended and beneficial employment of labor, as well as enabling the country to obtain all its necessary food without the need of importation and at less cost. In this way are realized coal conservation, home-grown food and the better utilization of labor, which are the three desirables pointed out by Mr. Ferranti's paper.

For this general idea to come about it is necessary to have an efficiency of conversion of at least 25 per cent. of the energy of the coal into electrical energy. At present Mr. Ferranti believes that the amount which we get back and use for work of one kind or another is much less than ten per cent. of the energy of the coal. He does not expect that this will be accomplished by the use of internal-combustion engines, and believes the steam gas turbine in which steam is used as a gas at high temperature will, in the course of time, supply the necessary means. A very high load factor will be obtained, Mr. Ferranti says 60 per cent., and machinery of the normal capacity of 25,000,000 kilowatts would have to be installed to produce a supply estimated at 151,400,000,000 kilowatt-hours as a basis upon which the scheme would be commercially profitable in Great Britain. Mr. Ferranti estimates that the scheme he proposes would enable electricity to be supplied throughout the country at the average price of one-eighth of a penny, or one-fourth of a cent per kilowatt-hour, and that it would only be a matter of time for all heating and cooking to be done by that means.

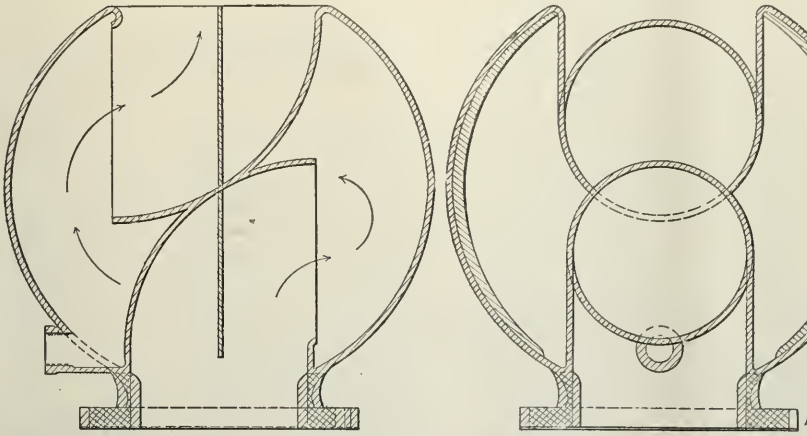
New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

The R-K Exhaust Head

This head is made in one solid piece of cast iron, external and internal, and there are, therefore, no internal parts to rust, corrode or become loose.

The illustration shows sectional views of the head, also the three changes in direction of exhaust steam before leaving the head. The steam on entering the head is immediately changed in its direction of travel and then expands into larger volume during the next change of direction and all moisture clinging to the outer surface falls to the bottom of the head and is led off through the drip pipe.



SECTIONAL VIEWS OF "R K" EXHAUST HEAD

The steam is once more changed in its direction of travel and leaves the exhaust head through a larger opening than the inlet of the same at the top.

Sizes up to and including 4 inches are for screwed pipe connections; all sizes above this are flanged.

This exhaust head is manufactured by Walter G. Ruggles Company, 54 High street, Boston, Mass.

Sarco Type "G" Multiple CO_2 Recorder

The Sarco multiple unit CO_2 recorder is designed to make continuous determinations of the CO_2 content in furnace gases and in its several sizes will make separate and simultaneous records from one up to ten boilers.

Furnace gas is led to the CO_2 recorder through individual pipe lines, each of which is connected to a seal *A*, Fig. 1. Each gas line has a ¾-inch sampling tube perforated with holes extending into the last pass of the boiler, and just outside the flue or boiler setting a filter is placed to remove soot from the gas. From the filters, a ⅜-inch pipe is used to conduct the gas to each unit of the recorder, the small diameter being used to reduce time lag, and because no clogging will occur as filtering and draining of moisture are done at the boilers.

Suction is produced by means of the motor-driven suction fan *B*, located at the bottom of the machine. A steady stream of gas is drawn through the seals *A* and internal gas line *C*, which is connected to the seals. The object of the latter is to produce an equal degree of suction for each unit which can be observed, as the seals are made of heavy glass tubing and the gas bubbles through the paraffin oil in the seals, the inlet pipes *D* being perforated. Before passing to each unit, the gas is again filtered at the internal filters *E*, which are cast-iron cylinders containing a brush through which the gas must pass.

The motive power for drawing the gas through each unit is provided by an overhead tank of paraffin oil *G*, located at the top of the machine. The oil is led to the units by the line *H*, and, by virtue of its head, draws samples of gas through the tees located between the seals and internal gas line. The gas passes through inlet lines *J*, Fig. 2, to the measuring or inlet burette *K*.

As the paraffin oil which is being constantly fed to the units gradually fills the pipe lines *L*, it rises in each measuring burette to the inlet connection *M* and prevents further gas being trapped. The oil continuing to rise forces the gas

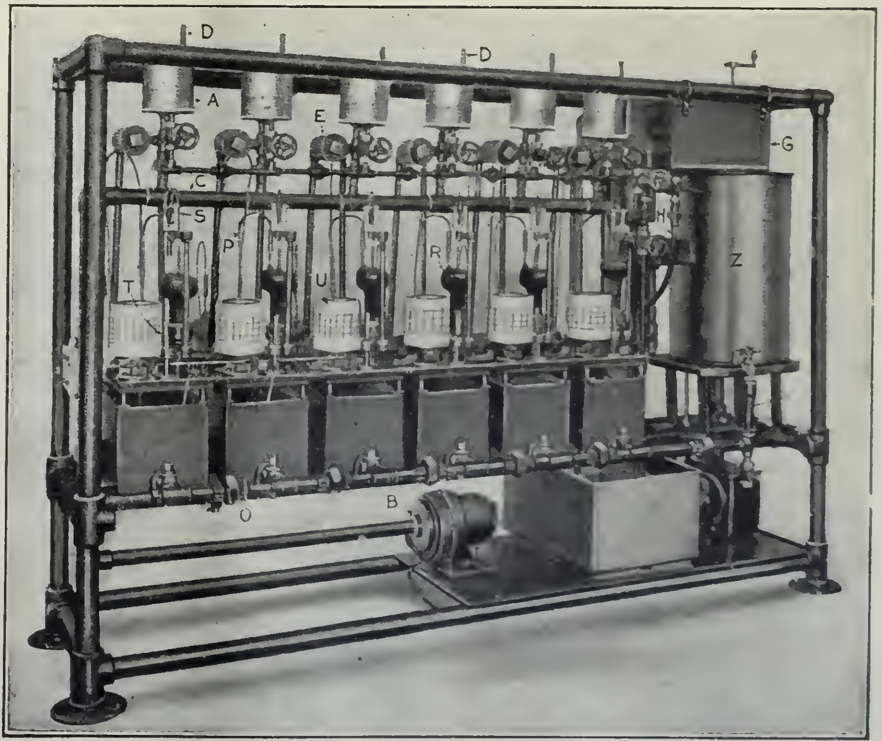


FIG. 1. SARCO MULTIPLE CO_2 RECORDER

through the capillary copper tubes *N* into the bottom of the absorption vessels *O*, Fig. 1, where it is deprived of its CO_2 content by being bubbled through the caustic potash solution contained therein. The remainder of the gas is forced up through the capillary tube *P* into the recording burettes *R*. The latter are also supplied with paraffin oil from the main supply and within each is suspended a metal float (not shown) which is frictionless. The gas, under pressure, forces the paraffin oil out of the burettes and the floats are carried down with the liquid.

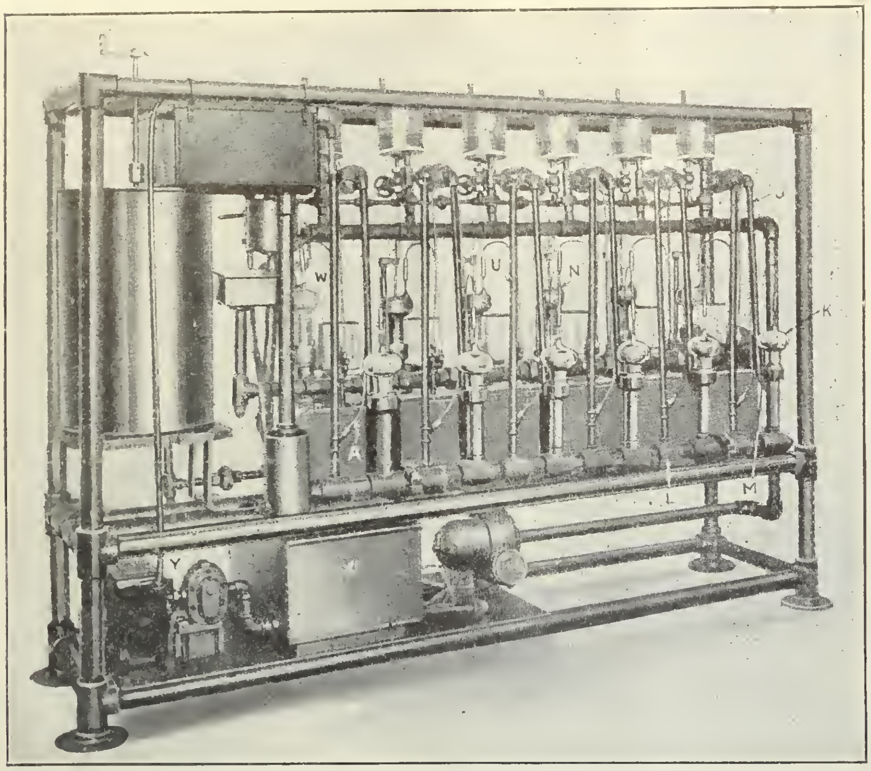


Fig. 2. View of Recorder from Opposite Side

Each float is suspended from the frictionless wheels *S* by means of strong threads which also carry the pen *T* for making the record of the CO_2 content in each sample on the charts *U*. While this is being accomplished the paraffin oil is continually rising in the risers *V*, Fig. 2, and central siphon *W*. The hight of the latter is such that as soon as a record is made, the siphon starts to flow and passes all the oil into the discharge tank *X*, where it is simply pumped up into the supply tank by means of the pump *Y*. The supply tank contains a filter to constantly cleanse the paraffin oil, and also has an overflow communicating to the discharge tank.

Caustic potash is placed in the tank *Z*, water added and the whole thoroughly mixed after dissolving. The solution is then fed to each absorption vessel through the line and plug cocks shut and if desired the remaining solution can be run off through valve (27) and the system washed with water to prevent crystallization of the solution which will stick to the inside of the tank and pipe line. When the solution in each vessel is saturated with the CO_2 gas and will absorb no more, the plug cocks are opened and the solution run off as before.

When the siphon starts to flow, the paraffin oil in the system having the pressure reduced recedes from the measuring burettes. The gas which has forced the float downward is then pressed back into the absorption vessel at the top through the tube *P* and leaves the vessel by the exit tube *A'*, whence it reaches the atmosphere by the riser tubes *U*.

The recording charts are mounted on drums which are driven by worm gears on a clock shaft. The latter is driven by a powerful clock. To remove the chart drums it is only necessary to lift them from their base plates. The latter receive their motion from spur gears located on their shank driven by the worms on the clock shaft.

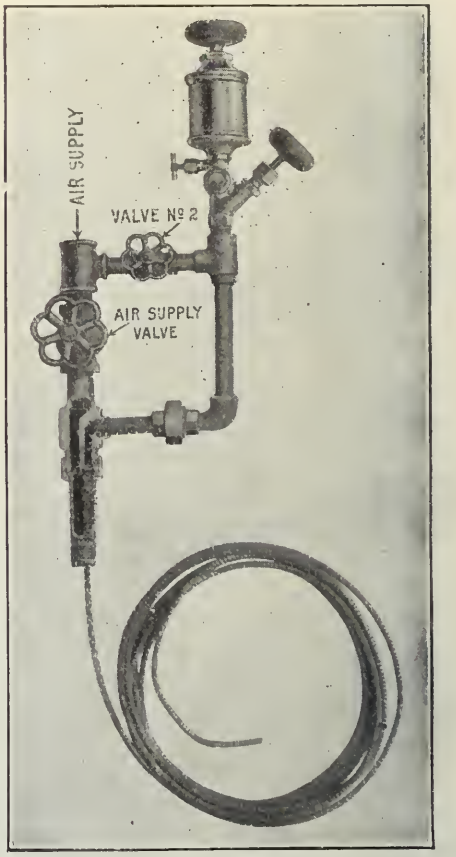
As soon as the siphon has stopped flowing one cycle is completed, the time being about three minutes if 20 records per hour are to be taken. The gas is again drawn through the inlets to each unit and another operation commences. The pen is automatically inked by returning to an ink-well after each record.

As to renewals of solutions—the paraffin oil is never changed; the ink-well has a capacity for a week's run. The caustic solution, if made to about 1.25 specific gravity, will last ten days on 24-hour shifts. The only adjustment necessary is setting the pen to zero when running on air. Everything else, except the winding of the clock, chart and caustic renewals, is automatic.

This CO_2 recorder is manufactured by the Sarco Fuel Saving and Engineering Company, West Street building, New York City.

Loganda Automatic Lubricator

It sometimes happens that when exceptionally hard scale is being bored out, the oil in the reservoir of the Lagonda tube cleaner will be exhausted before a tube is finished and the cleaner will run hot before the operator notices it. In order to provide against this difficulty the Lagonda Manufacturing Company, Springfield, O., has brought out a new automatic lubricator shown in the illustration. This lubricator has the advantage that the cleaner does not have to be withdrawn from the tube in order to replenish the oil supply. Furthermore, the sight-feed oil cup and suitable valves make it possible to exactly keep track of the rate of the oil feed and to adjust the supply to the needs of the cleaner.



Lubricator for Tube Cleaner

This lubricator can be attached to the air-supply pipe in any convenient place, and consists of a sight-feed oil cup and two valves. As practically all air tools use flexible rubber hose, it is inadvisable to mix the oil directly with the air and blow it through the entire hose, as oil causes rubber to deteriorate rapidly. The lubricator is provided with a separate flexible metal hose which conveys the oil directly to the tool and is placed inside of the rubber air hose.

The rate of oil feed can be regulated by the cock at the bottom of the oil cup, then by opening valve No. 2, connected to the air supply, this oil is thoroughly mixed with a small portion of air and blown through the inner tube to the boiler-tube cleaner or other pneumatic tool.

This lubricator can be placed on the wall or any other rigid object, where it will not be in the workmen's way or subject to derangement by the vibrating tool. It is stated that the thorough system of lubrication thus effected will increase the life of the tools by 50 per cent. and will save at least 50 per cent. of the oil, as the rate of feed can always be maintained proportional to the demand of the tool.

National Gas and Gasolene Engine Trades Association

The National Gas and Gasolene Engine Trades Association held its regular annual meeting at Racine, Wis., December 13, 14 and 15. The association has grown constantly from its start, each meeting being larger than the one preceding it, and its affairs are reported as being on a most substantial basis.

The papers presented at this meeting tended in general to the recommendation of more attention to detail on the part of the manufacturers of the various auxiliary devices with the object of raising the standard of reliability of service. In a paper entitled "Gas Engines," Frederick Lammert, of Lammert & Mann, Chicago, made a strong plea for higher-class auxiliary apparatus. Mr. Lammert called attention to the neat, compact, self-contained outfits furnished with automobiles and hoped that some such concentration and improved methods of installing auxiliary apparatus would be developed for the stationary field.

Charles Kratsch, of Colborne Manufacturing Company, Chicago, read a paper along similar lines entitled, "Does the Efficiency of the Gas Engine Depend on its Equipment?" Before giving the obvious and expected affirmative answer, Mr. Kratsch cited the records of his firm, which does an extensive business in placing second-hand gas engines and whose trouble department is unusually well qualified to speak on the subject of gas-engine defects. It was brought out that 29 per cent. of the breakdowns of gas-engine plants within his experience was due to the auxiliary devices. He cited cases where engines costing thousands of dollars had the cheapest of electrical equipment, wiring, etc., all of which is a detriment to the gas-engine business in general.

"Simplicity, the Basis of Reliability," was the title of a paper by J. C. Miller, of Mariner & Hoskins, Chicago. Mr. Miller urged that the strongest possible efforts be made to simplify the equipment in gas-engine plants, as this was really the solution of most of the difficulties experienced with gas engines. It was also pointed out that as the supply of gasolene is steadily decreasing, the question of utilization of kerosene and benzol, a byproduct of coke ovens, be taken up so that when the time comes that gasolene will actually go out of use, the present manufacturer will be ready to meet the question of fuel supply for his engine.

In a paper entitled "Gas Producers," H. F. Smith, of the Smith Gas Power Company, Lexington, O., emphasized the importance of attention to details in the design of producer plants. He said that the fact that a plant was running and had been accepted by the purchaser could not always be taken as evidence of the complete success of the design, and, in fact, the status of the producer power plant as revealed by the inside history of the business was not entirely satisfactory. Troubles with this class of installation were laid largely to lack of attention to details of manufacture and also to the prevailing idea that a producer is nothing more than a modified hard-coal stove. Purchasers have in the past been misled by this claim of simplicity to the great detriment of the business. It was pointed out that in reality there is a vast amount of essential detail in producer design, the subject being just as complex and far-reaching as any other branch of engineering. Installation defects have also been the cause of much dissatisfaction with producer plants, while the minor equipment and the practice of putting 15-cent auxiliaries on $15,000 engines was another frequent cause of failure. Mr. Smith said that in his estimation 60 per cent. of the cases of trouble in producer plants was due to the auxiliaries.

Officers for the coming year were elected as follows: C. O. Hamilton, of Elyria, O., president; O. C. Parker, of Racine, Wis., vice-president; O. M. Knoblock, of South Bend, Ind., treasurer; Albert Stritmatter, of Cincinnati, secretary.

Detroit was chosen as the place of next meeting, which will be held some time in June, 1911.

Profit Sharing and Pension Plan

The directors of the Edison Electric Illuminating Company, of Brooklyn, in recognition of faithful and efficient service and for the encouragement of thrift and investment in the securities of the company on the part of the employees, has authorized a plan of profit-sharing and pensions as an addition to the welfare plans now in operation.

At the end of the year 1910, employees will be credited with a sum out of the profits of the company's operations for that year, in accordance with the following schedule:

To those employees who have been in the company's service two full calendar years, a percentage of their salary or wages for the year 1910, equivalent to one-quarter of the rate of dividends paid on the capital stock during the year; to those employees who have been in the company's service three years, a percentage equivalent to one-half of the rate of dividends; for four years, a percentage equivalent to three-quarters of the rate of dividends, and for five years, or more, a percentage equivalent to the full dividend rate.

This profit-sharing plan is adopted for the current year only, and it will continue from year to year only as the board of directors may adopt it for each year.

OBITUARY

Walter L. Pierce, who for thirty-two years had been connected with the Lidgerwood Manufacturing Company and for twenty-nine years its secretary and general manager, died suddenly of heart failure at his winter home in the Hotel St. Andrews, New York City, in the early hours of Saturday, December 10, 1910. He was a son of John F. Pierce and was born at Dorchester, Mass., on June 8 1855. His parents survive him and he leaves a widow, Jane Hutchins, an only son, Walter L. S. Pierce, a brother Charles C. Pierce, and a sister, Mrs. E W. Jones.

Mr. Pierce's death was entirely unexpected and was a great shock to his family and associates. He had suffered for several years with nervous troubles but by devoting much of his time to out of-door pursuits he had apparently recovered. He was feeling particularly well when he left his office on the evening preceding his death.

He was known to a wide circle of personal and business associates. He wa remarkable as an organizer and so perfect was his work that no detail of th great business which grew up under hi hand was neglected during his long absences from his desk while seekin health, and the coherent body which h formed is a monument to the efficienc of his work. Besides his connectio with the Lidgerwood Manufacturing Company, he was treasurer of the Haywar Company and of the Gorton-Lidgerwoo Company.

His summer home was at Englewood N. J., where he was a member of th Englewood Country Club. He was als a member of the Apawamis Golf Club the Wright Fish and Game Club, of Canada, the Lawyers Club, the Engineers Club, the Machinery Club, in which he was also a director, an associate member of the Naval Architects and Marine Engineers and of the American Society of Mechanical Engineers and a past president of the National Metals Trades Association.

George F. Seward, president of the Fidelity and Casualty Company, died on November 28 at the age of seventy. He was a nephew of William H. Seward, Lincoln's Secretary of War, and spent a number of years in the diplomatic service, first as consul at several cities in the Orient and later as minister to China. During this period of service he very effectively handled several perplexing diplomatic problems.

In 1887 Mr. Seward entered the insurance business, being elected vice-president of the Fidelity and Casualty Company, and five years later succeeded to the presidency of that company. In spite of an active business career he found time to devote to public and social interests and was at one time vice-president of the New York Chamber of Commerce. Among the societies of which he was an active member are the American Geographical Society, the American Academy of Political and Social Science, the American Institute of Civics, the Electro-Chemical Society, the National Municipal League and the Merchants' Association of New York.

Mr. Seward's successor as president of the Fidelity and Casualty Company is Robert J. Hillas, who entered the service of the company in 1876, at the age of sixteen, as office boy and gradually rose to the position of vice-president and secretary about nine years ago.

PERSONAL

Joseph Breslove, formerly of the engineering department of the Westinghouse Machine Company, has joined with A. Setlotter and taken over the business of the Universal Standard Electric Company, 90 Verona street, Brooklyn, N. Y. It is the company's intention to engage in a general mechanical and electrical-engineering and manufacturing business.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

FLUID-ACTUATED MOTOR. John S. Ward, Chicago, Ill. 977,574.

ELASTIC-FLUID TURBINE. Charles W. Dake, Grand Rapids, Mich. 977,601.

GAS ENGINE HAVING AUTOMATIC EXHAUST OPENINGS THEREIN. Andrew Matson, Los Angeles, Cal. 977,667.

GAS ENGINE. Edwin E. Savard and Winfred E. Jones, Rochester, N. Y. 977,752.

EXPLOSIVE ENGINE. Jesse H. Brown, Belding, Mich. 977,779.

INTERNAL COMBUSTION ENGINE. Lysander E. Wright, Newark, N. J. 977,847.

INTERNAL COMBUSTION ENGINE. Joseph Valentin Laviolette, Amsterdam, Netherlands, assignor to Industrieele Maatschappij Trompenburg, Amsterdam, Netherlands (a Limited Company of Netherlands). 977,885.

ROTARY ENGINE. George F. Leiger, South Haven, Mich. 977,888.

TURBINE. Jan Procner, Pabianice, Russia. 977,899.

INTERNAL COMBUSTION ENGINE. Peter Peterson, San Francisco, Cal. 978,058.

ROTARY MOTOR. Henry F. Weinland, Springfield, Ohio, assignor to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 978,085.

ROTARY MOTOR. Henry F. Weinland, Springfield, Ohio, assignor to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 978,086.

BOILERS, FURNACES AND GAS PRODUCERS

BOILER. Ephraim M. Wilkinson, Oswego, N. Y. 977,915.

STEAM BOILER. Godfrey Engel, Brooklyn, N. Y. 978,009.

STEAM GENERATOR. Sidney John Ross and Harry Schofield, London, England, assignors of one-third to Oliver Prescott Macfarlane, one-third to Sidney J. Ross, and one-third to Harry Schofield, London, England. 978,061.

POWER-PLANT AUXILIARIES AND APPLIANCES

PRESSURE REDUCER. Charles G. Armstrong, Orange, N. J. 977,427.

PACKING FOR PISTON RODS. George F. Royer and Lewis F. Zweibel, Wilkes-Barre, Penn. 977,477.

VALVE. William E. Sloan, Chicago, Ill., assignor to Sloan Valve Company, Chicago, Ill., a Corporation of Illinois. 977,562.

VALVE. William H. Glenn, Hempstead, N. Y. 977,614.

PIPE COUPLING. Campbell P. Higgins, Roselle, N. J., assignor to the Babcock & Wilcox Company, Bayonne, N. J., a Corporation of New Jersey. 977,740.

VALVE. Gordon Land, Seattle, Wash., assignor to United States Flush Tank Company, Inc., Seattle, Wash., a Corporation of Washington. 977,807.

WATER-TUBE STEAM GENERATOR. Achille Bugnon, Paris, France. 977,927.

TRAP. William S. Elliott, Pittsburg, Penn. 977,943.

CALORIMETER. Alfred M. Parks, Philadelphia, Penn. 977,694.

STEAM INJECTOR. Robert Grundy Brooke, Macclesfield, England. 978,000.

CARBURETER. David M. Tilden, Springfield, Mass. 978,076.

TURBINE FOR DRIVING BOILER-TUBE CLEANERS. Henry F. Weinland and Hermon G. Weinland, Springfield, Ohio, assignors to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 978,089.

CARBONATING APPARATUS. Charles L. Bastian, Chicago, Ill. 978,103.

ELECTRICAL INVENTIONS AND APPLICATIONS

SYSTEM OF ELECTRIC MOTOR CONTROL. Wilbur H. Thompson, Wilkinsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 977,571.

ELECTRICAL EQUALIZER SYSTEM. William Cooper, Pittsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 977,598.

DYNAMO ELECTRIC MACHINE. John B. Wiard, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 977,768.

ELECTRICAL SWITCH. William P. Hammond, Passaic, N. J., assignor to William P. Hammond, Theodore A. Hammond and Woolsey A. Shepard, doing business under the firm name of Securo Manufacturing Company, Passaic, N. J. 977,797.

POWER PLANT TOOLS

PIPE WRENCH. Robert O. Dohm, Dane, Wis. 977,788.

PIPE WRENCH. Thomas James Nicholson, Kingston, Ontario, Canada. 977,896.

BOLT HOLDER AND WRENCH. Charles E. Wood, Danville, Ill., assignor of one-half to Orville J. Gunnell, Danville, Ill. 978,100.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., Col. E. D. Meier; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U. S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next meeting to be held September, 1911, in Boston, Mass.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., Carl S. Pearse, Denver, Colo.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Cincinnati, Ohio.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

INTERNAL COMBUSTION ENGINEERS' ASSOCIATION.

Pres., Arthur J. Frith; sec., Charles Kratsch, 416 W. Indiana St., Chicago. Meetings the second Friday in each month at Fraternity Halls, Chicago.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; acting sec., Charles P. Crowe, Ohio State University, Columbus, Ohio. Next meeting, Youngstown, Ohio, May 18 and 19, 1911.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., J. G. Hannahan, Chicago, Ill. Next meeting at St. Paul, Minn., September, 1911.

NATIONAL DISTRICT HEATING ASSOCIATION

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

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